# Development of Wayside Rail Curve Lubrication Model for Australian Heavy Haul Lines

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Doctor of Philosophy

By

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# ABSTRACT

Wheel and rail wear and ineffective lubrication are serious concerns for rail operators around the globe. These assets have to be maintained and operated effectively to maximise the benefits to the stakeholders and minimise the cost to rail operators. Proper management of the wheel/rail interface helps the rail industry to grow business, improve reliability of service and commitment to their clients. Rail wear and effective lubrication are significant issues that have direct impact on operating costs of the rail operation. Wear causes detrimental effects on rail/wheel life and maintenance costs. Thus a better understanding of causes, mechanisms and effects of wear at the wheel/rail interface is necessary. Lubrication is considered as one of the most effective maintenance programs to reduce wear, energy consumption and noise. Therefore, implementation of an effective lubrication program is necessary to achieve cost effective rail operation.

The wayside lubrication method is widely used in the rail industry. In the past, lubricators were installed either too close together or too far apart. The literature indicates that quite a few approaches have been attempted for developing an effective placement model for wayside lubricators. There are different types of lubricators, lubricants and a variety of lubrication strategies across different networks, and also within the same network. A critical literature review revealed that there is a need for a more scientific approach to develop a reliable rail curve lubrication model. Keeping this fact in view, field investigations were carried out on the live track in the current study with the assistance of maintenance team of the Australian heavy haul network operator~~s~~. The first observation revealed that the grease carried by the wheels lasts only a few meters from the wayside lubricator sites in many curves. The performance of lubricants in the track can vary significantly depending on the weather conditions, track characteristics, dispensing equipment, type of lubricant being used and maintenance activities. In this field study it is clearly identified that there is a need for an improved understanding of the effect of lubricator performance, applicator bars (short and long bars) and locations of the bars based on track geometry, direction of traffic etc. Proper application of wayside lubricators also includes selection of appropriate equipment and a suitable lubricant for the known operating conditions, measurement and management of the lubrication effectiveness, positioning of lubricators and their appropriate maintenance.

Wayside lubricator spacing for maximum benefits is directly related to the lubricant carry distance. Whereas, carry distance is affected by track type, traffic, lubricant and the lubricator itself. Hence, there is a clear need to develop an improved lubricator placement model based on the combined effects of lubricants, applicator bars, lubricator, locations and track/traffic characteristics. The scope of this study also includes the survey of the current practices of curve lubrication and assesses their effectiveness; develop a hierarchical wayside lubricator placement model based on the evaluation of effectiveness and cost/benefit analysis and set up a standard practice for lubrication of heavy haul lines. A detailed review on currently available technologies applicable for heavy haul lines was performed. This was followed by the field investigations for real data collection to confirm the performance evaluation, development of lubricator placement model, cost/benefit analysis, model development of the grease transport mechanism, and optimisation of modelled parameters. The study concluded that electric lubricators are highly reliable and effective compared to older technologies, long applicator bars in the tangent track with high quality grease generate the longest carry distances and short applicator bars in the transition curve, even with high quality grease, didn't achieve long carry distances.

The rate of grease application is very crucial from the perspective of both effectiveness and economic returns. It is also known that equipment, their limitations, performance, appropriate location, quality of installation and their maintenance, various properties of greases and column strength of the grease bead contribute significantly to the grease transport mechanism. This study suggests that to ensure the effectiveness of way side curve lubrication, remote condition monitoring could be an effective tool which can significantly improve the reliability, maintainability and operating cost of electric lubricators.

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# NOMENCLATURE

**Abbreviations**

| | |
|---|---|
| ARA | Australasian Railway Association |
| AAR | Association of American Railroads |
| RAMS | Reliability, Availability, Maintainability and Safety |
| BITRE | Bureau of Infrastructure, Transport and Regional Economics |
| CRC | Cooperative Research Centres |
| QR | Queensland rail |
| RCF | Rolling Contact Fatigue |
| EHL | Elastohydrodynamic Lubrication |
| LCC | Life Cycle Cost |
| MGT | Million Gross Tonne |
| FAST | Facility for Accelerated Service Testing |
| TTCI | Transport Technology Centre Inc. |
| EP | Extreme Pressure |
| NLGI | National Lubricating Grease Institute |
| incr | Increase |
| OHSE | Occupational Health Safety & Environment |
| COF | Coefficient of friction |
| ACOF | Average Coefficient of Friction |
| DOE | Design of Experiment |
| ASTM | American Society of Testing Materials |
| TOR | Top of Rail |
| GF | Gauge Face |
| Hi | High |
| Lo | Low |
| 2 SB-ER | 2 Short Bars on High Rail |
| 1 LB-ER | 1 Long Bar on Each Rail |
| 2 LB-ER | 2 Long Bars on Each Rail |

**Nomenclature**

| **Symbols** | **Description** |
|---|---|
| H | Oil Viscosity (cSt) |
| V | Sliding Velocity (m/sec) |
| W | Normal Load (N) |
| $R_{q,a}$ | rms surface finish of surface *a* (micrometre) |
| $R_{q,b}$ | rms surface finish of surface *b* (micrometre) |
| Λ (Lamda) ratio | Dimensionless Film Parameter |
| $h_{min}$ | Minimum Film Thickness |
| $C_{dd}$ | Design & Development Cost ($) |
| $C_p$ | Purchasing Cost ($) |
| $C_i$ | Installation Cost ($) |
| $C_a$ | Accessories Cost ($) |
| $C_l$ | Lubricant Cost ($) |
| $C_{lab}$ | Servicing Labour Cost ($) |
| $C_v$ | Servicing Truck Operating Cost ($) |
| $C_{sp}$ | Cost of Spare Parts ($) |
| $C_{ps}$ | Purchasing and set up cost of lubricator ($) |
| $C_e$ | Electricity consumption cost ($) |
| $C_{sm}$ | Servicing & maintenance cost ($) |
| $C_r$ | Repair cost ($) |
| $C_v$ | Vehicle Cost ($) |
| $C_t$ | Travelling cost /fuel cost ($) |
| $C_{em}$ | Emergency maintenance cost ($) |
| $C_l$ | Lubricant cost ($) |
| $C_{sp}$ | Cost of spare parts ($) |
| $C_{grind}$ | Grinding cycle cost ($) |
| $C_{rep}$ | Replacement cost ($) |
| $C_{lw}$ | Lubricant cost for wastage of lubricant ($) |
| $C_{downtime}$ | Track downtime cost ($) |
| $C_{breakdown}$ | Rail/wheel life loss due to breakdown of unit ($) |

| | |
|---|---|
| $C_{risk}$ | Risk cost due to derailments and incidents ($) |
| $C_{sm,i}$ | Servicing & maintenance cost/ service interval ($) |
| $C_{v,i}$ | Vehicle cost per service interval ($) |
| $C_{t,i}$ | Travelling cost per service interval ($) |
| $C_{lab, i}$ | Labour cost per service interval ($) |
| $C_{em,i}$ | Emergency maintenance cost per service interval ($) |
| $C_{sp,i}$ | Cost of spare parts per service interval ($) |
| $C_{lw,i}$ | Lubricant cost for wastage per service interval ($) |
| $C_{sm,yr}$ | Servicing and maintenance cost per year ($) |
| $C_{v,i}$ | Vehicle cost per emergency maintenance ($) |
| $C_{t,i}$ | Travelling cost per emergency maintenance ($) |
| $C_{r,i}$ | Repair cost per emergency maintenance ($) |
| $C_{em,yr}$ | Emergency maintenance cost per year ($) |
| $C_{grind, yr}$ | Grinding Cycle Cost per year ($) |
| $\mathrm{C}_{\mathrm{annum, t}}$ | Annual cost of lubricator at time, t ($) |
| C | Original length of curve (m) |
| C+S | Equivalent curve length for each curve (m) |
| $\mathrm{L}_{\mathrm{perf}}$ | Lubricator Performance Factor |
| P | Applicator bar Factor |
| G | Grease Performance Factor |
| $\mathrm{P}_{\mathrm{prof}}$ | Wheel/Rail Profile Factor |
| R | Radius Factor |
| T | Traffic Factor |
| $\mathrm{B}_{\mathrm{G}}$ | Bogie Factor |
| A | Axle Load Factor |
| L | Locomotive Factor |
| V | Speed Factor |
| M | Bogie Misalignment Factor |
| $\mathrm{B}_{\mathrm{R}}$ | Braking Factor |

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# CERTIFICATE OF AUTHORSHIP AND ORIGINALITY OF THESIS (DECLARATION)

The work contained in this thesis has not been previously submitted either in whole or in part for a degree at CQUniversity or any other tertiary institution. To the best of my knowledge and belief, the material presented in this thesis is original except where due reference is made in the text.

Signature redacted

(MD Gyas Uddin)

June 2015

# COPYRIGHT STATEMENT

# LIST OF PUBLICATIONS

The following manuscripts of journal articles and conference papers were published from the work of my PhD project which was supervised by Professor Gopi Chattopadhyay initially and then Associate Professor Mohammad Rasul as principal supervisor and Dr. Subhash Sharma as an associate supervisor. These articles were prepared at various stages of the project based on the results/outcomes obtained during those times. The supervisory team met regularly to provide academic advice and feedback to ensure successful completion of the project.

## REFEREED JOURNALS ARTICLES

- Uddin, MG, Chattopadhyay, G, MG, Rasul, MG 2014a, 'Development of effective performance measures for wayside rail curve lubrication in heavy haul lines', *In the Proceedings of the Institution of Mechanical Engineers Part F : Journal of Rail and Rapid Transit*, vol. 228, no. 5, pp. 481-495.
    - I prepared the draft manuscript after collecting and analysing data. Associate Professor Mohammad Rasul and Professor Gopi Chattopadhyay assisted me to develop the concept of the manuscript and write-up and provide technical feedback on the data analysis and the content of the manuscript. Both of them also reviewed the draft and also provided the valuable suggestions throughout the publication and review process.

## REFEREED CONFERENCE PAPERS

- Uddin, MG, Rasul, MG, Chattopadhyay, G, Leinster, M, Sharma, S 2014b, ‘Comparison of life cycle costing and economic benefits of different wayside lubrication technology in heavy haul railway’, *In the Proceedings of 27$^{th}$ International Congress of Condition Monitoring and Diagnostic Engineering (COMADEM 2014)*, 16-18 September, Brisbane, Australia.
    - I developed the concept and prepared the draft manuscript for the publication and conducted data analysis and economic evaluations. Associate Professor Mohammad Rasul and Professor Gopi Chattopadhyay assisted me to write-up the manuscript and provided feedback on the data analysis and the economic evaluation. They also

reviewed the draft and provided the valuable suggestions through the final submission of the publication. Malcolm Leinster and Subhash Sharma helped me through writing stage, data analysis and the technical comments on the publication.

- Uddin, M, Chattopadhyay, G, Rasul, MG & Leinster, M 2012, 'Modelling of grease transport mechanism in wayside rail lubrication and impact of related parameters', *In the Proceedings of Global Perspectives RTSA CORE 2012: Conference on Railway Engineering*, 10-12 September, Brisbane, Australia.
  - I developed the concept of the grease transport mechanism based on the field trial observation and prepared the draft manuscript. Associate Professor Mohammad Rasul and Professor Gopi Chattopadhyay assisted me to review the draft and improve the flow and quality of the content manuscript. Associate Professor Rasul also reviewed the draft and provided the valuable suggestions through the final submission of the publication. Malcolm Leinster has contributed to develop basic mathematical concept of grease transport mechanism.
- Uddin, M, Chattopadhyay, G, Rasul, MG, Sroba, P & Howie, A 2011a, 'Development of economic model for analyzing lubrication effectiveness in heavy haul lines' in M Singh, R Rao & J Liyanage (eds.). *In the Proceedings of the 24th International Congress on Condition Monitoring and Diagnostic Engineering Management (COMADEM 2011)*, 30th May - 1st June, Stavanger, Norway.
  - The entire manuscript of this publication was written and developed by me. Associate Professor Mohammad Rasul and Professor Gopi Chattopadhyay contributed through the write-up, economic model development and revising the draft to improve the quality and flow of the manuscript. Peter Sroba and Alex Howie contributed in data collection and providing industry data to analyse the economy of rail curve lubrication. They also contributed throughout the feedback on the draft content to improve the quality of the manuscript for publication.
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  - The manuscript of this publication was written and developed by me. Technical data and industry latest practice information was supported by Peter Sroba and Alex Howie. Associate Professor Mohammad Rasul and Professor Gopi Chattopadhyay

contributed through the write-up, revision, providing suggestions and recommendation throughout the submission process.

- Uddin, MG, Chattopadhyay, G, Sroba, P, Rasul, MG & Howie, A 2010a, ‘ Wayside Lubricator Placement Model for Heavy Haul Lines in Australia’, *In the Proceedings of the Conference on Railway Engineering (CORE) 2010*, 12- 14 September, Wellington, New Zealand.

  - I developed the placement model based on the experimental data and results. The entire manuscript was written by me. Peter Sroba has guided me to develop the placement model based on the currently available models and our field trial data and findings. Alex Howie and Peter Sroba also helped me in data collection throughout the field trials for an extended period of time. Professor Gopi Chattopadhyay significantly contributed in revision of the draft with his valuable comments and suggestions. Associate Professor Mohammad Rasul contributed through the write-up, revision and final submission of the manuscript.

- Uddin, MG 2010b, 'Wayside lubricator placement model for heavy haul lines' in Institute for Resource Industries and Sustainability 2010, *2nd Annual Conference of the IRIS Postgraduate Students: 'Resourcing for the future'. Conference Program*, 1st December, 2010, Rockhampton, Australia.

  - I was the sole contributor in the development of the manuscript. I wrote-up the manuscript based on the field testing data and outcomes of this project.

- Chattopadhyay, G, Uddin, MG, Desai, A, Sroba, P, Rasul, MG & Howie, A 2010, 'Best practice in rail curve lubrication and remote performance monitoring in heavy haul lines' paper presented to the AusRail 2010, 23 - 24 November, Perth, Australia.

  - Professor Gopi Chattopadhyay wrote the manuscript of this publication. Where I contributed significantly to provide the materials for the manuscript. I provided the relevant data of existing practice around Australia and also the findings and data from field trials of remote performance monitoring. Peter Sroba, Alex Howie and Ajay Desai also contributed in providing information and data from the project.

- Uddin, MG, Chattopadhyay, G 2009, 'Development of Economic Model for Analyzing Lubrication Effectiveness Based on Below Rail and Above Rail Assets', *In the Proceedings of ASOR 2009*, 27- 30 September, Gold Coast, Australia.
    - I developed the overall concept of economic model of rail curve lubrication and analysing the impacts of the assets. A through literature review and idea generation was conducted by me. Professor Gopi Chattopadhyay did his valuable contribution in this manuscript which was written by me. He contributed significantly to generate the concept, write-up the manuscript and through review of the draft.

# Chapter 1

# INTRODUCTION

## 1.1 Background and Significance

The rail industry is a cost intensive industry. Enormous amounts of capital and operational costs are incurred by the rail operators every year. In 2011, the railroad freight industry generated a record U$65 billion (AUD85.8 billion as of today's rate) in revenue. In 2013, the Class I railroad industry generated record revenue of US $72.9 billion (AUD96.22 billion), produced 1.7 trillion revenue ton-miles (2.737 trillion revenue ton-kms) and spent US $593 billion (AUD782.76 billion) on capital and maintenance costs of track and equipment (Lawrence 2015). It is clear how important railway transport is for the economy of a country. US trade with Mexico through railroads has tripled from U$20.4 billion (AUD20.93 billion) in 1999 to U$64.5 billion (AUD85.14 billion) in 2012, and trade with Canada by rail has increased 78% from U$58 billion (AUD76.56 billion) in 1999 to U$103 billion (AUD135.96 billion) in 2012 (Federal Railroad Administration 2013; USDOT/Bureau of Transportation Statistics 2013). The promising future of freight railways shows solid growth despite having problems with weather, congestion and service issues; Berman (2015) reported that, according to the Association of American Railroads (AAR), in 2014 total US carloads increased by 3.9% from 567554 carloads to 15176835, the highest carloads since 2008. Rennicke and Hontoria (2012) reported that the 2007 AAR/Cambridge Systematics National Capacity Study estimated 88% long term growth from 2007 to 2035. If the estimated capacity of 2035 is placed on the current rail network, more than half of the system could be near or over capacity; therefore, to maintain the growing capacity, railroads estimated cost to maintain operability and capacity at right phase for 2007-2035 is U$150 billion (AUD198 billion) and up to 2050 it would be U$200 billion (AUD264 billion) and need to focus on infrastructure and rolling stock.

The huge Australian rail network connects the nation from one corner to another throughout a harsh climate, carrying freight and passengers all over the country, including coal and ore from 'pit-to-port'. According to the Australian Trade Commission (2013), Australia runs the world's heaviest and longest heavy haul trains with 40 tonnes axle load and train lengths of 2.5 km or more. It is one of the most significant driving forces of the Australian economy and commerce. Rail is one of the high capital asset intensive industries. Its main assets have a long life which demands quality maintenance. Proper asset management and well planned maintenance has a large impact on the reliability, availability, maintainability and safety of rail operations (INNOTRACK 2009). To increase productivity and achieve a high level of safety, it is necessary to ensure that existing rail assets are maintained effectively.

Leading experts in Australian heavy haul rail shared their views on the next steps for that area to shape the future of the rail sector, indicating that maximum utilisation of existing assets, implementation of advanced technologies in condition monitoring, communication to assist staff to maintain their infrastructure and remote equipment and an increase in productivity via significant reduction in downtime should be targeted (Informa Insights 2013). The significance of Australian rail transport is enormous for Australian bulk transport from mine and firm to port, freight transport to and from the ports, and interstate and interurban passenger transport around the country. According to the Australasian Railway Association (ARA 2014), key facts of the Australian rail industry can be described as follows.

### 1.1.1 Australian Railways

Australia currently has more than 33000 route-kilometres, 800 locomotives and 32000 wagons and carriages (ARA 2013).

#### *1.1.1.1 Freight Capacity*

Australian railways carried more than 1 billion tonnes of freight in 2012-2013 which shows a 57% increase since 2007-2008 and where bulk movements account for 97% of the overall freight task. Iron ore and coal are the two largest bulk freight commodities; iron ore is mainly produced in Western Australia and coal in Queensland and New South Wales. Iron ore production has tripled in the last decade. Black coal production has increased by 45% since 2012 (ARA 2014; BITRE 2013). On the Pilbara iron ore railways, as of 2012 Rio Tinto operates 2.6 km long trains with a capacity of 26000 tonnes each, BHP Billiton operates trains of 37000 tonnes each and Fortescue Metals 33000 tonnes each, the latter at 40 tonnes axle load. There is no other mode of land transport available which can match the unparalleled capacity of the heavy haul railway to sustain the capacity of the world’s largest iron ore and coal exporting ports. Australian railways also extensively transport intermodal and agricultural freight throughout the states and through export import facilities. Passenger throughput in 2013 of 850.3 million passenger trips was conducted by heavy and light rail operations which are equivalent to 16.4 million passenger trips weekly. About 46% of the Sydney suburban commuters use the rail mode in their travel to and from the city (ARA 2014; BITRE 2014). Figure 1.1 shows the estimated Australian freight volume transported by different transport modes from 2000-2001 to 2011-2012. Figure 1.2 shows the total rail freight (in intermodal and bulk) from 2007-2008 to 2012-2013 though the data for 2010-2011 & 2011-2012 is missing in the source reference. It was not possible to reveal the data for these two years. Rail manufacturing and supporting infrastructure generates annual revenue of more than AUD4.2 billion and Australia should leverage competitive advantage of its capability in those areas.

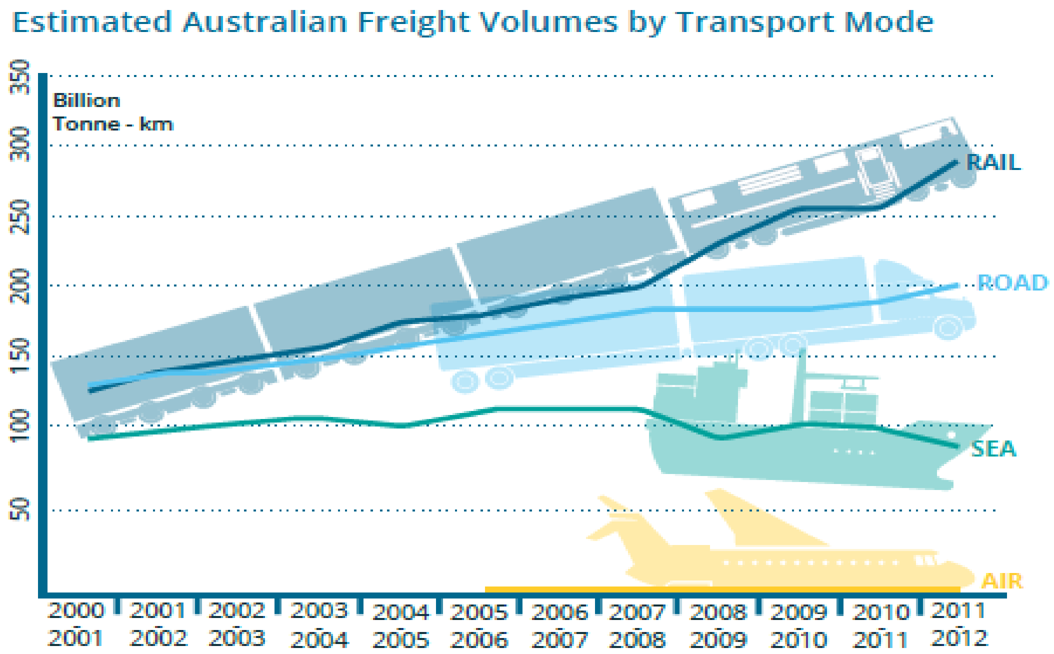



Figure 1.1: Estimated Australian freight volumes by transport modes (Rail, Road and Sea (ARA 2014a)

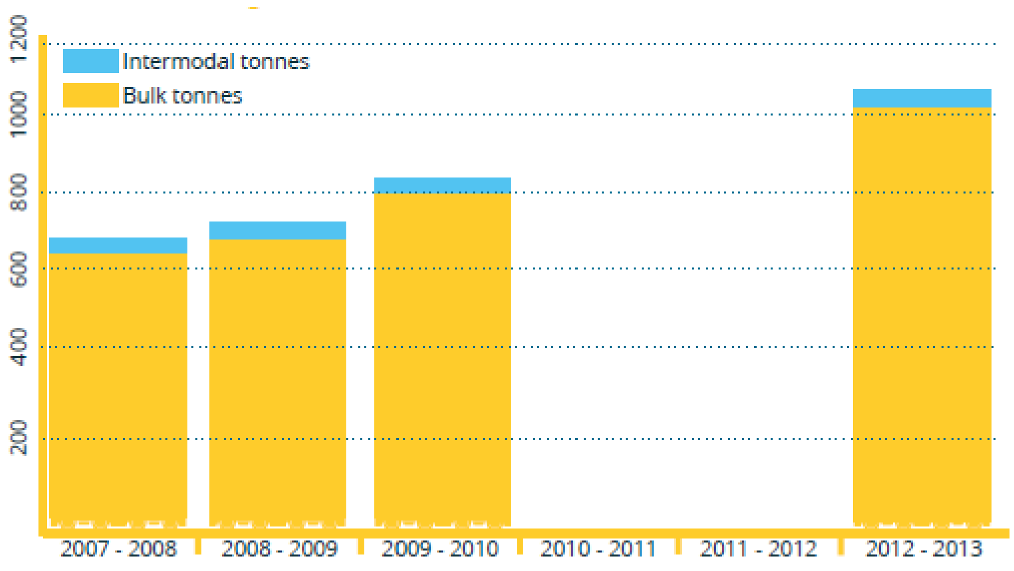



Figure 1.2: National rail freight task from 2007-2008 to 2012-2013 in million tonnes (ARA 2014a)

#### *1.1.1.2 Safety Performance*

Rail is the safest of the land transport modes. Deloitte Access Economics (2011) reported that road transport incurred approximately eight times more in accident costs compared to rail transport.

#### *1.1.1.3 Environmental Performance of Rail*

Carbon dioxide equivalent emissions of the rail industry have increased in the last decade due to the huge resource and passenger boom. However, its impact on the national economy also known as emission intensity is not so severe and the value is low as compared to road transport mode.. According to Natural Resources Defense Council (NRDC) "emission intensity is the ratio of carbon dioxide emissions to a measure of economic output". For a whole economy it is considered as emissions per dollar of gross domestic product (GDP). If emissions intensity does not improve faster than the economy grows the total emissions keep rising. According to (BITRE 2009) $CO_2$ equivalent emission values includes only contribution of direct greenhouse gases ($CO_2$, $CH_4$, AND $N_2O$). Bulk rail has the lowest emission intensity amongst the entire transport modes as shown in Figure 1.3. The unit used in the y-axis is gigagrams.

### *1.1.1.4 Federal Government Investment in Rail*

Compared to 2013-14, the current 2014-15 budget showed a 43% reduction in federal funding for rail projects, though the year 2015-16 will be unchanged. It also forecasted that there will be two significant reductions in rail funding in the year of 2016-17 and 2017-18, which is in complete contrast to the significance of and positive gains from the rail industry (ARA 2014).

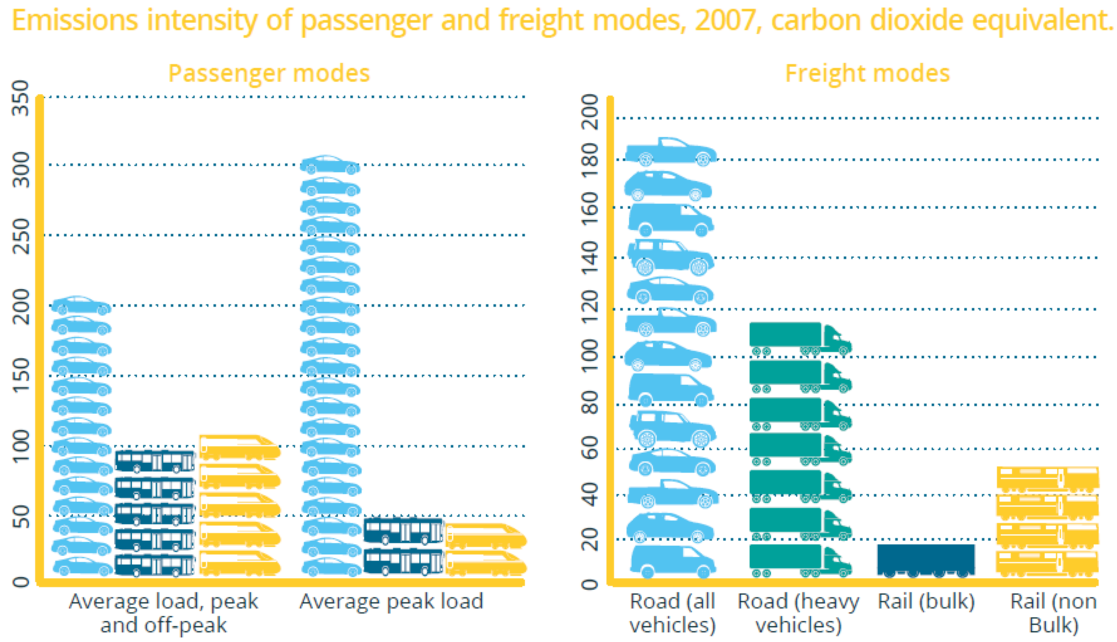



Figure 1.3: Emission intensity of passenger and freight modes, 2007, carbon dioxide equivalent (ARA 2014a), Y-axis in gigagram

According to the True Value of Rail Report (Deloitte Access Economics 2010) produced for the ARA, the significance of Australia's rail industry compared to its road transport industry was found to be as follow:

- The average freight train takes 110 trucks off the road which means a reduction of truck movements by 49.7 million truck kilometres every year (equivalent to travelling 3100 times from Sydney to New York).
- Australian traffic congestion costs around $15 billion per year, and an average passenger train can take 525 cars off the road (equivalent to 3.2 million vehicle kilometres).

- In Australia, an average of 1500 road fatalities and more than 30000 road injuries cost $35 billion annually compared to 37 fatalities and 130 injuries in rail transport, which shows that rail is the safest land transport.
- The Bureau of Transport and Regional Economics estimated in 2009 that 48.3 million tonnes of carbon was emitted from road transport compared to 40% less for each kilometre per person travelling by rail.
- The True Value of Rail Study (Deloitte Access Economics 2010) showed that replacing trucks equivalent to one freight train between Melbourne and Brisbane reduces carbon emissions by the same amount as a household of 3 people going without electricity for 46 years.

It is to be noted that the rails are the main load bearing element of the superstructure of the track construction which is continuously under stress due to the loads applied through the wheel/rail interface. The rails withstand dynamic loads in vertical, longitudinal and lateral directions. For uninterrupted operation, safety and positive economic performance, rails and wheels must have a reasonable life. Rail operators around the globe have taken initiatives to achieve an optimum service life of rails and wheels by reducing wear. Amongst the various efforts in rail and wheel life improvement, rail lubrication is one of the significant ones. Researchers are investigating ways of improving rail lubrication methods to achieve energy conservation by reducing the contact friction, and material conservation by reducing the wear rate of rails and wheels. There are a variety of lubrication methods that are used for wheel/rail lubrication ranging from manual spot lubrication by workers, on-board lubrication, hi-rail lubrication and the latest wayside lubrication technology. In this study, a wayside lubricator placement model has been developed for use in Australian heavy haul lines and has evaluated the sustainability of the lubricator placement in tangent track and in the transition of circular curves in order to reduce the wheel/rail wear, energy consumption, wheel/wheel maintenance cost and noise. More details on rail lubrication are given in the next section.

### 1.2 Rail Lubrication

Lubrication of the wheel and rail interface has an enormous effect on friction and the wear of rails and wheels. It increases the life of the rails and wheels, and reduces both energy consumption and noise such as wheel squealing and flanging through the curve. Curve squeal is one of the most significant environmental concerns from railway operations for residents living close to rail lines and may affect the rail operator's social license to operate (Hanson et al. 2014). The Association of American Railroads (AAR) estimated that the wear and friction occurring at the wheel/rail interface of trains due to ineffective lubrication costs American railways in excess of US$2 billion each year (Sid & Wolf 2002). Popovic et al. (2014) emphasised the need for effective rail management, including rail lubrication, to optimise track maintenance costs as only rail incurs more than 30% of the construction cost of new railway infrastructure for ballasted track. Petry (2012) reported that major US railroads are investing US$13 billion to improve freight rail networks due to a significant increase in demand to move freight by rail. There are three main methods of lubrication in use by railways around the world. These are wayside, on-board and hi-rail lubrication. The wayside lubrication method is widely used for its cost effectiveness and ease of operation. Grease is the most commonly used lubricant that is applied to rails worldwide using wayside lubricators and that is specifically studied in this thesis.

### 1.3 Problem Statement and Research Question

The growing needs of industry and commerce lead the railway operators towards the following options individually or in combination:

- Introduce more trains.
- Increase the number of wagons per train.
- Increase the load per wagon (i.e. Heavy Axle Loads).

Increase of the axle load in heavy haul lines increases challenges of maintenance, problems due to track deterioration, wear, change of track geometry and derailments resulting in loss of assets, lives and revenue due to disruption of service. Wear in general and fatigue are major problems in railway infrastructure and result from friction between the wheel and rail profile. Gauge side wear on the high rail in curves is a common problem (Turner 2008). Figure 1.4(a) shows heavily worn rail in a curve which can affect the life and performance of both below rail and above rail (wheels/bogies/vehicles) assets. Major influential wear factors include axle loads, lateral forces, longitudinal forces, creepage, curve radius, gradient of the track, track cant/superelevation on curves, track gauge, surface condition of the wheel and rail, speed, length, frequency and type of trains, rolling stock performance, operational and environmental issues.


(a) (b)

Figure 1.4: (a) Heavily worn rail in a curve with visible RCF (Rolling Contact Fatigue), (b) Typical gauge face area on an effectively lubricated curve where the grease should sustain rail gauge face wear reduction

Lack of effective lubrication has been recognised as a problem throughout the Australian heavy haul networks and concerned about the current practice and potential opportunity to improve the performance and economy of the practices. This research has been conducted to study and analyse the current practice and investigate the potentials of improvement by implementing the latest technology in wayside lubricators and different types of grease. The main research question that has been addressed in this thesis is "What is the rail curve lubrication best practice for Australian heavy haul rail lines?"

There are many significant factors that influence the effectiveness of wayside lubrication, and these can be categorised as lubricant properties, lubricator setup and lubrication transport mechanism, traffic patterns and vehicle types, rail track geometry, traffic patterns and vehicle types, operating environment and operating parameters. Under these categories, the specific factors that need consideration include: lubricant properties (consistency, viscosity, pumpability, retentivity, water washout), lubricator setup and lubrication transport mechanism (use of short or long applicator bars, use of single pair or double pair bars, lubricant type and application rate), rail track geometry (percentage of tangent track, curve radii and length, track alignment factor, gradients), and operating environment and operating parameters (sanding, wheel/rail temperature, surface roughness of rail and wheel, solid particle contamination, climate and geographical territory).

Amongst all of the available lubricator technologies, the electric lubricator has been found to provide higher accuracy of application rate and less wastage of grease, but grease application is still based on wheel or axle counts. Better friction management and the appropriate amount of grease application can be achieved through continuous improvement in lubricator design and development.

Factors such as grease splash and wastage of grease, excessive numbers of lubricators, lubricator breakdowns, and less than desirable grease carry distances cause real concerns for wayside lubricator placement. Applicator bars installed on the transition of curves need to be removed and refitted for every rail grinding cycle. Thus there is a need for a decision model to determine the optimal placement of lubricators. Figure 1.5 shows two wayside lubricator sites on an Australian heavy haul network installed in the transition of the curves. Both sites have installed two short applicator bars just before the curve. Image (a) shows the site where a grinding cycle was carried out without removal of the lubricator bars and left the rail segment without grinding and, as a result, the poor rail profile has resulted in severe RCF (Rolling Contact Fatigue). It also shows the severe wastage of grease and contamination of the ballast and ground around the lubricator site. Image (b) shows another site with severe blockage in the applicator ports with delivery of the grease being far below the gauge face; this has resulted in failure of grease pick up by wheels, leaving the whole rail unlubricated and shiny dry. These are common scenarios in the heavy haul network and need to be managed appropriately.

(a) (b)

Figure 1.5: Ineffective wayside lubrication on Australian heavy haul rail network, (a) RCF on the gauge side at wayside lubricator site, (b) Grease splash and contamination on the top of the rail

The factors that have been discussed above can be elaborated upon further by considering lubricator performance and wheel/rail profile effect. While considering the placement of lubricators, it is necessary to investigate wheel/rail profile influence on lubricant carry distance and overall effectiveness of lubrication. This demands a field study on the effect of applicator bars (short and long bars), locations of the bars with respect to curves, and types of lubricants. Before the implementation of a lubricator placement model, economic justification needs to be done based on cost-benefit analysis. Analysis of cost-benefit of the model and itemised cost evaluation of the wayside application methods is also needed for the benefit of rail operators. It has been revealed that current practices are not adequate and effective (CRC Australia 2014, Uddin et al. 2010a, 2014a, 2014b) , hence more research is required to achieve an acceptable standard of lubrication practices to achieve high rail asset life, low energy consumption and more comfort in terms of noise. This research has conducted actual field tests, optimised lubricator placement, evaluated a number of greases for their performance, and developed a model for improving economic benefits and grease carry distance.

## 1.4 Aims and Objectives

This is a comprehensive real life rail curve lubrication research project for Australian heavy haul railways. The aims and objectives of this research have been developed based on the drawbacks in current lubrication practices and the need to establish best lubrication practices in the Australian Heavy Haul Rail Industry. The main aims of this study are to:

- Survey the current practices of curve lubrication in Australian heavy haul railways and assess their effectiveness in reducing wheel/rail wear, energy consumption, wheel/rail maintenance cost and noise.
- Develop a wayside lubrication placement model based on the test results and evaluate the lubrication effectiveness, optimal location and positioning of lubricators and optimal dispensing rate of grease.
- Evaluate the sustainability of the lubricator placement in tangent track and curve transition spirals.
- Evaluate the effectiveness of long grease applicator bars in tangent track and short grease applicator bars in the curve transition spirals.
- Evaluate the basic economics of wayside lubrication based on lubricator performance on tangent track and in the curve transition spirals. Cost modelling for wayside lubrication methods and economic data analysis for different

configuration of field trial and current practice has been conducted based on specific predefined line capacity in Million Gross Tonnes (MGT). As heavy haul operators are operating their operations in highly competitive economic environment, real life operations data are highly confidential. Therefore it was very unlikely to adapt real data and conduct economic data analysis. The economic data analysis approach in this thesis can be implemented in any real life data analysis and get the expected cost for implementation of different configurations of equipment and grease.

- Observe grease transport mechanism between the rail, wheel and the applicator bars.

To achieve the main aims, the specific objectives/tasks carried out were to:

- Determine best practices in wayside gauge face lubrication for Australian heavy haul lines by evaluation of different lubricator systems, its placement and the optimal lubricant application rates and its quantity.
- Compare the effectiveness of short lubricator bar technology used in the transition spirals of curves, and long bar technology used in tangent track.
- Develop strategies for the optimal placement location of lubricators on heavy hauls lines.
- Explore the feasibility and the benefits of remote condition monitoring technology of the lubricator units to achieve uninterrupted grease supply by the applicators.
- A basic economic evaluation on both the trial short bar technology and long bar technology systems.
- Observe grease exchange behaviour in between applicator bars, rail and wheel and understand the impact of different contributors in grease transport mechanism in rail curve lubrication.

### 1.5 Scope and Limitations of the Study

The defined scope of this research was broad and vigorous, with comprehensive investigations and field trials which have been covered through the following activities:

- A thorough review on heavy haul lubrication practice around the globe, lubricator placement practice, railway terminology, wear and wear mechanisms, grease application and lubrication mechanisms in heavy haul railways, and economic benefits from lubrication in heavy haul railways.

- Conduct field trials in the Blackwater heavy haul railway system, Queensland, Australia, to develop a process for the ranking of lubricators and applicator bars based on their effectiveness.
- Investigate the suitability of different rail curve greases.
- Develop an economic model for rail curve lubrication practice in heavy haul railways and conduct a cost benefit analysis on various configurations of wayside lubrication practice.

The following activities are outside the scope of this research:

- ➢ Conduct any laboratory testing to measure the properties of any grease or equipment performance.
- ➢ Rank grease based on laboratory testing outcomes.
- ➢ Characterise grease based on chemical compositions.
- ➢ Improvement of lubricators which did not perform well.
- ➢ Improvement of applicator bars which did not perform well.
- ➢ Improvement of grease which did not perform well.
- ➢ Any modification of the lubrication equipment, lubrication measuring equipment and greases.

### 1.6 Structure of the Thesis

This thesis has been presented in an appropriately structured way which demonstrates the sequential flow of information on the research topic, methodological study, data analysis, and findings and recommendations on wayside lubrication in Australian heavy haul railway operations. The structure of the thesis is as follows:

**Chapter 1** provides a brief background of the study, problem statement, aims and objectives of the research, scopes and limitations of the research.

**Chapter 2** provides an in-depth literature review on railway track, relevant rail and wheel terminology, loads, wear and wear models, wheel/rail wear, lubrication and lubrication regimes, wheel/rail lubrication, wayside lubrication practice and equipment and influencing parameters, the economic benefits of rail lubrication and wayside lubrication placement modelling and other allied areas.

**Chapter 3** provides the details of design of experiment and methodology of this research. Methodology discussed covers the development of the test plan, determination of objectives of the test, test locations, selection of equipment and grease, and development of the detailed test procedures. It also provides the details of management of data collected from the field trials.

**Chapter 4** provides details of data analysis and results of all field trials covering the entire test configuration based on different equipment suppliers, types of applicator bars and types of greases used. Technical and economic evaluation of performance/effectiveness of different applicator bars and suitability of greases has been analysed in this chapter as a path forward for lubrication decision making.

**Chapter 5** provides information on current practice of wayside lubricator placement and detailed discussion on the proposed placement model development based on equipment, applicator bars, effectiveness of grease and other considered factors. The chapter also presents results of field trials that have been implemented in the modelling to achieve a realistic wayside lubrication placement model which will be highly implementable on the complex heavy haul networks. The Lubrication Effectiveness Index (LEI) for a rail network is derived.

**Chapter 6** provides a basic evaluation of the economics of different wayside lubrication practices. It emphasises the various needs for economic analysis of lubrication. Detailed cost elements in wayside lubrication and the life cycle cost modelling of wayside lubrication for heavy haul railways are discussed. Evaluation of various configurations of wayside equipment and grease has been conducted based on the predefined line capacity due to the lack of real life economic data as it is highly confidential for rail operators in competitive economic environment. This analysis is applicable in to real life operation and an effective estimation can be achieved for long term decision making. A rudimentary economic model has been developed based on the field investigation. That is a step forward in achieving excellent lubrication practices.

**Chapter 7** provides the details on how various parameters affect the grease transport mechanism. Emphasis has been given to the grease chemistry and necessary grease performance enhancer additive packages. Details of how various greases perform on carry distance are given.

**Chapter 8** provides the summary, conclusions and recommendations from this research and sets out future research opportunities.

# Chapter 2

# LITERATURE REVIEW

## 2.1 Introduction

An extensive literature survey into the significance of the railway industry (and specifically heavy haul railways) around the globe, general lubrication processes, wheel/rail lubrication, wheel/rail wear and the economic impacts was conducted during this research. The technical and economic significance of effective wheel/rail lubrication is a long standing issue for rail operators. Within different wheel/rail lubrication practices, focus has been given to the wayside rail curve lubrication, various technologies of rail curve lubrication, effective friction management for rail curve lubrication and wear, lubricator placement modelling, grease transport mechanisms, and cost-benefit analysis of rail lubrication in this research. An in-depth literature review found that, though a number of researches have been conducted in this field, significant research still needs to be done to resolve unanswered questions. This work has been also reported in detail by the author in the CRC for Rail Innovation Project Report (CRC Australia 2014).

## 2.2 Overview of Lubrication

Lubrication plays an important role to reduce friction between sliding and rolling parts. Proper lubrication can considerably reduce friction and wear by interposing lubricant between mating parts. The physical and chemical interactions between the lubricant and lubricating surfaces must be understood to generate improved life of machine elements and optimise their performance (Hamrock 1991). Depending upon the application and requirement, appropriate lubrication regimes need to be chosen and implemented in the interface of mating parts. Maru and Tanaka (2007) reported that lubrication regimes have a distinct relationship with the surface interaction mechanism and relevant friction and wear. In many cases, the friction and lubrication relationship is characterised by $\eta V/W$ (oil viscosity x sliding velocity/normal load) factor (Maru & Tanaka 2007; Wakuri et al. 1988; Wakuri et al. 1995; Bayer 1994) and represented in a curve called a Stribeck curve. For high values of $\eta V/W$, the friction coefficient is linearly ascending due to fluid film lubrication (dragging in oil film). Significant reduction in the ηV/W factor may cause metal-to-metal contact as the film thickness may be smaller than the height of the surface asperities and would result in boundary lubrication (Maru & Tanaka (2007), Ludema (1996)).

In consideration of two rough surfaces, several authors consider the Lambda value to characterise lubrication in rubbing or sliding contacts. A Dimensionless Film parameter, Λ is used to define the four important lubrication regimes. It is determined from the relationship between minimum film thickness and the rms surface finish of two mating parts. According to Hamrock (1991):

$$\Lambda = \frac{h_{min}}{(R_{q,a}^2 + R_{q,b}^2)^{1/2}} \tag{2.1}$$

where, $R_{q,a}$ = rms surface finish of surface a and $R_{q,b}$ = rms surface finish of surface $b$

Figure 2.1 shows lubrication regimes and the wear coefficient in sliding of metal surfaces as a function of the Lambda ratio.

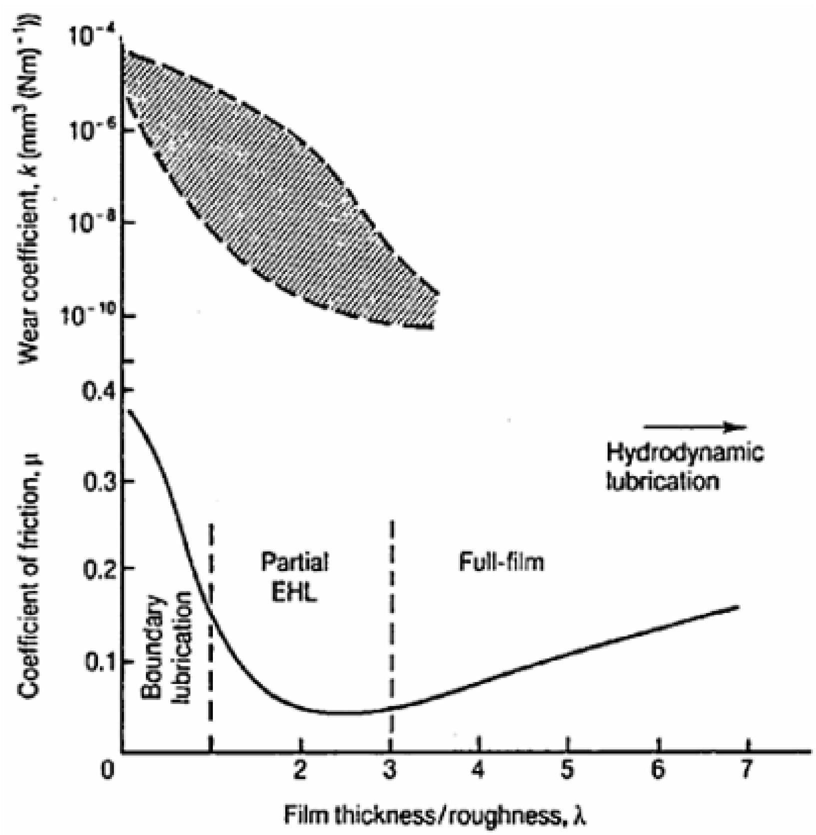



Figure 2.1: Lubrication regimes and wear coefficient in sliding of metals, as a function of Lambda ratio, λ (Maru & Tanaka 2007; Hutchings 1992, CRC Australia 2014)

There are three fundamental regimes of lubrication, namely fluid film lubrication, boundary lubrication and mixed lubrication. These are described below.

### 2.2.1 Boundary Lubrication

In boundary lubrication, the solids are not separated by the lubricant and fluid film effects are negligible. Thus there is considerable asperity contact (Hamrock 1991). The frictional characteristics of the contact are determined by the properties of the solids and the lubricant at the interface. Hargraves (n.d.) reported that the physical and chemical properties of the thin surface films govern the lubrication mechanism, the properties of the bulk lubricant are of minor importance and the coefficient of friction is independent of the viscosity of the lubricant. For this lubrication regime, the Lambda ratio is less than or equal to 1 and the coefficient of friction varies from 0.15 to 0.40 (Figure 2.1).

### 2.2.2 Partial or Mixed Lubrication

The behaviour of the interface in a partial lubrication regime is governed by a combination of boundary and fluid film effects. Some contact will take place between the asperities. Transition from elastohydrodynamic (EHL) to partial lubrication does not take place instantaneously as the severity of loading increases, but rather a decreasing proportion of the load is carried by pressures within the fluid. As the load increases, a larger part of the load is supported by the contact pressure between the asperities of the solids. The average film thickness in mixed lubrication is less than 1μm and greater than 0.01μm. The Lambda ratio is in a range of 1 to 3 and the coefficient of friction varies from 0.05 to 0.18 as shown in Figure 2.1.

### 2.2.3 Hydrodynamic or Fluid Film Lubrication

If the interposed lubricant between the mating surfaces is sufficiently thick to prevent the opposing solids from coming into contact, then the condition is described as fluid film lubrication. Hydrodynamic and elastohydrodynamic lubrication are the forms of fluid film lubrication. They are generally characterised by conformal and non-conformal surfaces. Here the lubrication films are generally thick so that opposing solid surfaces are prevented from coming into contact. This condition is often referred to as the ideal form of lubrication since it provides low friction and high resistance to wear. The lubrication of the solid surfaces is governed by the bulk physical properties of the lubricant. Coefficient of friction for fluid film lubrication is typically 0.001 (Hargreaves, n.d.). In hydrodynamic lubrication, the film thickness should be 4-6 times higher than composite roughness. The Lambda ratio is > 3 (Figure 2.1)

### 2.2.4 Lubrication Regimes within the Wheel/Rail Interface

In the rail curve lubrication application, lubrication regimes within the wheel/rail interface may vary all along the profile due to changes in their geometry and load distribution as a rail vehicle axle moves along the track. As far as variation of surface micro-geometry and surface characteristics is concerned, there is a huge variation in the wheel/rail contact all along the profile. The load speed conditions indicate that boundary lubrication prevails most of the time between the rail and wheel but mixed lubrication or Elastohydrodynamic lubrication could also exist in the contact depending upon the operating conditions and the geometry of the mating surfaces. Research needs to be carried on to identify lubrication regimes occurring at different locations all across the rail and wheel contact interface patch.

## 2.3 Lubricants

Lubricant is a material in fluid, semifluid or solid form that is applied between two mating surfaces to minimise direct metal to metal contact. Lubricants have lower shear strength and hence they engender low frictional forces. This lower coefficient of friction helps reduce both energy and material conservation. There are three main types of lubricants, i.e. solid, liquids (oils) and semifluids (grease) and, also gas is considered as fourth type of lubricant. Examples of solid lubricants are graphite and $MoS_{2,}$ while liquid lubricants are mainly petroleum based mineral oils and synthetic oils (mineral oils mixed with polymeric materials) (Hamrock 1991). Greases are semi-fluids that can be retained in the contact interface for longer periods of time. These are soaps of oils mixed with thickening agents. Greases are widely used in the rail industry because they can remain on the contact surface without using a continuous lubricant feeding mechanism. Dental drills and some pharmaceutical industry applications need air bearings due to precision requirements at high speed and low load operating conditions, prevention from contamination of products with fluids is also one of the reasons for using gas as a lubricant. The American Society for Testing Materials standards (ASTM 1961) defines grease as a solid to semi-solid product of a thickening agent in a liquid lubricant with other ingredients called additives to provide special properties. According to Hamrock (1991), grease is petroleum oil to which metallic soap thickeners are added. Oil is the largest component of grease which could be 65 to 95% and frictional characteristics of grease are based on this component. Thickeners make up 5 to 17% of simple grease and are responsible for retaining base oil and resist heat, water and extreme load.

Commonly used thickeners are hydroxides of lithium, calcium, molybdenum, sodium, barium and aluminium. Lithium soaps or thickeners are the most commonly used multipurpose greases having melting points of about 195° C. They are highly resistant to water and oxidation with strong mechanical working capacity. These are especially suitable for rolling and sliding contact interfaces. Keeping in view the self-retention property, the exposure to an open atmosphere and the relative ease of application, they are most suitable for rail curve lubrication. Calcium soap based greases have excellent water resistance and could be operated up to 120° C. However, oil separation takes place at higher temperatures. These are also used in rail curve applications. Most of the rail curve greases have calcium or lithium thickeners. Greases also contain additives which cater for specific functions, e.g. graphite and molybdenum are the most common additives known as friction modifiers that reduce the coefficient of friction in boundary lubrication regimes. There are many other types of thickener based greases like aluminium complex, microgel silica and others. The rail lubrication application of grease is different from other industrial applications. Common properties of grease are:

- Appearance
- Texture
- National Lubrication Grease Institute (NLGI) grade
- Thickener
- Base oil viscosity
- Drop Point
- Worked penetration
- Four-Ball Weld Load

The literature review revealed that established selection criteria for effective rail curve grease are rare in the heavy haul rail industry.

## 2.4 Rail Lubrication

Rail lubrication is required in two situations: firstly to reduce the friction and wear between the wheel and top of rail (TOR) as the level of friction required in such contacts is neither too low nor too high (0.3-0.4), and secondly in rail curves where the wheel flange and rail gauge face are in contact where the friction must be as low as possible. The literature review shows that efforts have been made to evaluate the effect of rail lubrication on friction and wear by using rigs, pin-on-disc tests and tests on operating railway tracks (Descartes et al. 2011; Beagley, McEwen & Pritchard 1975; Sundh, Olofsson & Sundvall 2008; Alp, Erdemir & Kumar 1996; Clayton, Danks & Steele 1989; Ishida et al. 2008). Studies have been conducted to understand the behaviour of the lubricant and contaminant mixture formation as a third body lubrication layer at the wheel/rail interface (Descartes et al. 2011; Hou, Kalousek & Magel 1997; Lu, Cotter & Eadie 2005). Descartes et al. (2011) also reported that the mechanism of lubrication on the rail gauge corner involves complex coupling phenomena which do not allow elementary parametric studies on site because control of the lubrication process requires the control of the initial formation of an efficient mixture and then sustaining it in the contact area; the authors recommended further study on the contact area morphology, chemical analysis and mixture thickness.

Lubricant is traditionally applied in the wheel/rail interface to manage friction both at the rail gauge face and the wheel flange. Proper management of friction in curved track can reduce wear, energy consumption and noise generation. An optimum level of friction needs to be available on the locomotive wheel treads to maintain not only the required traction to haul the train, but also to avoid bogie hunting of wagons in the tangent tracks and large lateral forces at the leading wheelset of vehicles. Setting for optimum level of friction would be different for different rail networks depends on the capacity to maintain the level of friction by implementing gauge face lubrication and top-of-rail friction management. Well accepted practice of optimum friction requirements has been discussed more in Chapter 2 (Friction Management Guidelines). According to Chen et al. (2014), Hooper et al. (2003), Eadie, Santoro and Kalousek (2005), Eadie et al. (2006), Kusuda, Yamaguchi and Fukagai (2009), Ishida, Ban and Fukagai (2009), and Chestney, Dakah and Eadie (2009), friction control at the wheel/rail interface by lubricant application is conducted by many countries. Very recently, friction control of the wheel/rail interface of the low rail by lubricant has also been initiated, though these types of lubricants are very different in their properties and purposes. Chen et al. (2014) also reported that lubricant application to the low rail on curves has a significant effect on reduction of lateral forces which cause wheel flange and rail gauge face wear of the high rail and corrugation on the low rail.

According to Lemma et al. (2014) and IHHA (2009), friction is not an easily quantifiable parameter, but the heavy haul operators have to achieve it as to maximise their benefits. Different methods of rail lubrication have been developed due to suitability of application, management and performance. Three methods of lubrication currently used are shown in Figure-2.2.

(a) Wayside (b) On board (c) Hi-rail lubrication

Figure 2.2: Wheel/Rail Lubrication Systems (Reddy et al. 2006)

In the wayside lubrication method, grease is applied to the track from lubricator units installed beside the track through the applicator bars attached to the gauge side of the rails. Reiff (2006) reported that, when curves are concentrated in specific locations, wayside applicators are useful. On-board lubrication is a method where the lubricator is mounted on the locomotive and the lubricant is applied to the locomotive wheel flange. When curves are uniformly distributed along the track, locomotive mounted applications are more useful. Hi-rail lubrication means the controlled application of a bead of grease directly to the wear face of the rail from a vehicle travelling on the track. The hi-rail vehicle is usually an adapted delivery vehicle equipped with a special storage and application system (de Koker 2004). One or a combination of the above systems is used by rail operators to achieve 100% effective lubrication and significant savings in fuel and wheel/rail maintenance.

## 2.5 Wayside Lubrication

The wayside lubrication method is used for both gauge face application and top of rail application. Rail lubrication is one of the most cost effective ways of rail profile management throughout the rail life. Compared to a single rail renewal using standard Grade 260 rail or premium HP rail or using a rail grinding train, gauge face lubrication is highly cost effective (Evans 2013). Table 2.1 shows the different rail profile management technologies used by the UK's Network Rail and their cost.

Table 2-1: Comparison of rail profile management cost based on various technologies used in Network Rail (Evans 2013)

| **Technology** | Grade 260 single rail renewal | Premium HP single rail renewal | Train based rail grinding | Gauge face lubrication | Top of rail friction modifier |
|---|---|---|---|---|---|
| **Cost** | £90 - 120 per metre | £100 - 130 per metre | £0.6 - 2.9 per metre per cycle | £2.2 - 3.1 per metre treated investment cost<br>£0.3 - 1.3 per metre treated per year operating costs | £15 - 25 per metre treated investment cost<br>£4 - 6 per metre treated per year operating costs |

### 2.5.1 Wayside Rail Lubricators

The complete set of equipment is called a rail lubricator. The whole unit consists of the reservoir tank, grease pump unit, controller, connecting hoses, power supply unit, applicator bars, wheel/axle sensor unit/plunger, and sometimes telemetry or condition monitoring system. Lubricator technology has changed over the years as a result of continuous improvement. There are three types of lubricators available in the market.

#### *2.5.1.1 Hydraulic Lubricators*

Australian rail networks predominantly use hydraulic lubricators. Figure 2.3 shows the hydraulic lubricator as applied in a wayside application.

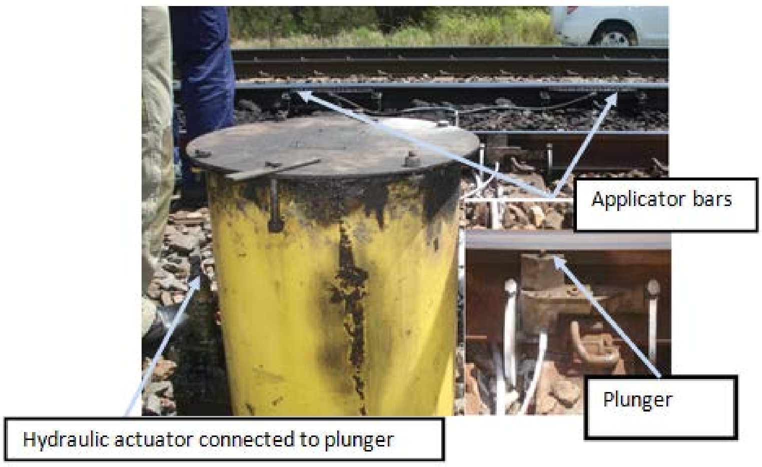



Figure 2.3: Hydraulic lubricator in ARTC Track Network (inset- hydraulic plunger) (CRC Australia 2014)

The main features of hydraulic lubricators are the grease reservoir, the hydraulic plunger/actuator assembly clamped to the field side of the rail with a single hydraulic line connected to the grease pump externally mounted on the grease reservoir and grease distribution units (applicator bars) and associated hose system. Hydraulic lubricators are very simple in construction. The grease pump is activated by the hydraulic actuator and delivers grease to the applicator bars when the wheels strike the plunger. No power supply is needed from an external source like electricity or solar power to operate the actuator. It is easy to install and needs minimal maintenance. It delivers grease with every passing wheel, providing only limited control of the grease application rate, explaining why a huge amount of grease wastage is found around the applicator bars. Due to the excessive amount of grease application, severe TOR contamination is often reported. Special precautions are needed when rail grinding is scheduled, applicator bars and plunger having to be removed to prevent damage during the grinding process.

#### *2.5.1.2 Mechanical Lubricators*

These are a simple effective mechanical device with a low maintenance requirement and high performance. Figure 2.4 shows the mechanical lubricator.

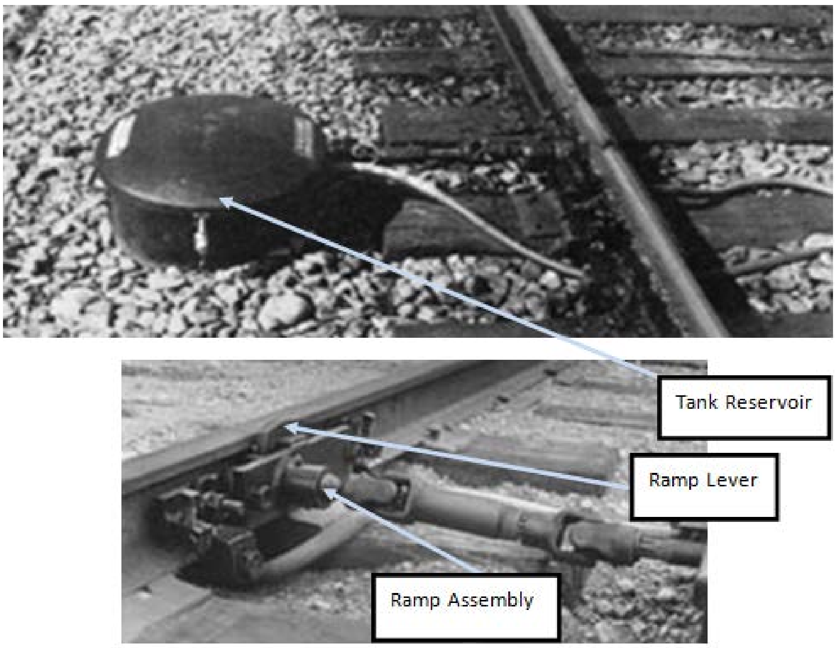



Figure 2.4: Mechanical lubricator (www.portecrail.com)

Mechanical lubricators consist of a grease tank, grease delivery pump and grease distribution unit. The ramp lever, which is connected to the pump through the drive shaft, rotates with the passage of each wheel. The drive shaft then automatically pumps the lubricant to the applicators. The entire pumping mechanism is housed in the reservoir and can be removed for servicing. The grease tank can be of various capacities and applicator bars also of different sizes. This device does not need any external power source.

It is easy to install and a low maintenance device. There is no precise control of the grease delivery rate. Grease is delivered with each wheel pass and causes excess grease delivery which often causes TOR contamination. Excessive lubricant can exacerbate another problem when it enters into the crevices of rolling contact fatigue (RCF) gauge corner or top running surface “checking” defects which are small cracks the lubricant acts like a wedge splinter and widens the cracks.  These are simple mechanical systems and hence intelligent or remote sensing facilities cannot be used. Evans (2013) reported that, to achieve effective gauge face lubrication, older mechanical and hydraulic technology should be replaced by reliable electric units.

#### *2.5.1.3 Electric Lubricators*

Electric lubricators are the latest generation lubricators that have precise electronic control of lubricant application based on axle or wheel count via the wheel sensor mounted beside the rail. It consists of the grease reservoir, electronic controller unit, delivery pump, and distribution unit. These are high pressure, positive displacement and positive distribution systems which are designed to dispense grease on the gauge face or top of the rail. Lubricators can be used for gauge face and top of rail application. They are available in different specifications of power supply, reservoir size, applicator units and telemetry. Figures 2.5 and 2.6 show electric lubricators with different applicator bars.

Figure 2.5: Lincoln electric lubricator used in the spiral of a curve on QR heavy haul line

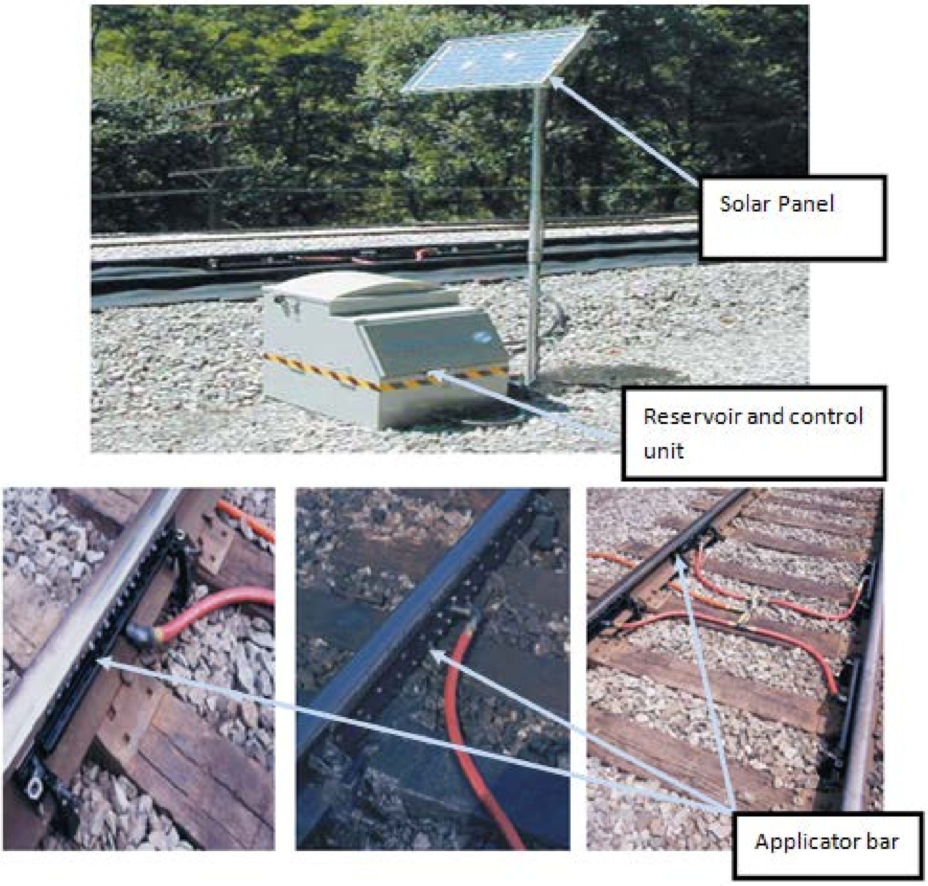



Figure 2.6: Electric lubricators and applicator bars (www.portecrail.com )

The most significant features of the electric lubricator are:

- Highly reliable and efficient operation.
- Apply grease based on the axle/wheel count.

- Precise control of grease application rate and reduced lubricant cost.
- Survive longer in harsh weather and less total cost of ownership.
- Flexibility in grease application due to change of conditions.
- Continuous performance in all weather and seasons.
- Less maintenance cost and time.
- Intelligent condition monitoring unit can transfer data to remote authority.
- Continuous power generation from solar energy or power grid and rechargeable battery for emergency back-up.
- Most of them have higher capacity tank.
- A massive tank as grease reservoir.
- Wheel sensor on track.
- Heavy duty reliable delivery hose from pump to applicator bars.
- Robust applicator bars.
- Solar/Electric power supply.

Electric lubricators also have some limitations. If the control unit breaks down, there will be no grease deliveries to the gauge face. Solar powered electric lubricator units have to be installed in a location where sunlight is available. Installation and removal are more complicated. All the lubricators have clogging problems, and there is also a cavitation problem in lubrication delivery which demands an improved design. Continuous improvement of lubricators is always needed for better performance and effectiveness.

### 2.5.2 Lubricant Applicator Bars

Lubricant applicator bars are mounted on the gauge face of the rail to deliver lubricant to the gauge corner. There are two types of applicator bars, which are popularly known as short bars and long bars. Short bars are suitable to be placed in the transition spiral of a curve whereas long bars are suitable to be placed in the tangent track before a curve. Long bars have some advantages over short bars, namely that they do not need to be removed during grinding cycles, they deliver grease to a greater length of the track, and passing wheels also have more chance to pick up grease. Short bars have to be removed before every grinding cycle and need to be placed back again. During this period the track remains unlubricated and this can cause severe wear. Short bars apply grease to a smaller length on the track and are not effective for short trains with smaller numbers of wagons. This is mainly because of a significant amount of grease wastage in the short bar site in the transition of curve and b in some cases lockage reduce the number of effective grease bead potentially. Due to the above and various other causes the ultimate effective amount of grease picked up by few wheels on small train would be very low and may not sustain the grease exchange in between rail and wheels for a long distance.

### 2.5.3 Lubrication Transport Mechanism

The wheel flange and its contact with the rail are used as the lubricant transport mechanism. Success of any lubrication strategy depends on the transport mechanism. According to Thelen and Lovette (1996), lubrication can be successful only if the transport mechanism is handled in an effective manner. Hanson et al. (2014) reported that the rail profile is one of the important aspects of wheel/rail interaction. Properly managed wheel and rail profiles spread the contact over a larger area and contribute to reduced contact stresses, whereby grease can effectively lubricate the gauge corner contact area. On the other hand, poorly managed wheel/rail profiles lead to severe two point contact in the curves, which leads to excessive contact stresses which reduces the effectiveness of the grease and increases wear and noise. Different wheel/rail contact patterns or poor grease quality may cause discontinuity of grease distribution throughout the curves. Evans (2013) suggested reducing the number of mechanical lubricators and increasing the introduction of more electric units to maintain effective lubrication and longer life of rail profiles, but did not provide any guidelines on how to improve grease distribution through the curves.

### 2.5.4 Factors Influencing Effective Lubrication

Literature on rail curve lubrication reveals various factors affecting wayside lubrication (Reiff 2006; Thelen & Lovette 1996; Dunseth et al. 1987; Sroba et al. 2001; de Koker 1994). Many of the potential factors are listed below:

- Location and placement including positioning of applicator bar.
- Lubricator unit itself depending on the technical aspects and ability for precise application of lubricant.
- Properties and composition of lubricants.
- Wheel/rail temperature.
- Axle loads, lateral and longitudinal forces, creepage, curve radius and gradient.
- Speed, frequency, length and condition of trains.
- Wheel/rail vibration (causes severe grease fall off and less carry distance).

- Surface roughness.
- Wheel/rail profile conformity.
- Track surface irregularities.
- Environmental factors such as climatic condition weather seasons, rain, dust, snow, vegetation and different terrains
- Dedicated maintenance, management and condition monitoring of the systems.
- Technical expertise.

Detailed research into application methods, dispensing equipment, rate of grease application, impact of grease and its properties are rare.

Lemma et al. (2014) found significant differences in friction measurements inside tunnels, in the open atmosphere and under snow cover. They also emphasised further investigation on the influence of surface roughness, humidity and temperature on friction levels. Lewis et al. (2014) reported that laboratory tests showed grease retentivity is significantly affected by surface roughness of rail wheel discs; smooth discs with 7.5% slip achieved the best retentivity compared to smooth discs with 10% slip, and rough discs with 10% slip achieved the worst retentivity for the same grease. Smooth rail wheel discs with least slip achieved highest retentivity compared to rough rail wheel discs with higher slip. High wear rates were observed under lubrication starvation tests due to grease wearing out and causing severe localised wear. It was also reported that there is a distinct inverse relationship between surface roughness and grease retentivity, and an inverse relationship between retentivity and disc wear rate. Grease retentivity is widely used in the wayside rail curve lubrication research. Grease stability and retentivity are considered as vital factors for the performance of rail curve grease though various other factors are still unknown which may contribute in the grease exchange and carry down towards the furthest distance (Eadie et al. 2013). IHHA (2001) defined retentivity as a measure of the time or number of wheel pass or MGT that a lubricant is able to retain its lubricity. When the microscopic asperities on rail and wheel surfaces contact each other, it causes frictional heat generation and may give rise to flash temperatures from 600 to 800 degree C. At such high temperature greases are burned off and leave the residue on the rail wheel surfaces. Due to this effect the coefficient of friction rises from .05-0.1 to dry lubrication with coefficient of friction about 0.60. Retention of grease between the mating surfaces is necessary in industrial bearings. It is easy to retain grease in the bearing system as it has a thickener that retains the base oil. It has been suggested that there are clear performance differences between different greases based on laboratory tests, but no field tests were conducted to justify this from real life applications. Evans (2013) suggested that high quality more durable grease should be developed to perform better and not to degrade within storage tanks and grease delivery systems. Lundberg and Berg (2000) reported that grease for rail curve applications must be suitable for the actual railway application, must be fully formulated with extreme pressure additives, commercially viable, recommended by suppliers and customers of the desired application with other appropriate technical and environmental needs.

Literature also reported the following issues that need to be considered in proper application of wayside lubrication systems (Reiff 2006)-

- Selection of the most appropriate equipment for dispensing lubricant.
- Selection of the optimal type of lubricant for the particular operating conditions.
- Regular measurement and management of lubrication effectiveness.
- Evaluation of wear data at regular million gross tonne (MGT) traffic intervals and reporting of lubricator performance.
- Optimal positioning of lubricators for optimal grease pick-up and longer carry distance.
- Dedicated maintenance and servicing program with effective training.
- Regular evaluation of program policy, performance of lubricators and lubricants.
- Regular communication with vendors about product performance and problems.

QR Standard Practice No. 3707 (1995) suggested that lubricators should not be located where heavy sanding for locomotive traction is common, where maintenance operations may be obstructed or there is a possibility that grease splatter on the head of the rail could cause trains to over-run signals or platforms. According to Sroba et al. (2001), some selection criteria for appropriate lubricator units are:

- Ease of installation and simplicity of operation.
- Reliability of performance and easy to maintain.
- Availability of spare parts.
- Availability of lubricant to be used.
- Economically viable.

Sroba et al. (2001) also recommended that a remote monitoring system for on-line condition monitoring should also be considered.

### 2.5.5 Friction Management Guidelines

As stated earlier, coefficient of friction management is necessary because, in some cases, minimal friction is required whereas, in other cases, an optimum value is required. Friction management is the process of controlling friction properties in the wheel/rail interface to achieve various benefits (Lemma et al. 2014; Harrison, McCanney & Cotter 2002; Sroba et al. 2005; Transportation Safety Board of Canada 2006). The literature reports the following guidelines:

American Railway Engineering and Maintenance-of-Way Association (AREMA) recommends (Reiff 2006, CRC Australia 2014):

- Gauge Face (GF) friction values should be < 0.20.
- Gauge corner friction value should be < 0.20 which was under review.
- TOR friction value should be 0.35 +/-0.05.
- Left to right rail friction value differential should be < 0.1.

Canadian Pacific Railway (CPR) recommends (Sroba et al. 2001; Roney 2004; Sroba et al. 2005; Lemma et al. 2014, CRC Australia 2014):

- Maintain top of rail friction coefficient differential, left to right < 0.1.
- Top of rail friction coefficient should be $0.3 \leq \mu \leq 0.35$.
- High rail gauge face coefficient of friction $\mu \leq 0.25$.

In rail lubrication, the coefficient of friction was also categorised as follows (Kalousek et al. 1996; Frohling, de Koker & Amade 2009):

- Low < 0.2.
- Intermediate between 0.2 and 0.4.
- High > 0.4.

Sims, Miller & Schepmann (1996) considered the coefficient of friction of the rail head for various lubrication levels should be as follows:

- Dry rail >0.45.
- Unlubricated rail between 0.35 and 0.45.
- Effective lubrication between 0.10 and 0.25.

A lubrication chart was developed by South Africa's Transnet Freight Rail based on the level of surface protection to quantify the effectiveness of high rail application (Frohling, de

Koker & Amade 2009). Table 2.2 shows the summary of lubrication implemented in Transnet Freight Rail's system.

Table 2-2: Lubrication chart summary (Frohling, de Koker & Amade 2009)

| Classification | Coefficient of friction | Description |
|---|---|---|
| Dry | 0.35 to 0.57 | No grease on wear face. |
| Poor | 0.30 to 0.35 | Lubricant on 10 to 40% of the wear face |
| Acceptable | 0.25 to 0.30 | Lubricant on 40 to 60% of the wear face. Metal still visible through lubricant. |
| Average | 0.20 to 0.25 | Lubricant on 60 to 90% of the wear face. Grease sticky and thick. |
| Good | 0.15 to 0.20 | Lubricant on 100% of the wear face. Grease is still fresh and wet. |
| Too much | Less than 0.15 | Gauge face and rail head covered by a film of lubricant. |

### 2.5.6 Coefficient of Friction Measurement

There are several methods to measure the coefficient of friction between rail and wheel at the contact interface. The following instruments and/or techniques have been widely used to measure rail lubrication effectiveness in field applications.

- Hand-pushed tribometer as shown in Figure 2.7 (Sroba et al. 2001; Harrison, McCanney & Cotter 2002 ; Roney 2004 ; Lemma et al. 2014; Areiza et al. 2015; Uddin et al. 2009 ; Uddin et al. 2010 ; Uddin et al. 2011 ; Uddin et al. 2013 ; Uddin et al. 2014, CRC Australia 2014) is used to measure the friction between the rail and wheel.
- High speed tribometer measures the coefficient of friction of the tread and gauge face both rails. It is used to measure the coefficient of friction over large distance of track network. Reiff (2006) reported that high speed tribometer can only produce a database of friction values which only determines part of the wheel/rail interface condition which does not always fully correlate with the total system performance. Wheels would be subject to lubricants and friction modifiers. IHHA (2001) reports that a hi-rail tribometer measures coefficient of friction for a large distance at speeds of up to 30 km/hr. Another advantage of hi-speed tribometer is it can simultaneously measure top of rail and gauge corner data for both rails.

- Lubrication level (Goop) gauge is a calibrated simple template to measure the level of grease on the rail and useful to assess the rate of grease propagation around the rail curves (Rastall n.d.).

- Instrumented wheel set temperature measuring instrumentation is equipped with temperature measuring sensors. It is capable to measure the wheel surface temperature while the train travelling at its speed. This data is analysed to determine the effectiveness of the applied grease in the wheel/rail interface.


Figure 2.7: Schematic diagram of tribometer (Lemma et al. 2014)

Due to high reliability and repeatability of measurements and location control, the hand-pushed tribometer has been widely accepted as an industry practice for measurement of gauge face and top-of-rail friction.

In laboratory testing, commonly used instruments are (Areiza et al. 2015):

- Pin-on-disc tribometer (Harrison, McCanney & Cotter 2002; Lewis, Lewis & Olofsson 2011; Olofsson & Sundvall 2004; Sundh, Olofsson & Sundvall 2008).

- Twin disc machine (Lewis et al. 2014; Gallardo 2008).

- Mini traction machines (Zhu 2011; Zhu, Olofsson & Persson 2012).

- Full scale roller rigs (Zhang et al. 2002; Jin et al.  2004) and many others.

## 2.6 Benefits of Lubrication

There are various benefits rail operators enjoy from rail lubrication and friction control. Major benefits from rail lubrication are shown in Figure 2.8.

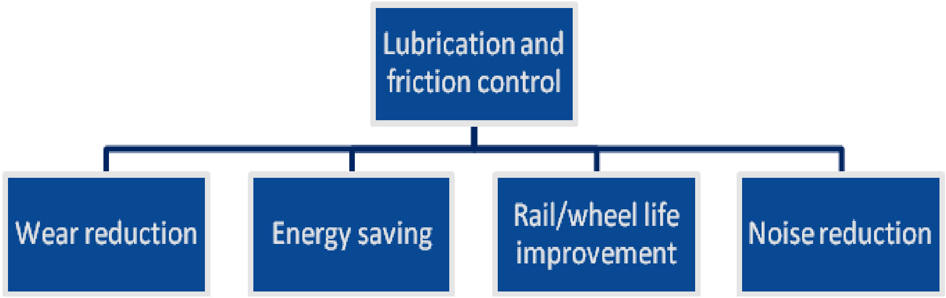



Figure 2.8: Benefits of lubrication and friction control

Train resistance around curves can be reduced dramatically by lubrication of the wheel flange and rail gauge corner interface. Successful lubrication can produce enormous benefits for the rail industry by managing friction at the desired level, reducing wear of rails and wheels, improving the life of rails and wheels, saving energy, reducing noise and indirectly making a huge reduction in the maintenance expenditure on train fleet management. According to the 2014 Annual Report of the Transport Technology Centre Inc. (TTCI), proper friction control and management systems for North America can reduce fuel consumption by approximately 4%, improve rail life by 25%, and reduce train delays (by reducing rail replacement work). The estimated annual benefit of improved friction control and management systems is about US$82 million (AUD107.42 million), or US$1.2 billion (AUD1.57 billion) in total since 1994 (TTCI Annual Report 2014). Table 2.3 shows how wheel wear changed between dry and lubricated rail conditions for different types of wheel material. This table refers to the worn wheel profile as it quantifies the area of cross-section of the wheel profile removed by wear in $inch^2$/1000 miles ($mm^2$/1000 km) of travel. Table 2.4 shows how the rail wear rate changes for different levels of lubrication. The data was recorded by using witness groove technique where transverse groove is cut through the rail. Profilometer was used to measure the wear over the time. Inch /MGT or mm/MGT unit is used to quantify the lateral or vertical wear or gauge corner wear rate per million gross tonnes load transport.

Table 2-3: Wheel type wear comparison for different operating conditions (Sims, Miller & Schepmann 1996)

| Wheel Type | Dry Rail Wear ($in^2$/1000 miles) ($mm^2$/1000 km) | Lubricated Rail Wear ($in^2$/1000 miles) ($mm^2$/1000 km) | Factor of Improvement over Dry Rail |
|---|---|---|---|
| U-low carbon | 0.03104 (12.416) | 0.00140 (0.56) | 22 |
| U | 0.02868 (11.472) | 0.00137 (0.548) | 20 |
| C | 0.01912 (7.648) | 0.00107 (0.428) | 17 |

Table 2-4: Rail wear rate for different lubrication levels (Sims, Miller & Schepmann 1996)

| Level of Lubrication | Wear Rate (inch/MGT) (mm/MGT) | Factor of Improvement over Dry Rail |
|---|---|---|
| Dry rail | 0.005-0.007 (0.127-0.1778) | 1(base) |
| Low | 0.001(0.0254) | 5 |
| Medium | 0.00029 (0.007366) | 17 |
| High | 0.000064 (0.0016256) | 80 |

### 2.6.1 Improvements in Energy Consumption

Reduction in the wheel/rail interface coefficient of friction reduces the train resistance, leading to significant savings in fuel consumption. Effective lubrication must be present on both tangents and curves to obtain the highest fuel savings. If only the curves are lubricated, the flanging effect of bogies rapidly dries off the wheels on long tangents and it is impossible to maintain adequate lubrication between widely separated curves. The study at the Facility for Accelerated Service Testing (FAST) produced Table 2.5 which shows the fuel savings when lubrication was used on the rail (Sims, Miller & Schepmann 1996). de Koker (1993) and Sroba et al. (2001) reported that Spoornet in South Africa required 51% less energy to traverse a very tight curve (8.7 degree or 200m radius) and achieved 28% energy savings by heavy haul trains on the Richards Bay Coal Line by use of rail lubrication.

Reduction in rolling resistance due to rail and wheel flange lubrication of up to 50% around curves and up to 30% on straight or tangent track was measured against unlubricated track in the USA, leading to energy savings between 20% and 30% under service conditions (de Koker 2004). A good correlation exists between energy savings and rail lubrication. Spoornet in South Africa  (year ) conducted a test on a 200m radius curve and found that, for the unlubricated condition, wagons required 54 Newton/tonne load to traverse the curve and only 28 Newton/tonne for the lubricated condition, which is 48% less energy (de Koker 2004).  Operators are able to increase tonnages and add 10% to 20% more wagons in a train if the line is consistently and well lubricated (IHHA 2001; Sroba et al. 2001; de Koker 1993). Table 2.5 shows considerable improvement in energy savings with application of different lubricating systems. It shows wayside lubrication has highest energy savings compare to any other lubricating system as it allows lubrication through all the curves and tangents.

Table 2-5: Energy savings comparison of different lubrication applicators (Sims, Miller & Schepmann 1996; Uddin & Chattopadhyay 2009)

| Lubricating System | Efficiency (gal/MGT) (Lt/MGT) | Energy Savings over Dry Rail |
|---|---|---|
| Dry Rail | 6000 (22712.4) | n/a |
| Wayside lubricator-active | 4100 (15520.14) | 32% |
| 1-in-4 Lub. Car | | |
| Graphite | 4800 (18169.92) | 20% |
| Low graphite | 5300 (20062.62) | 11% |
| Hi-Rail vehicle (1-in-35 trains) | 5500 (20819.7) | 8% |
| On-Board | 5140 (19456.956) | 14% |

The unit Gal/MGT in Table 2.5 demonstrates the efficiency of the locomotive in gallons for hauling a million gross tons (MGT).

IHHA (2001), Sroba et al. (2001) and Reiff and Creggor (1999) reported that a 30% reduction in fuel savings was achieved with generous lubrication compared to dry conditions in the test conducted at the FAST facility. A lubricated top of the low rail and generous high rail gauge face lubrication also significantly contributed to the reduction of lateral forces in the curves. A NUCARS analysis by TTCI showed energy savings of 15% (wayside lubricators), 39% (top-of-rail friction modifier alone) and 65.5% (top-of-rail and gauge face lubrication together) (IHHA 2001; Sroba et al. 2001; AAR 2000). In one of the toughest railway operating environments in Canadian Pacific Railway's (CPR) Thompson Subdivision, tests conducted between March 2000 and May 2001 found that improved wayside lubrication reduced gauge face wear by 87% on all sharp curves and, with an optimal setting of grease application, the gauge face wear rate between February 2001 and May 2001reduced by 100% compared to the base case.

Hanson et al. (2014), Anderson (2004), Jiang et al. (2013a), Uddin et al. (2010a) and Jiang et al. (2013b) reported that a study conducted by RailCorp showed that the gauge face lubrication of both rails delivered the greatest mitigation in squeal noise and identified that the squeal noise was generated under four distinct wheel/rail interactions, namely at the top of the high and low rails, and at gauge corner of the high (majority of squeal) and low rails. Figure 2.9 shows the summary of the test results which indicates significant reduction of noise achieved from gauge face lubrication on both rails. In this test it has been considered that the influence of three main system such as the rolling stock, the track form and the wheel/rail interface are the main source of squeal. Data from all of these sources has been weighted and normalised to develop the summary of the test results. These results are supported by trials conducted by Sroba et al. (2005) and Uddin et al. (2010a). Effective lubrication is seen as a key component of squeal mitigation.

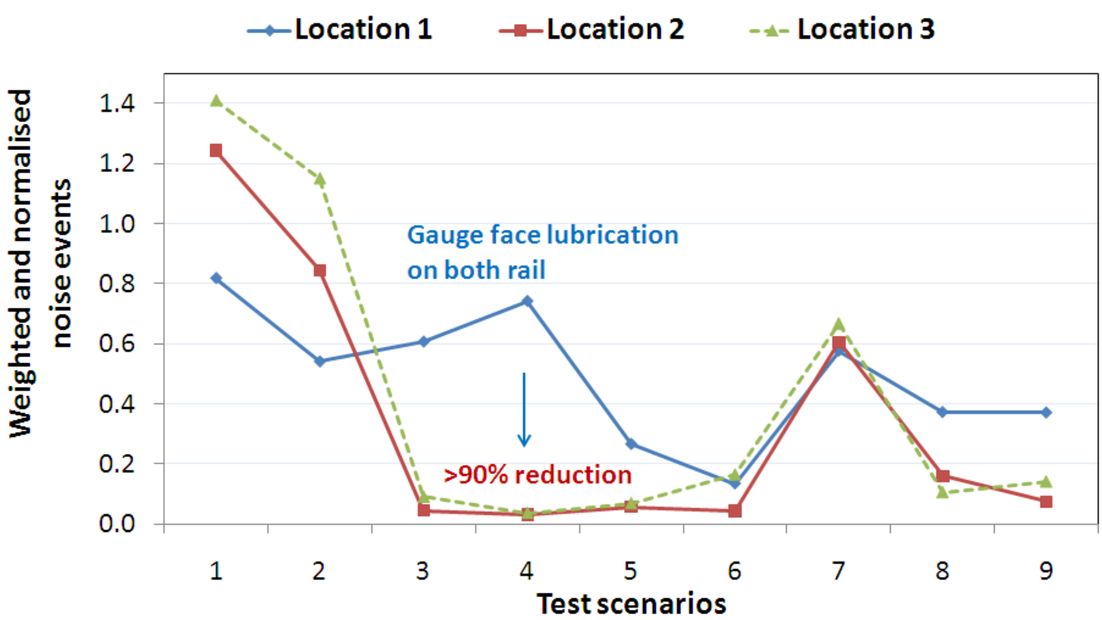



Figure 2.9: Summary of test results conducted by RailCorp to mitigate wheel squeal through effective gauge face lubrication (Hanson et al. 2014)

### 2.6.2 Wheel/Rail Life Improvement and Cost of Lubrication Implementation

Lubrication at the wheel/rail interface dramatically reduces both wheel and rail degradation. Reddy et al. (2006) reported Eurostar's estimated annual savings of £1,000,000

(AUD 1,840,000) on maintenance and wheel replacement costs achieved with effective lubrication. Table 2.6 shows the Express Rail Link/Malaysia recorded data on improvement of wheel life and reduction of annual wheel costs compared to no lubrication.

Table 2-6: Reduction of wheel maintenance due to lubrication (Larke 2003; Reddy et al. 2006; Uddin & Chattopadhyay 2009)

| Track/vehicle condition | Wheel life (km) | Wheel life (week) | Annual wheel cost (£) AUD) |
|---|---|---|---|
| No lubrication | 170,000 | 20 | 1.6 million (3.024 million) |
| Rail lubrication | 300,000 | 35 | 825,000 (1.559 million) |
| Vehicle lubrication | 1,000,000 | 118 | 250,000 (472,500) |
| Target | 1,500,000 | 177 | 170,000 (321,300) |

Compared to wheel/rail life improvement, energy savings and other benefits of lubrication are significantly lower. Table 2.7 shows the cost of lubrication by different railway operators around the globe.

Table 2-7: Lubrication cost to rail players (Larke 2003; Reddy et al. 2006; Uddin & Chattopadhyay 2009)

| Railway | Quantity of Lubricant (tonnes/yr) | Lubricator (£/yr) (AUD/yr) | Lubricant (£/yr) (AUD/yr) | Cost (£/yr) (AUD/yr) |
|---|---|---|---|---|
| Spoornet | 200 | 125,000 (236,250) | 134,000 (253,260) | 259,000 (489,510) |
| IBCV | Not known | 325,000 (614,250) | 279,000 (527,310) | 604,000 (1,141,560) |
| HKMTR | 0.3 (depots) | 5,600 (10,584) | 550 (963.9) | 6,150 (11,623.5) |
| Eurostar | 1.1 | Not known | Not known | 70,000 (132,300) |
| Banverket | 20 | Not known | 31,000-62,000 (58,590-117,180) | Not known |
| DSB | 25 | 95,000 (179,550) | 100,000 (189,000) | 195,000 (368,550) |
| SNCB | 40 | 1,000,000 (1,890,000) | 479,000 (905,310) | 1,479,000 (2,795,310) |

Network Rail’s rail life cycle cost analysis shows that a joint strategy of rail profile management with appropriate preventive rail grinding and lubrication incurs the least rail life cycle cost (LCC) (Evans 2013). Figure 2.10 shows Network Rail’s rail life cycle cost analysis based on use of different grades of rail and implementation of lubrication and preventive rail grinding. This graph shows that the payback period for premium rail with grinding and lubrication is 3 to 4 years.

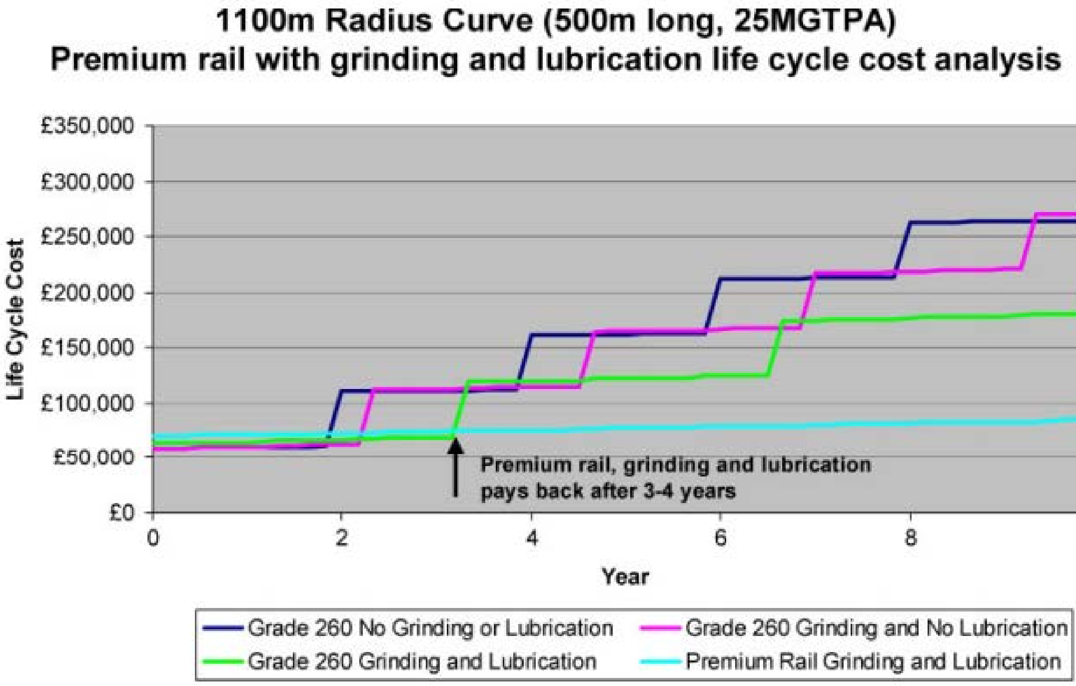



Figure 2.10: Network Rail’s rail life cycle cost analysis based on use of different grades of rail, implementation of lubrication and preventive rail grinding (Evans 2013)

Union Pacific’s Tehachapi studies and Canadian Pacific’s total friction management have confirmed the benefits of gauge face and top-of-rail friction management. The improvement of rail life in these studies provided the justification for financial commitment to widespread implementation of friction management (Gilliam & Rholfs 2013; Roney 2009).

Caldwell (2013) reported that wear rate reductions of 8 to 20 times are readily achievable for standard carbon rail with gauge face lubrication and proves its effectiveness for the minimisation of gauge face wear. For head-hardened rail, wear rate reductions are only 2 to 3 times. That author also reported that effective lubrication, improved rail quality, wheel/rail profile design and the use of flexible bogies combined together can virtually eliminate measurable gauge face and wheel flange wear and can also minimise rolling contact fatigue.

## 2.7 Demerits of Excessive Lubrication

Though lubrication has enormous benefits in rail operations, excessive lubrication is considered harmful. Excessive lubrication could cause:

- Adverse impacts on operating conditions and the environment.
- Rollover or derailment.
- Excessive sanding which cause excessive wear.
- Abnormal bogie behaviour.
- Excessive braking distance.
- May cause difficultly in stopping trains at stations if the lubricators are around the station.
- Top of rail contamination which can result in abrasive wear and reduced traction and braking capacity.

The Federal Rail Road Administration has reported that brake related failures reduced the life of wheel sets by more than 50% from 393006 miles (632739.66 km) on average to 194406 miles (312993.66 km) and cost around U$ 555 million (727.05 million) for wheel replacement (FRA 2006).

## 2.8 Lubricator Placement Model

Wayside lubricator placement guidelines were developed by different rail networks over the last few decades. Most of them depend on arbitrary assumptions and no specific formula was developed. The first placement formula for wayside lubricators was developed by de Koker (1994) which was extended by Sroba et al. (2001). Marich et al. (2001) recommended that lubricator position on track should be as follows:

- Within the transition spiral of curves of 400-600m radius.
- Ideal position is within the transition where wheel flanging just starts.
- Within the body of the curve with radius greater than 600m up to 1000m.
- Lubricators should not be placed on the tangent track for curves greater than 1000m radius since there is no flanging.
- If possible, lubricators should not be placed at curves with radius less than 300m.
- In the transition spiral at the beginning of the curve or the end of the curve depending on traffic direction.

In the model described in de Koker (1994) and Sroba et al. (2001), the length of track was adjusted by a number of track and traffic related factors to determine the lubricator placement interval. That lubricator placement model can be described as follows.

This is a basic model of wayside lubricator placement. For the total placement model, a hierarchical approach needs to be considered. If the effects of considered factors can be determined, the data can be utilised to calculate the distance between lubricators and consequently the positioning. Field and laboratory tests are needed to quantify the effects of the factors.

Properties of curved and tangent track, locomotive axle loads and wheel configurations, train speed and length, wheel/rail profile, wheel/rail temperature, weather conditions, and lubricator reliability have significant effects on the carry distance of grease. Carry distance determines the distance between consequent trackside applicators. Carry distance is measured based on the coefficient of friction on the gauge face. Sroba et al. (2001) suggested that the required coefficient of friction is considered to be 0.25 with measurements taken by a hand-pushed tribometer. Lundberg et al. (2015) reported that more research is needed to determine the optimal volume of lubricant and the distance covered by the lubricant/friction modifiers.

Model development for wayside lubricator placement should consider location and position of lubricator, types of applicator bar, lubricant characteristics, track and traffic parameters, plus environmental and human factors.

It is necessary to investigate the following issues and extract the information on the effectiveness of lubrication

## 2.9 Effect of Wheel/Rail Temperature on Rail Lubrication

Wheel/rail temperature has a significant effect on wheel/rail lubrication and their physical structure. Frohling, de Koker and Amade (2009) and Pasta et al. (2005) reported on the joint investigation by Companhia Vale do Rio Doce (CVRD), Estrada de Ferro Vitoria a Minas (EFVM), and the Transport Technology Centre Inc. (TTCI) on lubrication practice and found that all of the above operators emphasised the improvement of rail lubrication through high durability/retentivity grease and optimisation of the hi-rail grease application schedule. Over a 10 day period, monitoring of coefficient of friction and rail temperatures showed that a typical 15800 tonne loaded 160-wagon ore train in a 160m radius curve increased the temperature of a dry rail by 31.6 ° C, whereas the temperature of a lubricated rail increased by only 7.9°C and the coefficient of friction μ on the gauge face exceeded the target value of 0.35 for between five to ten trains. Frohling de Koker and Amade (2009) reported on Transnet Freight Rail's (South Africa) investigation to establish any possible relationships between temperature rise in the gauge corner versus coefficient of friction (μ) and the mass, speed, wagon number, or the bogie type of the train. The results found a strong correlation between the increases in temperature rise and the length of the trains, and also between the temperature rise and the accumulating flange forces. Wheel/rail temperature could have a significant role on grease durability and lubrication effectiveness. Fischer, Daves and Werner (2003) reported that temperature at the wheel/rail interface may vary from ambient temperature to nearly 900°C based on maximum pressure, coefficient of friction, contact patch, creepage and other factors. Further, de Koker (2010) reported that the temperature distribution of 1.2 million wheels on the Richards Bay Coal Line between February and May 2008 showed that 64% of wheels were at temperatures between 130°C and 190°C. 2% of wheels had temperatures above 225°C, the flash point of the lubrication grease used. According to Ertz and Knothe (2003), thermal stresses caused by wheel/rail temperature plays a significant role on the elastic limit and the shakedown limit of wheels and rails as it is superimposed on the mechanical contact stresses. This reduces the elastic limit of the wheel and rail steels and yielding begins at lower mechanical loads. It was claimed that elastic and shakedown limits of rail can be reduced by a factor of two.

## 2.10 Effects of Rail and Wheel Profiles on Lubrication Carry Distance

Rail and wheel profiles play an important role in lubrication effectiveness. According to Thelen and Lovette (1996), success of the lubrication strategy depends on the transport mechanism; wheel flange contact with the rail is used as the lubricant transport mechanism in wayside and on-board lubrication systems. Determination of the level of friction at the wheel/rail interface is a significant condition monitoring requirement for the rail industry and it is very critical to ensure long term success (Lemma et al. 2014; Harrison, McCanney & Cotter 2002). Conformal flange contact is an optimum condition for non-steering vehicle bogies and supports lubrication (IHHA 2001). Contact at the wheel/rail interface needs to be considered in the evaluation of lubrication effectiveness.

## 2.11 Overview of Wear

Wear involves chemical and physical interactions with the mechanical components (Meng & Ludema 1995). Instead of a material property, wear is a systems response. The type of friction needed to generate wear can be classified as either rolling wear, sliding wear, fretting wear or impact wear. These could be further subdivided into abrasive, adhesive, two body, and three body wear. Based on mechanisms, wear can be explained as mechanical wear, chemical wear and thermal wear. Mechanical wear is mainly a process of deformation and fracturing. Chemical or tribo-chemical wear is governed by the growth rate of a chemical reaction film which is accelerated mechanically by friction. Thermal wear is governed by local surface melting due to frictional heating. Wear is a multi-parameter sensitive phenomenon (Kato 2002). To quantify or reliably predict wear rates, a wear model is important. Constructed wear models with data taken from tests with limited test conditions are empirical wear models. Barwell's study (cited in Meng & Ludema 1995, p. 444) suggested that wear rates may be expressed using any of the following three equations.

$$V = \frac{\beta}{\alpha}\{1 - \exp(-\alpha t)\} \tag{2.2}$$

$$V = \alpha t \tag{2.3}$$

$$V = \beta \exp(\alpha t) \tag{2.4}$$

where V is the volume loss ($m^3$), $\alpha$ is a constant and t is time (sec) and the parameter $\beta$ represents some characteristics of the initial surfaces.

Archard's wear model (Archard 1953) is one of the earliest models, and can be expressed as follows:

$$W = Ks\frac{P}{Pm} \tag{2.5}$$

where W is the worn volume, s is the sliding distance, P is the applied load, $P_m$ is the flow pressure and ratio of the last two is the contact area and K is the wear coefficient.

According to Ludema (1996) and Meng and Ludema (1995), although there are more than 300 wear equations and more than 1000 research papers published in the last 40 years, the shortage of appropriate and effective models is due to the complicated nature of friction and wear. A multidisciplinary approach to combat complexity in wear modelling is very rare.

To overcome the lack of completeness of wear models, Ludema (1996) recommended to:

- Consider a wide range of variables in tests, describe the conditions of tests.
- Describe test conditions, test materials, mechanical dynamics, environment, and test geometries completely.
- Write equations as a reflection of actual data.
- Engage researcher from all relevant disciplines which contribute in wear and friction.
- Use field tests to ensure the fruitfulness of laboratory tests.

Wear is related to contact pressure on an elastic plastic body and the elastic and plastic physical deformation. "According to the 'adhesion theory' of friction of Bowden and Tabor, wear particles are plucked out of the softer surface by strong adhesion.The coefficient K in Archard's 1953 wear model (Johnson 1995) is understood as the probability of the adhesion at any asperity to be strong enough to pluck out a wear chip".

### 2.11.1 Wear Mechanism in Dynamic Contact

Continuous sliding behaviour was introduced in a study by Challen and Oxley (1979) where a hard asperity was modelled as a rigid wedge which traverses the contact surface. Three types of deformation were identified, namely:

- Plastic flow of the surface by a wave pushed ahead of the wedge.
- A deformed prow which becomes detached from the surface.
- A cutting mode in which a chip is continuously cut from the surface.

### 2.11.2 Material Response to Cyclic Loading

Shakedown and ratchetting are responses to cyclic loading. Johnson (1995) identified 3 categories of response of an elastic-plastic structure under cyclic loading:

- At sufficiently small loads, smaller than the elements yield point, the response is perfectly elastic and reversible.
- Loads within the 'elastic shakedown limit' involve a situation where plastic flow takes place on first loading, residual stresses, strain hardening or deformed geometry may enable the structure to 'shakedown' to a perfectly elastic response in the cyclic steady state. The maximum load for which this is possible is known as the 'elastic shakedown limit'.
- Above the shakedown limit, plastic deformation takes place for each cycle of load;

either a closed cycle of plastic strain is obtained (plastic shakedown) or repeated increments of accumulated unidirectional plastic strain (ratchetting).

Figure 2.11 shows that there is a discontinuity in the map at $\mu \approx 0.25$. This critical value of shakedown is controlled by subsurface stresses and, above it, plastic flow occurs at the surface. The shakedown limit decreases inversely as the value of $\mu$ increases.

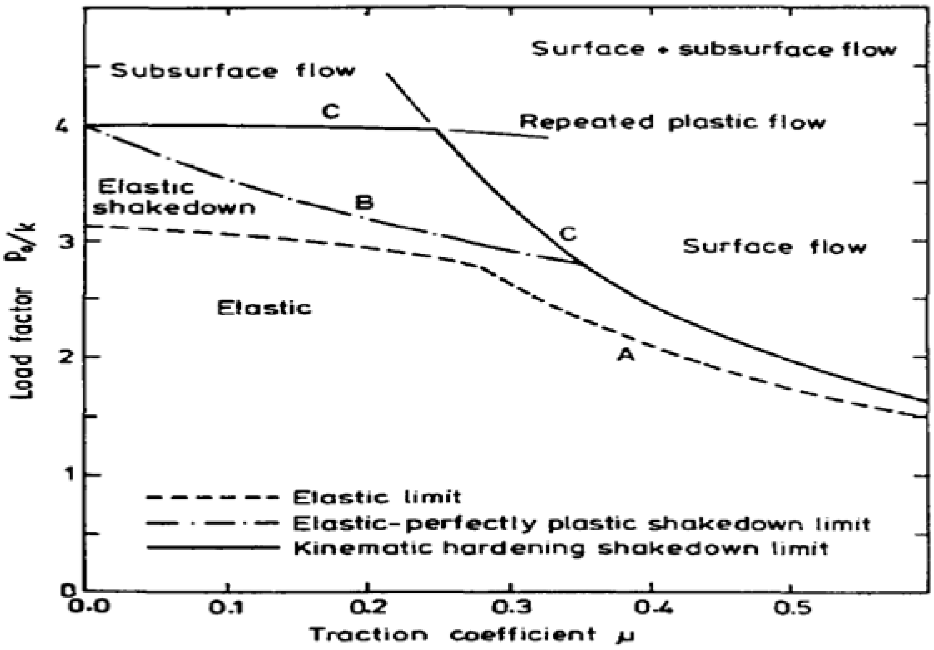



Figure 2.11: Shakedown map for repeated sliding of a rigid cylinder over an elastic-plastic half space (Johnson 1995)

### 2.11.3 Running-in Wear on Rough Surfaces

Running-in, mild and severe wear are three typical wear regimes of a wear system (Figure 2.12). Running-in wear is of particular interest in MMC/Steel systems (Zhang, Zhang & Mai 1996). It refers to the change in profile of nominally flat, rough surfaces during repeated sliding. It may take place by the action of wear, but in the short term it occurs by plastic deformation as a shakedown process. It assumes that the height and curvature of the plastically deformed asperities is reduced such that, in the steady state, the local contact pressures are equal to or less than the shakedown limit (Johnson 1995). Running-in also describes the change of the wear rate in lubrication. In the early stages, wear is caused by the brittle fractures in the surface grains and, in the later stages, by tribo-chemical reaction (Kato 2002).

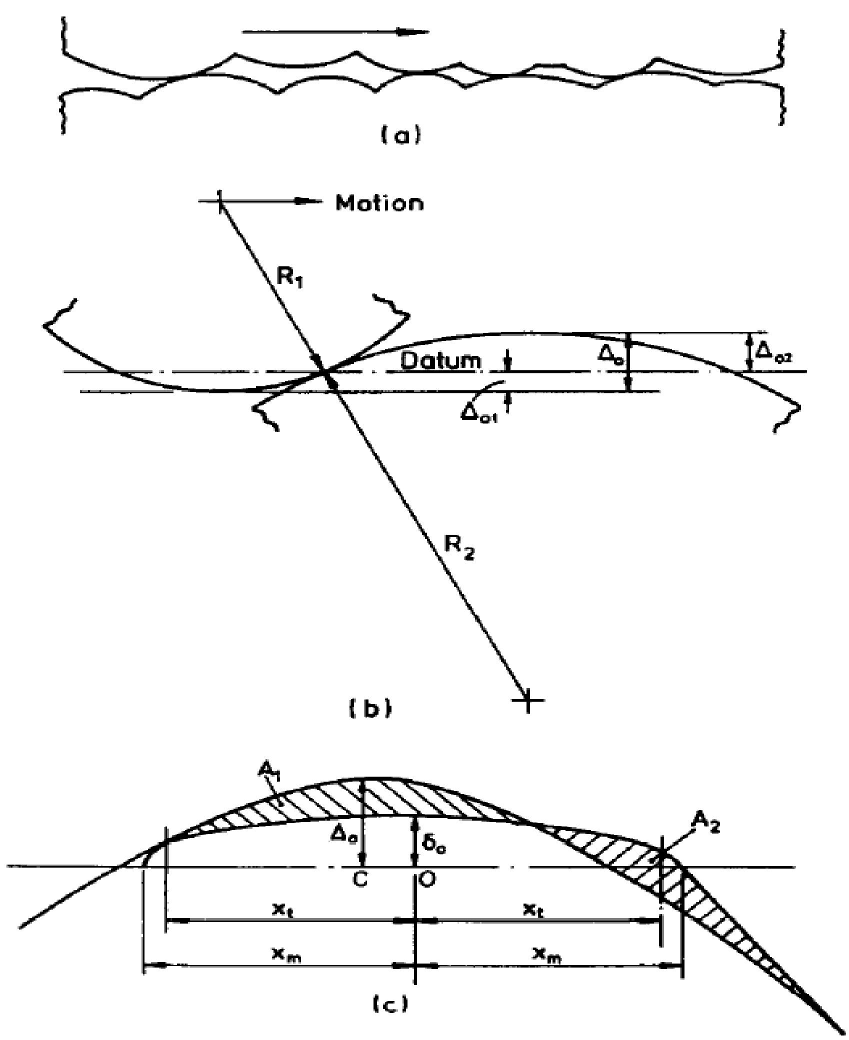



Figure 2.12: Running-in rough surface: (a) Model of 2-D rough surfaces in sliding contact, (b) the 'unit event': two asperities with initial interference $\Delta_0$, (c) Steady state profile (Johnson 1995)

It is considered that, in the 'running-in' process, the unit event takes place when the cylindrical asperities push past each other with an initial interference $\Delta_0$ as shown in Figure 2.12 (b). In case of different hardnesses, if only one surface is deformed plastically the shakedown limit is reached when the soft asperity is squashed flat. Steady state profiles of the asperities can be calculated according to Figure 2.12 (c)

### 2.11.4 Ratchetting in Sliding Contact (μ<0.2)

In lubricated conditions, wear takes place in a very thin film <1.0μm thick, which is extruded from the edges of the asperities on the softer surface. This behaviour was described by Akagaki and Kato (1987) as 'filmy wear' shown in Figure 2.13. The experiments were carried out with a hard slider in contact with a machined softer surface under boundary lubricated (μ<0.1) conditions. It was observed that materials were extruded from the trailing edge of the transversely oriented soft asperities. However, the lateral extrusion occurred also when the sliding action was applied along the asperities.

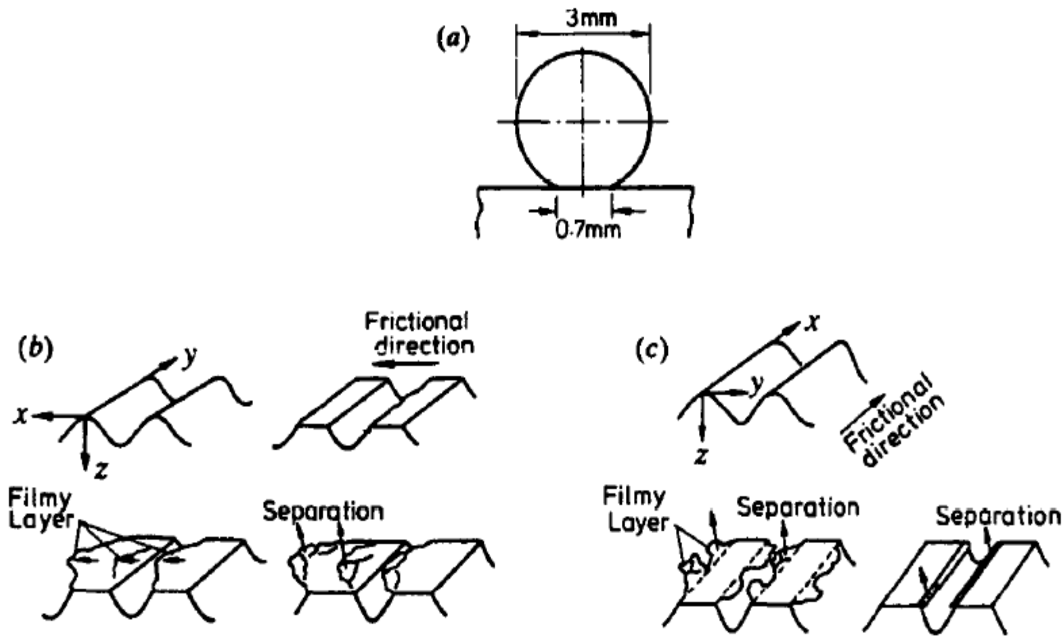



Figure 2.13: Formation of 'filmy wear' particles: (a) Hard ball with ground flat sliding on softer ground surface. Sliding (b) perpendicular and (c) parallel to soft asperities (Akagaki & Kato 1987)

Kapoor (1994) proposed a kinematically acceptable mechanism of extrusion in which a thin layer is uniformly compressed by shearing on the interface between the layer and the bulk of the softer asperities. Bower and Johnson (1989) analysed ratchetting that occurred by an elastic cylinder sliding over an elastic-hardening half-space. Johnson (1995) reported that the ratchetting rate is much enhanced by friction coefficients over 0.25 and plastic deformation is located at and close to the surface.

## 2.12 Wear in the Wheel/Rail Interface

Wheel/rail systems go through a variety of damage modes such as rail gauge face wear, surface fatigue cracking, corrugation, head checking and wheel spalling. These may cause the failure of a heavy haul wheel/rail system or ultimately derailment (Wang et al. 2014; Stuart et al. 2002; Zhong et al. 2011; Frohling, de Koker & Amade 2009). Wheel/rail wear significantly affects the life and performance of rails and wheels. Side wear of rails and wear of wheel flanges significantly affects the service life of rails and wheels (Wang et al. 2014; Wang et al. 2013). Continuous increases in axle load and excessive traffic volumes leads to excessive wear and significantly decreases the service life of rails and wheels in heavy haul lines; Datong-Qinhuangdao railway became the heavy haul line with the highest transportation volume of coal. The statistics of annual transport volumes on the Datong-Qinhuangdao network show it reached 254 million tonnes in 2006 compared with the 100 million tonnes of annual design capacity, and rapidly increased in volume up to 410 million tonnes in 2010 which caused a significant increase in the rail side wear (Wang et al. 2013; Sandstorm & Ekberg 2009; Li et al. 2008; Gonzalez-Nicieza et al. 2008). This capacity growth and resultant wear is shown in Figure 2.14. Rail side wear measurements are taken in the hi rail gauge face side in the curves. Pre-defined data points are selected and then wear measurements in mm are measured and can be calculated average wear in volume per mm length of rail. it is the amount of metal in its volume removed from rail head due to rail wheel contact in the curve.

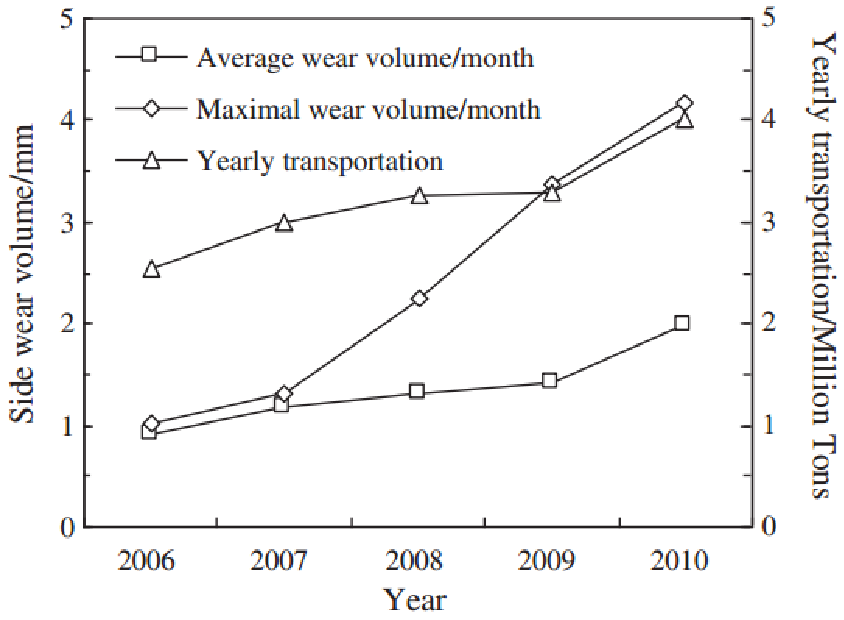



Figure 2.14: Change of rail side wear volume from 2006 to 2010 in a curved rail with radius 400m on Datong-Qinhuangdao heavy haul railway (Wang et al. 2013)

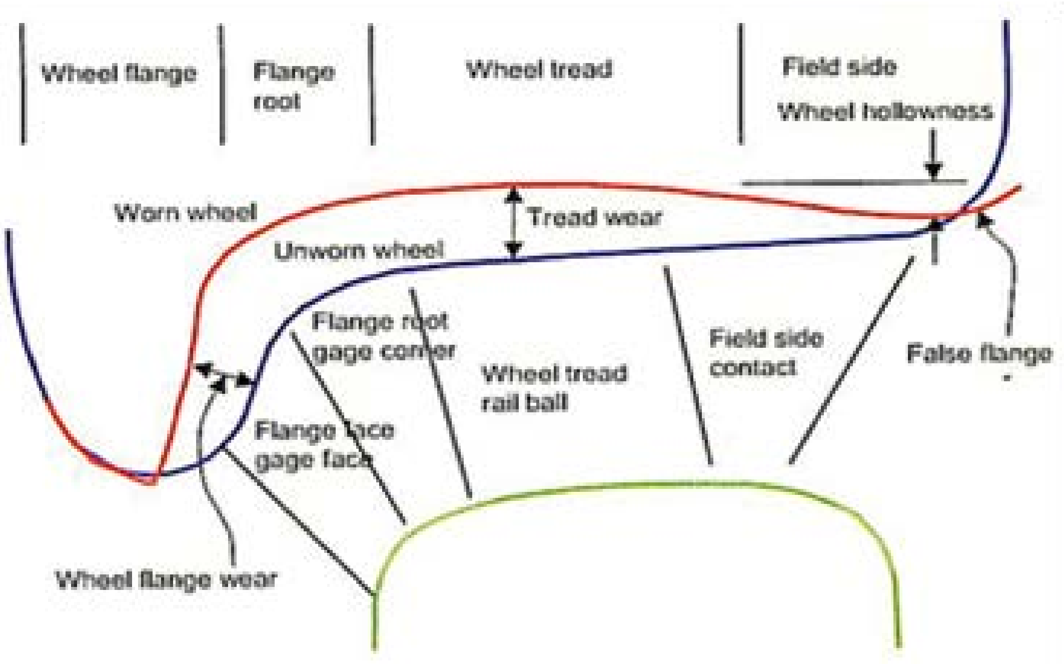
Wang et al. (2013) also reported that the track carrying empty trains has very light rail side wear due to the light axle loads and transport volumes which indicate these two quantities, along with wheel/rail profiles, are heavily correlated to the heavy haul rail damage. Evans (2013) reported that service life of rail is seriously influenced by prolonged duty conditions and deteriorated by a combination of factors. In Network Rail, the total inspection, maintenance and renewal cost is estimated at 200-220 million British pounds (AUD378-415.8 million) per year, where nearly 40% is due to combating RCF and, Evans

(2013) suggested, the combined efforts on implementation of rail profile management technologies such as wheel/rail lubrication, rail grinding, top-of-rail lubrication and high quality premium rail steel in rail renewal. In sharp curves, wear becomes more dominant. The level of wear depends on temperature, track geometry, applied force, type of material or material layers, operating speeds and other operating conditions. Wear in railroad applications involves traction, angle of attack and load, where angle of attack has the greatest effect on flange wear (Waara 2001). Danks and Clayton (1987) investigated wear by using an Amsler twin-disk machine, namely wear on the top of the rail and wear on the gauge face. Zhong et al. (2010) and Kapoor, Schmid and Fletcher (2002) reported on rail damage as the attrition loss, the corrugation loss, the contact fatigue, the side abrasion and stripping.

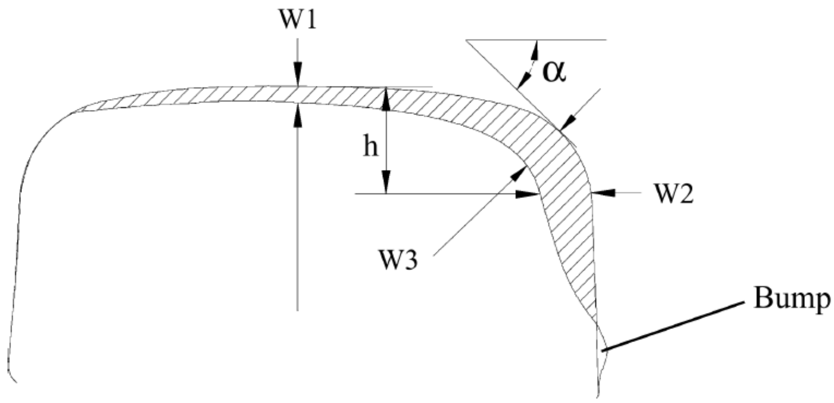
Allen and Reiff (1985) investigated the effect of different levels of lubrication on wear rates. From field tests, Rippeth, Kalousek, & Simmons (1996) showed that the life of track sections originally worn out after 18 months could be extended up to four to five years through proper lubrication and rail grinding. Elkins, Reiff and Rhine (1984) showed that even moderate levels of lubrication on standard carbon rail could reduce wear by a factor of 17 compared with dry rail. For a low level of lubrication, the relative improvement was by a factor of 5.

### 2.12.1 Wear Zones on Wheel and Rail

The area worn away is defined as the area between the unworn wheel and rail profiles and the currently measured profiles and is calculated from where the profiles intersect on the rail head to the intersection point with the lower inner flange (Waara 2001). Figure 2.15 shows the various wear zones on wheel and rail.




(a) Wheel wear zones (Esveld 2001; Uddin & Chattopadhyay 2009)




(b) Worn rail profile (the area worn away is shaded) - W1, rail head wear; W2, horizontal rail flange wear; W3, gauge corner wear (Waara 2001; Uddin & Chattopadhyay 2009)

Figure 2.15: Wear zones on wheel and rail

Danks and Clayton (1987) analysed three types of wear by using an Amsler twin-disk machine. Wear is common on the top of the rail and on the gauge face. According to Waara (2001), four methods can be used to evaluate rail wear -

- Comparing the difference in worn area between two rail profiles.
- Comparing the vertical wear on the rail head – W1.
- Comparing the horizontal wear at a vertical distance, h, from the rail head – W2.
- Comparing the wear measured at some angle, α, on the rail or gauge corner between two profiles - W3.

### 2.12.2 Factors Causing Wear

The nature of the shape change of the rail and wheel is a function of the wear and material flow caused by various contact conditions which depend on the track curvature, vehicle alignment, axle load, vehicle speed, vehicle type, traction and braking (Tourney & Mulder 1996).

#### *2.12.2.1 Wheel/Rail Interface Condition*

Wear is a result of friction between wheel and rail. Hardwick, Lewis and Eadie (2014) reported that twin-disc wear testing under dry, water and grease lubricated conditions of 260 grade rail against R8T wheel material showed that each third-body condition has its own distinct wear line and, under the same slip range, water contaminated contact will reach the severe regime and grease contact will reach the mild regime. It also reported that wear increases when slip increases, little to no surface damage or subsurface deformation of rail discs was observed under grease application compared to water conditions and dry condition. Figure 2.16 shows the shakedown diagram with dry, water and grease tests overlaid using actual traction coefficients and contact pressures (1500 MPa) with shear yield strength values for 260 grade rail (Hardwick, Lewis & Eadie 2014; Eadie et al. 2008).

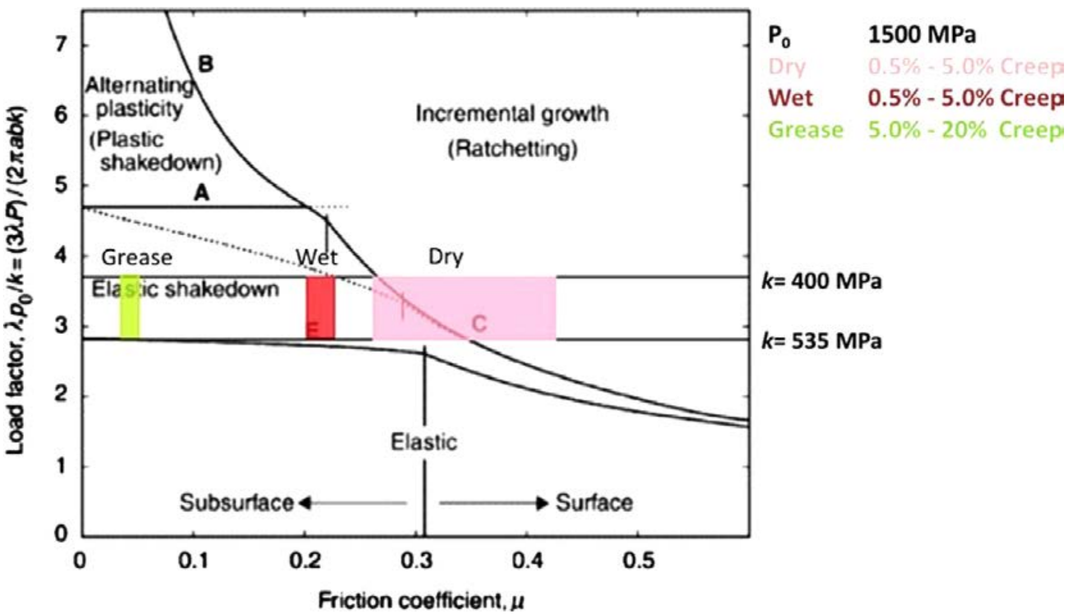



Figure 2.16: Shakedown diagram with dry, water and grease tests overlaid using actual traction coefficients and contact pressures (1500 MPa) with shear yield strength values for 260 grade rail (Hardwick, Lewis & Eadie 2014, Eadie et al. 2008)

### *2.12.2.2 Common Causes of Rail Wear in Curves*

Gauge side wear for high rails in curves is common (Turner 2008). Wear affects the life and performance of rails and wheels. Some of the influential wear parameters are: axle loads, lateral forces, longitudinal forces, creepage, curve radius, track gradient, gauge width, cant, wheel/rail factors (surface condition, temperature and material composition), rail size, train speed, train length, train frequency, train type, rolling stock performance and operating conditions. Evans (2013) reported that vehicle steering forces generated contact stresses between the high rail gauge face and the wheel flange that are so high that, without effective lubrication, rail side wear and wheel flange wear quickly reaches excessive levels and forced repairs to be made much earlier than planned.

Povilaitiene and Podagelis (2003) reported that curve radius, rail steels, rail track geometrical parameters such as rail rise and gauge width adjustment on curves have significant influence on rail side wear. If the curve radius increases from 300 to 600m, side wear intensity decreases by a factor of 2.1-3.2 and, if it is increased from 600m to 900m, side wear intensity decreases by 1.6-1.9 (Povilaitiene & Podagelis 2003). The quality of rail steel has a significant effect on the rate of rail side wear. Simple and heat treated (tempered) rails behaved markedly differently. When curve radius was within 400 to 600m, the wear intensity of simple rails is 30% larger than that of tempered rails; and when curve radius is within 800 to 1000 m, wear intensity is respectively 20% larger than that of the tempered rails. The influence of gauge width on wear intensity was also found to be significant.

According to Povilaitiene, Podagelis and Kamaitis (2006), research showed that the effective standards that regulate the gauge width increase on curves and tolerances do not assure the lowest wear intensity of the rail head on curves; standards should be specified for different curve radii to reduce wear on curves. The results of the experimental research carried out in Lithuanian railway lines have shown that widening the gauge on the curves with radius less than 650m decreases rail head side wear by up to 1.72 times. Sadeghi and Akbari (2006) reported that gauge deficiency is the most influential geometric factor which influences rail wear in tangent track and switches. They also reported that narrow gauge increases the lateral wear and widened gauge increases the vertical wear, and that regular track inspections can manage and decrease track geometrical parameter deficiencies and highly viscous lubricants can reduce vertical wear. Knothe and Liebelt (1995) reported that sliding contact with any pressure fluctuation and surface damage may cause significant effects on tribological behaviour and increased temperature at the wheel/rail interface.

Alp, Erdemir and Kumar (1996) reported on simulated tribological conditions of the wheel/rail interface in a curve to analyse lubricants and ranked them according to the performance in power consumption, friction coefficient, sliding distance and duration of lubricant breakdown. It showed that, in the early stages of sliding contacts, applied load is transmitted through the interface and/or lubricant film and causes gross sliding when the tangential stress exceeds the shear strength of the contact surface. The shear strength of the lubricant film plays an important role in the extent of sliding friction coefficient of a given system. Frictions decreases and load carrying capacity within a pair of contacting surfaces increases when lubricant is used (Alp, Erdemir & Kumar 1996). Venter (1989) and Tournay (1993) reported that, while excessive flange wear was occurring on locomotive wheels and rails due to mismatched rail and wheel profiles (two point contact), wayside lubrication resulted in an 8 to 15 fold reduction of locomotive wheel flange wear. Stehly (2008) reported a study on the Burlington Northern & Santa Fe Corporation (BNSF) railroad that found the wheel/rail interaction on tangent track without lubrication is typically 2 pounds-force (0.9 kilograms-force) per tonne of train weight and causes excessive gauge face wear in curves. BNSF found gauge face wear rates on dry rail of 6 inches (152.4 mm) per 1000 MGT and that a low to medium degree of lubrication application reduced the wear rate to less than 1 inch (25.4 mm) per 1000 MGT with negligible lateral forces compared to pre-lubrication conditions.

## 2.13 Conclusions

Research in the area of rail curve lubrication is yet to answer so many questions regarding lubricant selection, short bar versus long bar selection, lubricator location and questions related to their economic benefits. Laboratory tests regarding some of these areas do not give adequate information. Claims made by lubricant manufacturers about energy savings cannot be verified in many cases because variability in operating conditions and operators' practices may cause the marginal difference that lubricant companies claim. Comparative studies of the proper placement of applicator bars are essential. Research needs to be conducted on the design and appropriate application of applicator bars. Lubricator and applicator bar issues are significant factors in lubrication effectiveness. Therefore improvement of placement of applicator bars and dedicated maintenance of lubricators are necessary for effective lubrication. Field studies show that, even though lubrication is a state-of-the art technology and location is perfect, success depends on how you care for the unit. Seeking answers to these questions will be addressed in this thesis.

# Chapter 3

# DESIGN OF EXPERIMENT AND METHODOLOGY

## 3.1 Introduction

An experimental plan was designed and developed to evaluate the effectiveness of various methods of wayside lubrication currently being used using long applicator bars and short applicator bars. A number of rail curve greases have been compared and explanations provided as to how they differ in effectiveness under different configurations. Results from a long duration field study showed that the performance of greases under different application configurations, including variations in applicator bar placement, varies drastically. A thorough field trial/experimentation has been conducted on a live high traffic heavy haul railway track network. Collected data has been analysed and conclusions have been drawn regarding their impact on lubrication performance. This work has been also been reported in detail by the author in the CRC for Rail Innovation Project Report (CRC Australia 2014).

## 3.2 Design of Experiments (DOE)

A proper experimental design can provide most of the expected results from the comprehensive identification of the appropriate factors to be considered. Design of the experiment helps to understand the relationship between 'cause and effect' in an engineering process. It helps determine the relationships between various factors and quantify their effects on each other. It also eliminates unnecessary complexity, experimental errors, unexpected delay and budget failure for the completion.

Designs can be of various categories such as Trial and Error, One-Factor-At-a-Time (OFAT), Full Factorial, Fractional Factorial and others.

An advantage of the full factorial method is that it examines every possible combination of factors at the levels tested. Since it includes the impact of each factor influencing the outcome in a variety of combinations of these factors, the full factorial design is an experimental strategy that allows us to answer most questions completely.

The minimum number of tests required for a full factorial experiment can be derived as:

$$N = X^k \tag{3.1}$$

where N = Minimum number of tests, x = Number of levels, k = Number of factors.

This method is rigorous and time consuming. One of the disadvantages of this method is that it may not be accommodated within the budget, resources and time constraints.

In order to overcome these limitations of the full factorial method the 'Fractional Factorial' method can be applied. This approach is useful when resources are limited. It investigates a fraction of all the possible combinations contained in a full factorial experimentation. The resources necessary to complete a fractional factorial approach are significantly more manageable, economic and quicker compared to a full factorial experiment. The number of tests required for a fractional factorial experiment can be determined by:

$$N = 2_R^{k-p} \tag{3.2}$$

where, N = Number of tests, k = number of factors, $2^{k-p}$ is the number of tests, $2^{-p}$ is the size of the fraction (p=1 for 0.5 fraction and p=2 for 0.25 fraction), R = resolution.

In this study, a fractional factorial approach has been adopted and a satisfactory level of understanding of the parameters influencing lubrication has been derived. Based on these findings, recommendations have been made to maximise the wayside lubrication performance.

Heavy haul rail transportation is a high rail and wheel stress mechanism where wayside lubrication has been studied as a system as suggested by Czichos (1978). The experimental design process for wayside lubrication systems can be represented in a flow diagram as shown in Figure 3.1. The system indicates input and output parameters assuming various environment conditions such as rail surface conditions, ambient temperatures, contamination levels etc. Figure 3.1 shows that the type of greases, applicator bars (including their locations), application method, and equipment used for lubrication are the governing factors. Impacts of these factors on wayside lubrication performance are measured based on their effects on the lubrication mechanism, and those effects result in output parameters of the lubrication system such as responses related to system tribological performance, responses related to system reliability, and responses related to operation and maintenance costs.

Primarily, factors influencing the overall system performance have been identified through an analysis based on an understanding of the lubrication mechanism associated with the-wheel/rail contact surface pair. Various levels of these factors were chosen to study their influence on other factors. Thus, applying the fractional factorial method, the optimum number of experimental setups was determined.

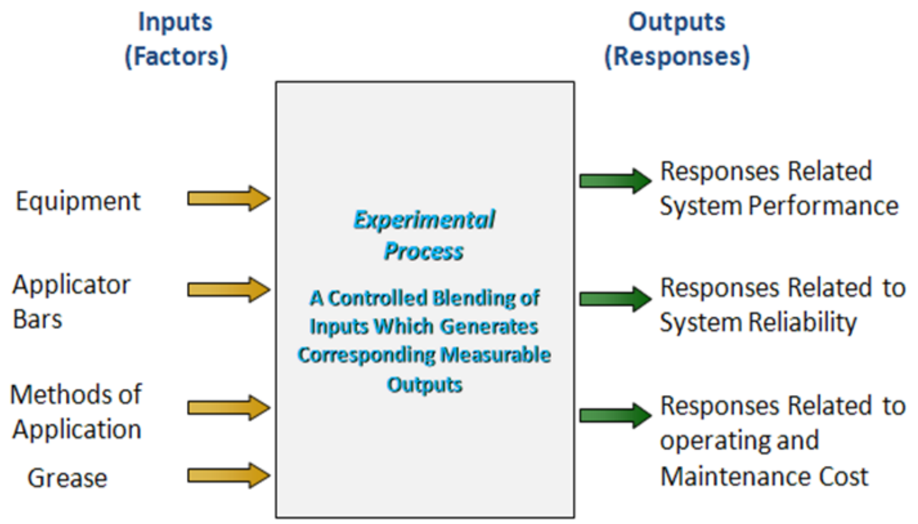



Figure 3.1: Experimental design flow diagram (Schmidt, Launsby & Kiemele 1994)

The study throws a challenge of determining how long the lubricant will last once collected from the applicator bars by the wheels during the passage of a train and how long a distance it will travel. Current practices clearly show that no method exists to predict how long the lubricant will last and how far it will travel once picked up from the lubricator.

The matrix shown in Table 3.1 shows the input parameters as a part of the experimental design for this field study.

Table 3-1: Input parameters as a part of experimental design for field study

| Equipment | Applicator Bars | Method of Application | Grease |
|---|---|---|---|
| Low Pressure System Supplier X | Short Bars | On the Transition Spiral | Grease A |
| Low Pressure System Supplier X | Long Bars | On the Tangent Track | Grease A |
| Low Pressure System Supplier X | Long Bars | On the Tangent Track | Grease C |
| Low Pressure System Supplier X | Long Bars | On the Tangent Track | Grease D |

| | | | |
|---|---|---|---|
| Low Pressure System Supplier X | Long Bars | On the Tangent Track | Grease B |
| Low Pressure System Supplier X | Long Bars | On the Tangent Track | Grease E |
| Low Pressure System Supplier Y | Long Bars | On the Tangent Track | Grease C |
| Low Pressure System Supplier Y | Long Bars | On the Transition Spiral | Grease C |

## 3.3 Methodology

A proper methodology is one of the keys to successful research. The adopted methodology plays a significant role in the outcomes and economy of real life industrial tests and demonstrations. Heavy haul railways have strong time constraints on track access. Strong supervision and effective communication is vital to undertake the track test activities safely. All activities are focused on a ‘Zero Harm” safety policy. The Australian heavy haul railway industry has a very strong Occupational Health Safety & Environment culture in place which is the reason why the long hours of tests and inspections were accident free events during this project.

This research has led to comprehensive investigations into current wayside lubrication practices and full scale field trials. From the experimental tests, a plan has been developed for appropriate equipment, grease and test location selection, equipment procurement and installation, and data collection, storage and analysis.

### 3.3.1 Wheel/Rail Lubrication

Wheel/rail interface lubrication is necessary to prolong the life of both the rail and wheels. It has much more significance in heavy haul freight operations than for light passenger train operations. Friction between the rail gauge face and wheel flanges is very high at the curves. It is desirable to apply lubricant to minimise friction and wear at the wheel/rail interface. Reduction in friction reduces both the energy losses and the wear rate of the mating surfaces, thus prolonging the life of components.

Wheel/rail lubrication is an established and accepted practice across the rail industry. Data collected from Queensland Rail (QR) heavy haul lines showed that effective lubrication could be revealed quickly by observing the progressive change in wheel flange worn surfaces as well as the rail gauge face wear rates. Typically, once lubricant is picked up by the wheel, it is distributed along the entire wheel flange surface peripherally as well as across the rail gauge face surface that comes into contact with wheel flanges during the train passage. Though the quantity of lubricant that it carries reduces progressively, it is desirable that the wheel carries lubricant as far as possible along the track.

In heavy haul railways, wayside lubrication is the most common practice in Australia and North America. However, there are differences in the type and location of the applicator bars. Australian practice is to use one or two short (approximately 600mm) bars placed on one rail only in the transition spiral of a curve. The North American practice is to use one or two long (approximately 1500mm) bars on each rail in tangent track. The location of these bars influences the lubrication effectiveness of the system and hence both methods were the focus of this study.

As a part of the experimental design phase of this project, lubricator trials on the live track have been performed. These trials are for the comparison of the effectiveness of a variety of lubricants using the Australian short bar on curve method and comparing it with the North American long bar on tangent method.

### 3.3.2 Test Plan

Significant efforts have been made to select a test site that has curved tracks and easy access without causing production losses. A number of surveys have been conducted jointly by the research team and the QR heavy haul maintenance team to identify the most relevant sites. The team decided to perform field tests on the QR North Coast Line.

#### *3.3.2.1 Test Location*

A rigorous discussion on selecting a suitable test site resulted in choosing the QR North Coast Line (NCL) between Gladstone and Rockhampton. This section of the NCL is shared with the Blackwater Coal System and carries both the NCL mixed traffic and the Blackwater Coal System unit coal trains. The proposed test location is between the 553km and the 555.5km on the Up Track of the Yarwun Bank between Callemondah Yard and Mt Larcom as shown in Figure 3.2. This section of track has two narrow gauge (1067mm) bidirectional tracks. Both tracks have 60kg rail on concrete sleepers with resilient fastenings. The maximum permissible axle load is 26.5 tonnes and the traffic includes:

- Blackwater coal trains distributed power ≈ 10 000 tonnes gross.
- NCL intermodal and unit freight trains (carrying grain, livestock, molasses).
- Loco hauled passenger trains.
- Electric tilt trains.
- Diesel tilt trains.

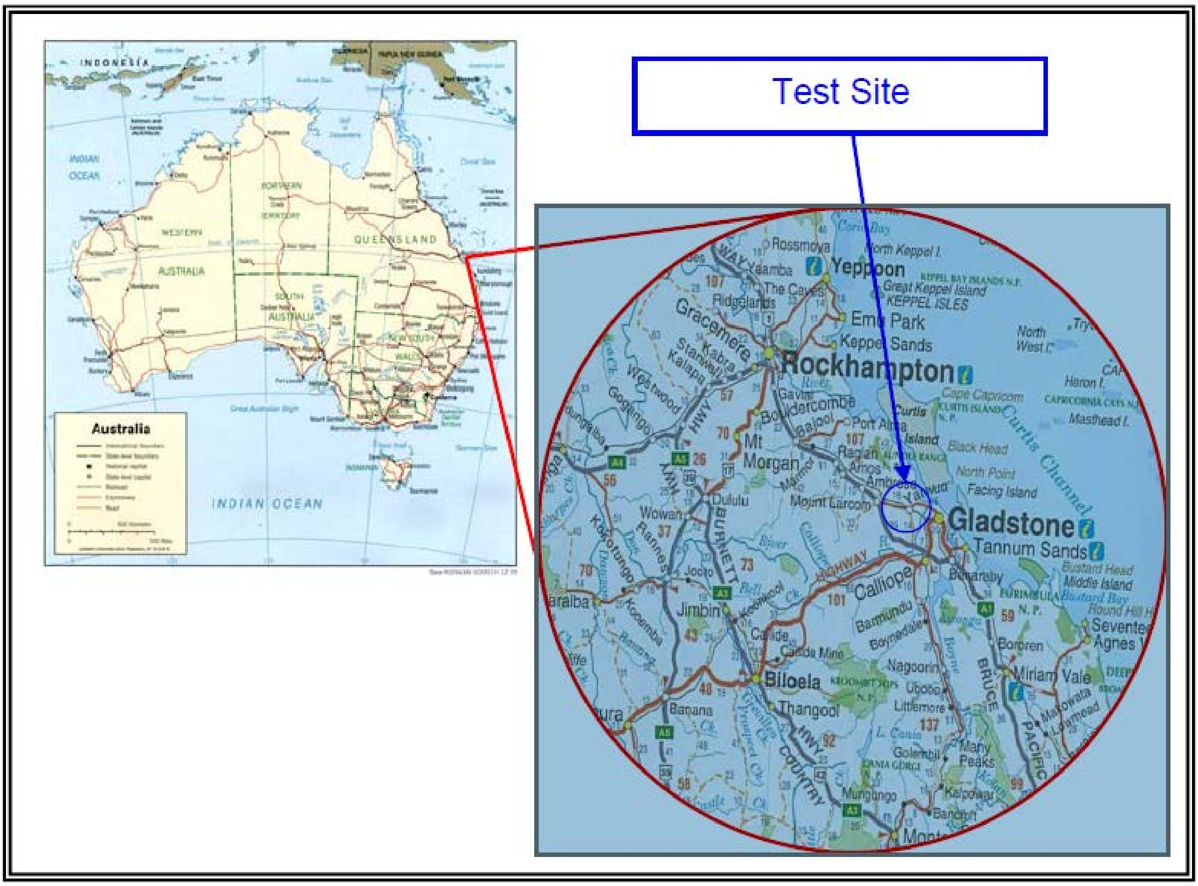



Figure 3.2: Geographical location of field test site

There are three operators on this section of track, Aurizon (formerly QR National), Pacific National and Queensland Rail (which operates long distance passenger services) and the train control is via Remote Controlled Signalling operated from Rockhampton.

There are three test sites, two on curves and one on tangent track, as follows:

A. 553.400km- This site is on the leading transition in the loaded train direction (mine end of the curve) of a 595.7m radius left hand curve.

B. 553.900km- This site is on the leading transition in the loaded train direction (mine end of the curve) of a 595.7m radius right hand curve.

C. 554.00km- This site is in a long section of tangent track.

All three sites are on the Up Track which carries the majority of the loaded coal trains as can be seen in Figure 3.3.

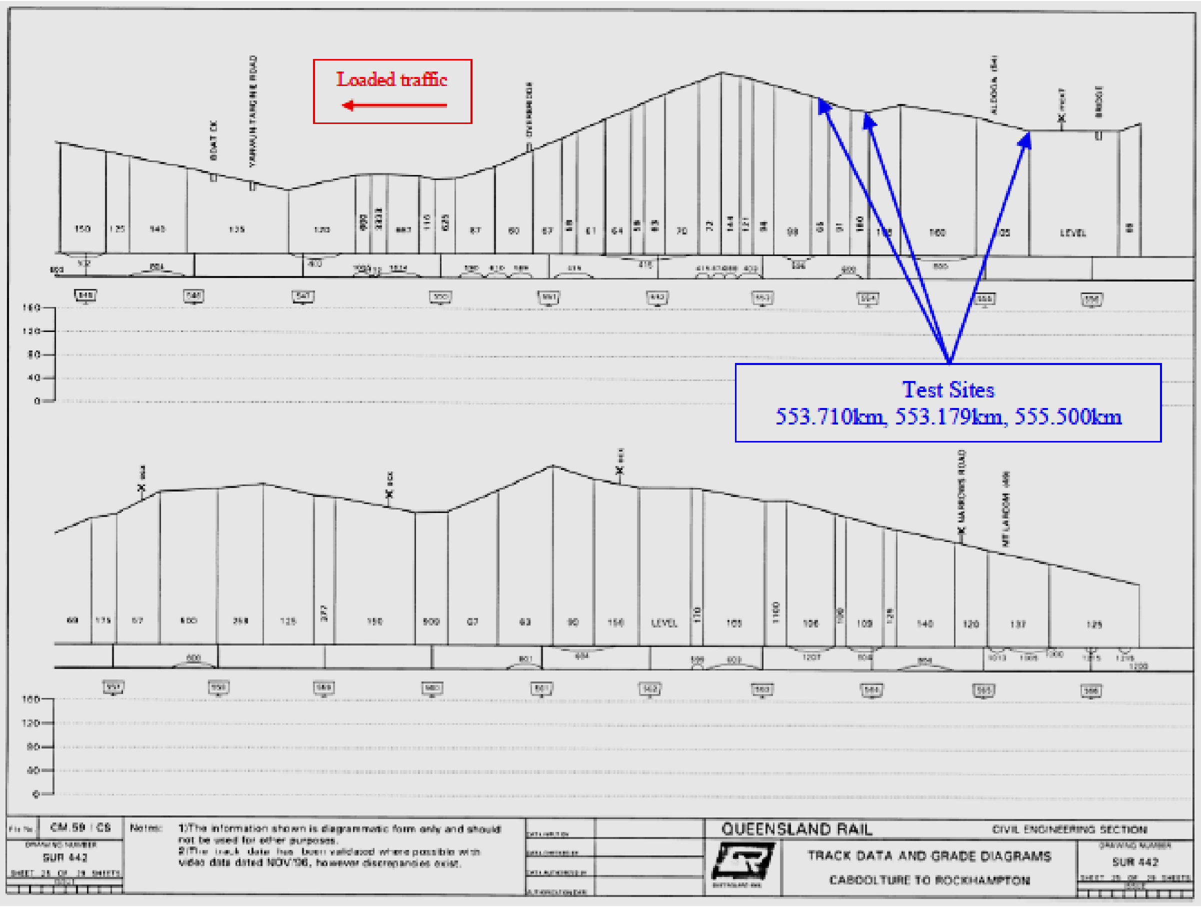



Figure 3.3: Grade and curve diagram of field trial sites and surrounding network

### *3.3.2.2 Selected Wayside Lubricators and Greases*

Latest state-of-the-art electric lubricator technology with remote condition monitoring and telemetry enhanced capability was tested on the trial sites. The trial compares long applicator bars and short applicator bars with lubrication systems from two renowned suppliers to the Australian rail industry. Three units were supplied by each supplier. High quality and high performance rail curve greases were selected from different international and national suppliers to quantify the effectiveness of lubrication application. Two short bar units were installed to lubricate both left hand and right hand curves. Two of the standard short bar units were installed for the test locations on curve transition spirals. One standard long bar unit was installed at the tangent test location as it covers both left hand and right hand curves. Five different rail curve greases, identified in this thesis as Grease A, Grease B, Grease C, Grease D and Grease E were used for the effectiveness testing.

#### 3.3.2.3 Test Procedure

Each test in this field trial adhered to the pre-defined steps sequentially. Details of activities are described below. As the tests were pre-planned, the procedure sequence has great potential to be considered as Authorised Best Practice (ABP) for wayside lubrication in the heavy haul railway industry.

***Pre-test measurements***

Essential track parameter measurements were recorded for the selected sites. These measurements were performed on unidirectional tracks during the testing period. The pre-measurement activities were as follows:

- Mark out measurement sections through the defined rail curves in between the test sites and Gladstone port. The measurement sections on each curve should be at least 50 metres in length on both rails in the body of each rail curve. Coefficient of friction data was collected using a hand pushed tribometer on the curves to estimate the surface conditions.
- Mark location of sample sites and photograph the gauge corner lubrication at these points.
- Measure rail profile of both rails at the three installation sites with a MiniProf Rail instrument and a wheel gauge as shown in Figure 3.4 (a) and (c).
- Measure track gauge as shown in Figure 3.4 (b) at the three installation sites to determine the appropriate hunting free location. Bogie hunting of wagons may cause severe damage to grease applicator bars.
- Perform dye penetrant tests on the rails in the tangent and in the test curves to evaluate the condition of Rolling Contact Fatigue (RCF) to ensure that no running surface “checking” defects were present (Figure 3.5).
- Decide appropriate location for lubricator.

(a) (b)

(c)

Figure 3.4: Onsite Measurements: (a) Precise rail profile measurement on test sites with MiniProf, (b) Track gauge measurement on test sites, (c) Wheel gauge rail profile measurement on field test sites

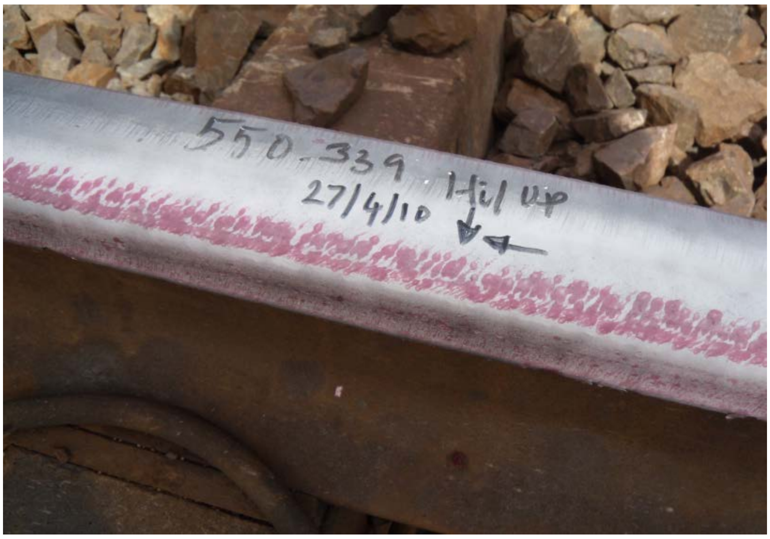



Figure 3.5: Dye penetration test on high rail gauge side

***Test execution phases***

Each test was conducted according to the pre-defined test procedure sequence. To eliminate any confusion and complexity, each activity in each phase as set out below was clarified before execution. All necessary records and documentation were maintained during all phases of the tests.

Phase 1- In the first phase, the existing lubricators around the test site were shut off for the duration of 3 days and, during this period, the following tasks were performed:

- The existing gauge face lubricators were shut down, both up-stream and down-stream.
- Note was made of any crossovers that might permit a train into or out of the test section within the measurement area.
- Train traffic was allowed to run the rail dry while the existing lubricators were shut down.
- The hand pushed tribometer was run over test measurement sections to determine the dry coefficient of friction.
- If tribometer readings were less than 0.45 on the high rail gauge corner of the two test curves, then there is another source of lubrication which must be determined and turned off.

Phase 2- The test lubricator units were installed within two days and the following tasks were subsequently performed: After installing all three units, they were tested to ensure they were working satisfactorily. Units were installed in accordance with the manufacturer's instructions, and the applicator bars on tangent track were installed using a typical QR "worn flange" wheel profile template to set the required height of the bars. Installation activities of one of the test lubricators are shown in Figure 3.6.

(a)

(b)

(c)

(d)

Figure 3.6: New electric lubricator installation on-site: (a) Placement of lubricator grease tank, (b) Preparation for the splash test after installation of applicator bars, (c) Gauge face contact measurement with worn wheel gauge, (d) Installed electronic wheel sensor

The installation activities of test lubricator shown in the Figure 3.6 can be illustrated as follows:

- Each supplier was invited to give technical assistance in the installation and testing of their units.
- Splash tests needed to be undertaken to determine the optimum lubricant pump rate for each location. Based on changeable settings of grease pump rate and frequency of wheel pass in digital control box of electric lubricator grease

application rate has been trialled and determined optimum value for different configurations of applicator bars and grease.

- When the correct height was achieved for the tangent track units, these bars were removed from the track until the actual tests were performed.
- This work would need to be undertaken while track was in use (unless a suitable shutdown was able to be availed of).

Phase 3- Curve units were turned on for the duration of 3 days and the following tasks were performed:

- The two units at 553.179km and 553.710km were turned on; these are the lubricator units set up with short bars in the curves.
- Lubricator units were run for 3 days to lubricate the track as and when trains passed through the site.
- The units were monitored through remote performance monitoring to check if there was any interruption in the units operation and grease distribution.
- It was anticipated that friction levels would settle down within 3 days as a significant number of loaded trains pass through the test site.

Phase 4- Required test parameters were recorded for 2 days following the process as given below:

- At the end of each test run using each of the greases, run the tribometer over the measurement sections to determine the coefficient of friction.
- Record the friction data and activities.
- Record any significant issue or condition change.
- Photograph spot test sites and conduct finger tests.

For the complete trial there were seven more phases covering one trial of the various combinations of applicator bars and grease. Details are not outlined because of the repetitive nature of the information. Same phases have been repeated in each test configuration of equipment, applicator bars and applied grease.

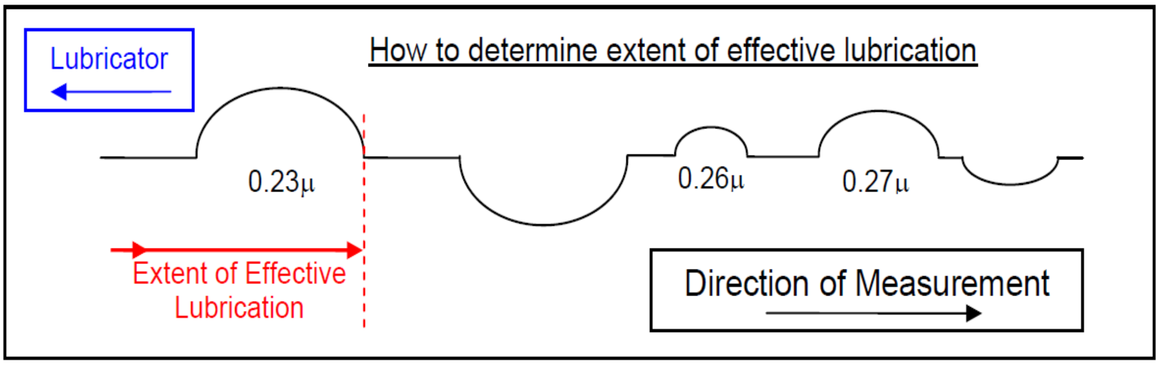
#### *3.3.2.4 Coefficient of Friction & Grease Carry Distance Measurement*

Measurement of carry distance using a tribometer is not common in Australian heavy haul rail operations compared to many other heavy haul networks around the world. It ensures a cost effective process to understand the effectiveness of any wheel/rail lubrication program. Railway operators can set up their own target value for the acceptable limit of the coefficient of friction to suit their specific operational requirements. Based on the acceptable limit of coefficient of friction, the carry distance can then be defined as the maximum distance that grease can cover while still maintaining the acceptable state of lubrication.

To measure coefficient of friction and carry distance, weather conditions and traffic movement is important. The standard conditions for taking such measurements can be laid down as follows:

- Weather must be clear and there should not be any rain.
- There should be normal traffic flow.

Special attention must be paid if there is any rail grinding facets left on the gauge corner of the rail as grease may be trapped in spherical facets and cause a localised low friction condition. Carry distance measurement (Figure 3.7) was conducted by using a hand pushed tribometer moving along the track towards the Up direction of the North Coast Line (i.e. south towards Gladstone). The field trials have been conducted on the Up Track on which loaded trains travel in the direction of the friction measurement and very rarely do empty train travels towards the mines on this track. The coefficient of friction measurements were conducted at the gauge face of the high rail in the curves, and on top of both the high and low rails in the curves. To avoid the entry and exit transition spirals of the curves, data was collected at the central 50 metres of the body of the curve. When coefficient of friction exceeded 0.25, it was presumed that the lubrication was no longer effective. Therefore, the carry distance was considered to be from the start of the curve to the point where coefficient of friction reduced to a value equal to 0.25.




(a)


(b) (c)

Figure 3.7: Measurement of grease carry distance using tribometer: (a) Method of determination of extent of effective lubrication, (b) Measurement of coefficient of friction on gauge corner by hand pushed tribometer, (c) Close-up look at measuring section of a tribometer on gauge corner (Uddin 20; Uddin et al. 2011a; CRC Australia 2014)

### 3.3.3 Test Summary, Observations and Findings

Summary records of significant field works, observations and findings during the test of Supplier X Equipment and Test Greases have been presented below in Table 3.2.

Table 3-2: Summary of significant field works, observations and findings

| Date | Description of Field Work | Observation & Imaging |
|---|---|---|
| 26/04/2010 | Long bar units installed to find the correct location and position of bars.<br>After Splash test on short bar site, bar positioning was revised due to splash and grease bead conditions. | <br>Installation of 2 long bars on each rail in tangent unit.<br><br>Splash test set up for Supplier X, long bars with grease trial. |

| | | |
|---|---|---|
| 27/04/2010 | Splash test and bar height adjustment with optimal setting of grease application rate. | <br>TOR contamination & heavy splash when bar was below 1 inch (25.4 mm) from rail head and no optimal setting.<br><br>Revised bar height at 15/16 inch (23.81 mm) below rail head and little splash observed.<br><br>Grease with optimal setting.<br><br>Dispensed grease amount measurement for optimal setting, 61 gm for 12 pump cycle. |

| | | |
|---|---|---|
| 30/04/2010 | Tribometer run and visual inspection. | <br>Gauge face had dry graphite and no grease found up to the bottom of the gauge face. Evidence of RCF on the gauge corner. |
| 03/05/2010 | Determine grease consumption. | For 57,676 axles in 7 days dispensed volume of grease 16.2 kg. 360 kg tank should last for 155 days. |
| 03/05/2010 | Visual inspection. | Found dry gauge corner with burnt graphite and flanging noise. Determined that the short bars may need to go higher but cannot lift any higher than 7/8 inch (22.22 mm). |
| 06/05/2010 | Tank topped up and rpm data monitoring and track inspection, carry distance measurement. | <br>Curve 9 had dry graphite on gauge face with very poor lubrication. Severe site contamination on ballast and TOR. Remote performance monitoring units live data supply on pump volts, ambient temperature, train time, tank product level, and wheel count. Pump cavitation detected from motor amps data. Result of carry distance was unsatisfactory with short bars Grease A. |
| 19/05/2010 | Installed the 4 long bars at 5/8 inch below top of rail. Grease application rate was 82 g per 12 pumps. Splash test set up for optimum application. | <br>Splash test with 4 long bars<br>Grease was dry on the bars. Pumping stopped during the night due to cavitation from low level in the tank. Tank topped up to 20%. |

| | | |
|---|---|---|
| 24/05/2010 | Track and lubricator site inspection. Tribometer run to measure carry distance. | <br>After 42,627 axles, considerable splash and evidence of train sanding was noticed up to 30 metres down the track.<br><br>Abrasive wear due to sanding.<br><br>Residual grease below the gauge face and dry graphite in the lower gauge corner with excessive grease had fallen off due to quick burning. Grease carry distance was unsatisfactory with long bars Grease A. |

| | | |
|---|---|---|
| 27/05/2010 | Existing lubricator site inspection. Tribometer run to measure carry distance. |  Uneven grease bead size due to airlock in the pump, ports clogging, excessive grease wastage, broken equipment, excessive maintenance need, low tank capacity, empty tank.  Low carry distance with short bars Grease A |
| 15/06/2010 | Tribometer run and track inspection after testing of Grease C. | Grease was providing good coverage on the gauge corner and the gauge face up to a distance 4.6 km. Grease was substantially carried by wheel and remained on rail gauge face. |
| 22/06/2010 | Tribometer run and track inspection after testing of Grease D. | Grease was providing good coverage up to a distance of 2.65 km, substantially further than Grease A but not as far as Grease C. |
| 29/06/2010 | Tribometer run and track inspection after testing of Grease B. | Grease was providing good coverage up to a distance of 2.96 km, substantially further than Grease A but not as far as Grease C. |

### 3.4 Conclusions

The field study covered the installation of the electric wayside lubrication equipment from two suppliers and 5 different types of lubricants to compare their effectiveness for standard test conditions. The flexibility of appropriate bar positioning at the appropriate height played a very important role in achieving transfer of grease from applicator bar delivery ports to the gauge face so that wheels could pick it up. It was observed that there was a maximum height restriction for the short bars installation which did not leave enough room for adjustment to obtain a perfect set up. The optimum delivery rate of grease was crucial to avoid severe splash and rail head contamination and to ultimately avoid sanding by trains to maintain traction. Excessive grease does not generate any extra benefit, but rather causes excessive contamination, the risk of losing train traction and undesirable environmental issues. It was shown that, if a perfect rate of grease delivery is achieved, it is possible to minimise splash, avoid sanding and effectively distribute grease to the wheels. The performance of both lubricators and applicator bars was observed to exhibit very significant differences.

The equipment from Supplier X provided very good performance with most of the greases compared to Supplier Y's equipment. Again, the applicator bar types played an enormous role in reliable grease distribution for effective lubrication. Short applicator bars on the transition spirals of the curves did not do well compared to long applicator bars in the tangent track. It was observed that the long bars in the tangent track achieved a very long carry distance compared to short bars. It was also observed that lubricants currently being used by the Australian heavy haul railway industry had performed poorly based on measured carry distance and coefficient of friction, plus survival on the gauge face. One of the scopes of the study was to analyse the effectiveness of lubricants in reducing wear in curved tracks between the high rail gauge face and wheel flanges. Observations found that Grease A disappeared from the gauge face within a very short distance or only the burnt/dry graphite residue of the grease survived on the gauge face. The grease had fallen off from the gauge face contact area within one or two curves from the test site. Other greases had varying degrees of observed success. It was observed that Grease C carried much further away from the lubricator site than other greases while maintaining the required coefficient of friction.

The optimal characteristics of lubricants are defined in the specifications. Carry distance is considered to be from the test lubricator location up to the point where the measured coefficient of friction reduces to a value of 0.25. The Lubricator Effectiveness Index (LEI) for the test lubricator units is considered to be calculated at the end of each trial and needs to be used to do a preliminary lubrication placement design for the Blackwater System. One of the remarkable findings was the significance of implementing remote performance monitoring equipment in the wayside lubricator. It has been observed that this remote condition monitoring generates so many live data which can be used to improve the reliability, maintainability, operability and cost effectiveness of the lubricators. Field investigations on the existing sites showed that the old mechanical and hydraulic lubricators did not do well in properly dispensing grease and were found to have various defects in situ. These old units are highly inconvenient for affordable maintainability and operability. Most of the time the rail was left dry due to the lack of grease delivery and severe contamination was noticed on site.

# Chapter 4

# DATA ANALYSIS AND RESULTS

## 4.1 Introduction

This chapter gives an analysis of data, collected from the field tests of installed wayside lubrication equipment supplied by two different manufacturers. Performance of this equipment has been compared for five greases supplied by different manufacturers. The study has revealed that the lubricants being used by the Australian heavy haul railway industry has poor performance based on expected grease carry distance and coefficient of friction. The study includes an investigation of the performance of several lubricants used by North American heavy haul railways also. The scope of the study was to analyse effectiveness of lubricants in reducing high rail wear in curves and wheel flange wear. The characteristics of lubricants are described by their specifications initially; subsequently, their performances as measured in field trials on the Central Queensland coal rail lines are detailed. The electric wayside lubricators have been evaluated for their optimal performance in tangent track and in the transition of the curves and a model have been developed for cost effective placement of these units along the track.  Details of the data collection procedure have been discussed earlier in Chapter 3 (DOE & Methodology). This work has also been reported in detail by the author in the CRC for Rail Innovation Project Report (CRC Australia 2014). Wheel profile data has not been collected in this research due to resource and time constraints. It was not possible to collect live data on wheel profile during the accessible track time available, collection of sample rail profile data for various curves was the essential part of the study. To quantify the effect of different field trials it would need a reasonable volume of traffic in MGT pass through the test sites which would be period of months and significant amount of track downtime needed to collect data. Therefore wear data has not been collected to quantify the effect of different lubrication field trials. If each trial would be executed for long time there would be a great opportunity to collect a representative wear data for each curves around the test sites.

The following types of data were collected during the field tests:

- Track data.
- Rail profiles, rail wear and rolling contact fatigue (RCF).
- Grease application rates or volume of dispensed grease.
- Wheel counts for various test periods.
- Coefficient of friction data measured by a hand pushed tribometer.
- Measured grease carry distance by various methods.
- Grease application intervals for different greases and equipment.
- Evaluation of remote performance monitoring data.
- Review of existing lubricators, their type and status around the sites.
- Collection of economic data for various test set-ups.

The objectives of the data analysis were:

- To determine best practices in wayside gauge face lubrication for the Australian heavy haul lines by:
    - Determining the most efficient lubricator system and its placement.
    - Determining the optimal lubricant application rates.
- To compare the effectiveness of short lubricator bar technology used in the transition spirals of curves with the long bar technology used in tangent track.
- To develop a well-established strategy for the placement location of lubricators on heavy hauls lines.
- To explore the feasibility and benefits of remote condition monitoring technology with the lubricator units.
- To perform an economic analysis on the two trial lubricator systems, short bar technology and long bar technology, compared to the existing number of lubricators on a specific heavy haul network. To meet these requirements, the following aspects have also been studied in detail:
    - The volume of grease dispensed by the total number of units required.
    - The requirement to remove and re-install the lubricators for the rail grinding program.
    - The maintenance requirements.

## 4.2 Detail Classification of Test Lubricators, Applicators & Test Configurations

The most commonly used lubricators are based on mechanical, hydraulic and electric lubricator technology. In this study, electric lubricators with various combinations of long bars and short bars have been tested in field trials. All electric lubricators and applicator bars have been supplied by two suppliers. To supress their actual names, they are identified as ‘Supplier X’ and ‘Supplier Y’ respectively. Figure 4.1 shows the details of test lubricator and applicator bar combinations tested in the field trials. Supplier X equipment was tested with all five test greases identified as Grease A, B, C, D and E respectively. Subsequently, the grease that performed the best on the supplier X equipment was tested on the Supplier Y equipment. Table 4.1 shows the various test configurations used on Supplier X and Supplier Y equipment and with which specific greases.

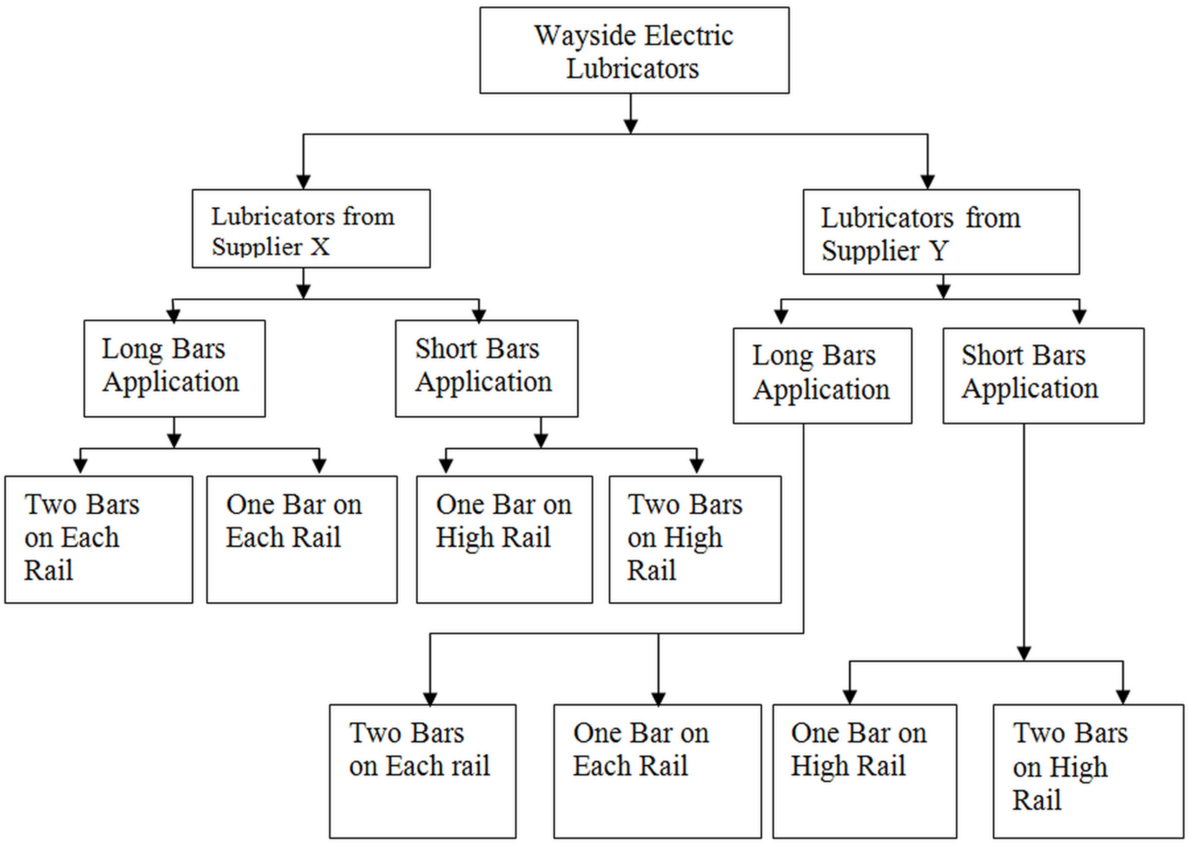



Figure 4.1: Details of test lubricator & applicator bar combinations trialled in the field tests

Table 4-1: Test configurations for electric lubricators with different applicator bars and test greases

| Configurations | Supplier | Equipment | Applicator Bars | Grease |
|---|---|---|---|---|
| 1 | Supplier X | Electric | Two short bars on high rail | Grease A |
| 2 | Supplier X | Electric | One long bar on each rail | Grease A |
| 3 | Supplier X | Electric | Two long bars on each rail | Grease A |
| 4 | Supplier X | Electric | Two long bars on each rail | Grease C |
| 5 | Supplier X | Electric | Two long bars on each rail | Grease D |
| 6 | Supplier X | Electric | Two long bars on each rail | Grease B |
| 7 | Supplier X | Electric | Two long bars on each rail | Grease E |
| 8 | Supplier X | Electric | Two short bars on high rail | Grease C |
| 10 | Supplier Y | Electric | One long bar on each rail with brush | Grease C |
| 11 | Supplier Y | Electric | Two short bars on hi-rail with brush | Grease C |
| 12 | Supplier Y | Electric | One long bar on each rail with brush | Grease C |
| 13 | Supplier Y | Electric | Two long bars on each rail with brush | Grease C |

### 4.3 Performance Measurement

Performance of lubricants can be adjudged and compared with a variety of parameters using standard ASTM tests. In this study, to be more practical, the performance of grease has been measured by two key parameters: 1) coefficient of friction under lubricated conditions, and 2) the capacity of the grease to lubricate the track length, called “grease carry distance”.

#### 4.3.1 Coefficient of Friction Measurement

The performance of grease is measured by its overall energy saving characteristics, thus coefficient of friction data on rail curves has been collected by using a hand pushed tribometer which generates digital data at the walking pace. Data points on each curve show the value of friction on the rail at progressive points around the curve. The surface conditions vary along the path and so therefore does the surface micro-geometry, thus the coefficient of friction is the average value measured within a surface area. The value of coefficient of friction within a region of a curve may either vary slowly or stay relatively constant. Further, the coefficient of friction value may vary significantly from curve to curve or it may remain stable along long distances covering several curves. Individual coefficient of friction values or tribometer readings along each curve show the change of friction within the body of the curve.

To assess the change of level of friction along the curves practically and reliably, an average coefficient of friction has been calculated for each curve which shows the overall level of friction on each curve at a known distance within the surrounding region. All the tests have been conducted on the Up Track (on which all loaded coal trains travel towards the Gladstone Port). Detailed characteristics of the track including the extent and geometric parameters of all curves are shown in Table 4.2.

Table 4-2: Rail CRC project R3-110 Curve Lists for trials

| **Rail CRC Project R3-110 Curve List for Trials: April / May 2010** | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| Note: Curves are listed in decreasing kilometrage. Curve hand (left hand or right hand) is listed when facing the direction of increasing kilometrage. | | | | | | | | | | |
| **NORTH COAST LINE** | | | | | | | | | | |
| Track | Curve No. | To | From | Radius | Cant | Speed | Transition or Cant Ramp | | Direction | Length |
| | | (km) | (km) | (m) | (mm) | (km/h) | START | END | | (m) |
| ALDOGA | | | | | | | | | | |
| UP | 1 | 554.944 | 554.301 | 804 | 60 | 90 | 40m | 40m | L | 644 |
| | | | | Tangent | | | | | | 371 |
| UP | 2 | 553.930 | 553.664 | 596 | 65 | 80 | 40m | 40m | R | 265 |
| | | | | Tangent | | | | | | 176 |
| UP | 3 | 553.488 | 553.175 | 600 | 65 | 80 | 40m | 50m | L | 313 |
| | | | | Tangent | | | | | | 157 |
| UP | 4 | 553.017 | 552.720 | 399 | 55 | 60 | CTP | 45m | R | 297 |
| UP | 5 | 552.720 | 552.613 | 384 | 55 | 60 | CTP | CTP | R | 107 |
| UP | 6 | 552.613 | 552.500 | 470 | 55 | 60 | 40m | CTP | R | 113 |
| | | | | Tangent | | | | | | 23 |
| UP | 7 | 552.477 | 552.350 | 411 | 70 | 60 | 40m | 40m | R | 127 |
| | | | | Tangent | | | | | | 40 |
| UP | 8 | 552.310 | 551.460 | 419 | 70 | 60 | 40m | 40m | L | 850 |
| | | | | Tangent | | | | | | 40 |

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| UP | 9 | 551.420 | 551.032 | 411 | 70 | 60 | 40m | 40m | R | 388 |
| | | | | Tangent | | | | | | 184 |
| UP | 10 | 550.849 | 550.629 | 585 | 65 | 80 | CTP | 70m | R | 220 |
| UP | 11 | 550.629 | 550.406 | 606 | 65 | 80 | CTP | CTP | R | 223 |
| UP | 12 | 550.406 | 550.165 | 587 | 65 | 80 | 60m | CTP | R | 241 |
| | | | | Tangent | | | | | | 2124 |
| UP | 13 | 548.041 | 547.809 | 1620 | 35 | 100 | see note | 1 in 1000 | R | 232 |
| UP | 14 | 547.809 | 547.718 | 867 | 50 | 100 | 1 in 500 | see note | R | 91 |
| | | | | Tangent | | | | | | |
| YARWUN | | | | | | | | | | |
| XOVER | 15 | 0.062 | 547.535 | 500 | 0 | 50 | - | - | R | |
| | | | | Tangent | | | | | | |
| UP | 16 | 547.564 | 547.528 | 1400 | 45 | 100 | 1 in 1000 | 1 in 1000 | R | 36 |
| | | | | Tangent | | | | | | |
| XOVER | 17 | 0.136 | 547.389 | 319 | 15 | 50 | 1 in 1000 | 1 in 1000 | R | |
| | | | | Tangent | | | | | | |
| UP | 18 | 547.339 | 546.721 | 549 | 70 | 80 | 80m | 80m | L | 618 |
| | | | | Tangent | | | | | | 795 |
| UP | 19 | 545.926 | 545.386 | 799 | 60 | 90 | 40m | 40m | R | 540 |
| | | | | Tangent | | | | | | 183 |
| UP | 20 | 545.203 | 544.759 | 546 | 70 | 80 | 40m | 40m | L | 444 |
| | | | | Tangent | | | | | | 20 |
| UP | 21 | 544.739 | 544.351 | 599 | 65 | 80 | 40m | 40m | R | 389 |
| | | | | Tangent | | | | | | 51 |
| UP | 22 | 544.299 | 543.921 | 808 | 60 | 90 | CTP | 40m | L | 378 |

| | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|
| UP | 23 | 543.921 | 543.856 | 934 | 60 | 90 | CTP | CTP | L | 65 |
| UP | 24 | 543.856 | 543.621 | 741 | 60 | 90 | 40m | CTP | L | 235 |
| | | | | Tangent | | | | | | 87 |
| UP | 25 | 543.534 | 543.091 | 808 | 60 | 90 | 40m | 45m | L | 443 |
| | | | | Tangent | | | | | | 45 |
| UP | 26 | 543.046 | 542.624 | 800 | 60 | 90 | 40m | 40m | R | 422 |
| | | | | Tangent | | | | | | 1038 |
| UP | 27 | 541.586 | 541.377 | 604 | 65 | 80 | CTP | 40m | L | 209 |
| UP | 28 | 541.377 | 541.164 | 608 | 65 | 80 | 40m | CTP | L | 213 |
| | | | | Tangent | | | | | | 198 |
| UP | 29 | 540.965 | 540.682 | 773 | 60 | 90 | 40m | 60m | R | 283 |
| | | | | Tangent | | | | | | 865 |
| UP | 30 | 539.817 | 539.569 | 2186 | 40 | 120 | see note | 80m | R | 248 |
| | | | | Tangent | | | | | | 31 |
| UP | 31 | 539.538 | 539.311 | 1956 | 40 | 120 | 80m | see note | R | 227 |
| | | | | Tangent | | | | | | 421 |
| UP | 32 | 538.890 | 537.736 | 890 | 65 | 100 | 80m | 80m | R | 1154 |

The following data were collected for each test:

- Title of the Project- Rail lubrication test project-Mt. Larcom.
- Date of the test.
- QR Curve Number.
- Start point of curve and end point of curve as per kilometre post.
- Measurment direction towards the Gladstone Port.
- Point distance in m.
- Time of the test.
- Ambient temperature in degrees Celsius.
- Relative humidity (%).
- Rail (Low rail/high rail), Low rail is the rail on the inside of the curve and high rail in the rail on the outside of the curve.
- Rail Conditon: Dry (all surrounding existing lubricators & test lubricators off), Wet/lubricated (all surrounding existing lubricators off and only test lubricator on).
- Position of data collection (Gauge face/ Top-of-Rail).

To maintain the highest integrity and authenticity of the data, similar test conditions, track conditions, traffic conditions and surrounding ambient conditions have been taken into account. Data collection was not conducted if any of following events occurred:

- Any rain event and visible water particles or dew on the rail.
- Any grinding event occurred. Grinding cycles were aligned with the lubrication tests so that the track condition was maintained in a normal condition.
- If any high rail vehicles (four wheel drive road vehicles with adapted wheelsets that can travel on the rail track for inspection purposes) pass through the test sites. In this case data collection was suspended until after a few loaded trains passed through the site and the grease distribution on the rail attained equilibrium.
- While there was excessive splash along the test site and severe sanding was conducted by trains to to maintain adequate traction.
- Any technical failure of the lubricator or applicator bars such as running out of grease, cavition of pumps, damage to applicator bars, etc.
- A negligible number of trains passed through the test site and there was no established grease distribution on the curves.

- If any curve is inaccessible due to geographical location or any other uncontrolled situation.

Locations of the three test sites are as follows:

- Site-1 location at 553.440km- on the leading transition (mine end of the curve) of a 595.7m radius left hand curve (as defined by note in Table 4.2).
- Site-2 location at 553.908km- on the leading transition (mine end of the curve) of a 595.7m radius right hand curve.
- Site-3 location at 554.00km- on a long section of tangent track.

The trend of changes of coefficient of friction and the distance covered over which the expected level of friction was found to be sustained are recorded in this field study.

Accepted target values of Coefficient of Friction (COF) on the gauge face of the high rail, the top of the high rail and the top of the low rail in the rail industry are shown in Figure 4.2.

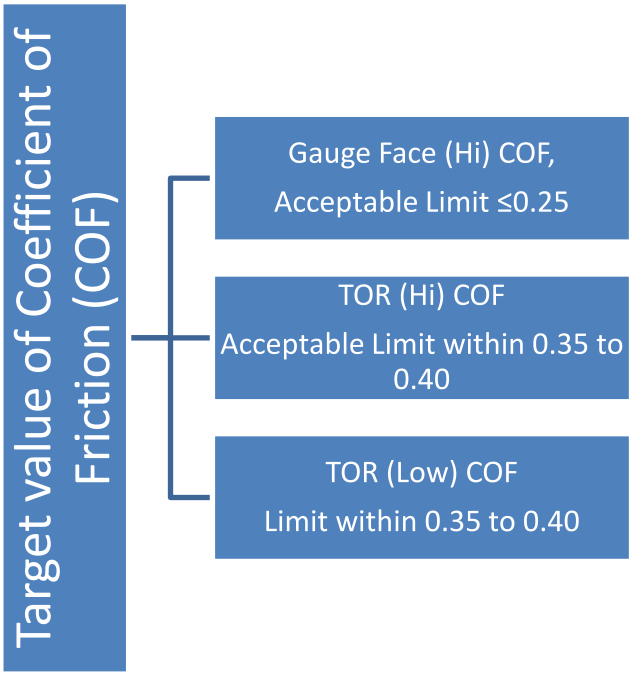



Figure 4.2: Typical target coefficient of friction values on areas of wheel/rail contact (Sroba et al. 2010)

## 4.4 Data Analysis

Equipment from Suppliers X and Y (lubricators and applicator bars) and five different greases were used in these field tests. All standard sets of applicators bars from both suppliers have been tested with different greases as indicated in Table 4.1. Though all available standard options of applicator bars have been tested with the existing "standard" grease

(Grease A), the rest of the greases have been tested with either two long bars on each rail (for tangent track location) or two short bars on high rail (for curve transition location). Experimental error and uncertainty analysis has been studied for coefficient of friction data for different trials. No physical quantity can be measured with perfect certainty. Data has been collected by using electronic instrument hand pushed tribometer to avoid manual data collection and gain greater confidence to confirm that our data are very close to the true value. Before data collection the probable source and type of errors, clearly and correctly recorded data and uncertainties in the data, and development of experimental plan have been considered to ensure high accuracy of data and reduce the experimental errors. The experimental error measured here through error analysis. If the independent variable are $x_1$, $x_2$, $\ldots x_n$ are combined to have y by the relation,

$y = y(x_1, x_2 \ldots x_n)$

Then the error combining rule is

$$\Delta y^2 = \sum_{i=1}^{n} \left(\frac{\delta y}{\delta x_i} \Delta x_i\right)^2$$

The partial derivative of y with respect to $x_i$ provides the sensitivity of the result to the particular variable. The sum of squares of the experimental error, SS(exp) can be approximated as

$$SS(\exp) = \sum_{i=1}^{n} (\delta N_{i,obs})^2$$

Where $\delta N_i$ is the estimate of the uncertainty in $N_{i,obs}$. The sum squares of the experimental error depend on the number of observations.

This research has confirmed the most effective configuration of relevant lubricator, type of applicator bars and grease from all the different configurations that have been tested in the trials.

### 4.4.1 Configuration 1: Supplier X Grease A with 2 SB-HR (2 Short Bars on High Rail), Test condition: All Lubricators Offline (Track Condition- Dry)

To avoid any influence of ongoing lubrication, prior to commencing the trials it had been ensured that the track was completely dry and had no trace of lubricant on the rails. All the lubricators around the test curves had been turned off and data collected to prove the complete dryness of the curves. Similarly, before every subsequent configuration test, complete dryness of the curves had been confirmed by data collection. If the tribometer data showed any level of lubrication existed, the surrounding lubricators continued to be shut off for an extended period and measurements were carried out to ensure the dry condition of rails. Collected data from the first curve to the furthest curve showed the coefficient of friction on the gauge face and top-of-rail was very high and there was no sign of lubrication at all. Though all the coefficient of friction data in each of the curves (Curve 2, Curve 3, Curve 5, Curve 8, Curve 12) are too high, Curve 5 from lubricator site is considered to be the representative curve, and Figure 4.3 shows its COF trends.

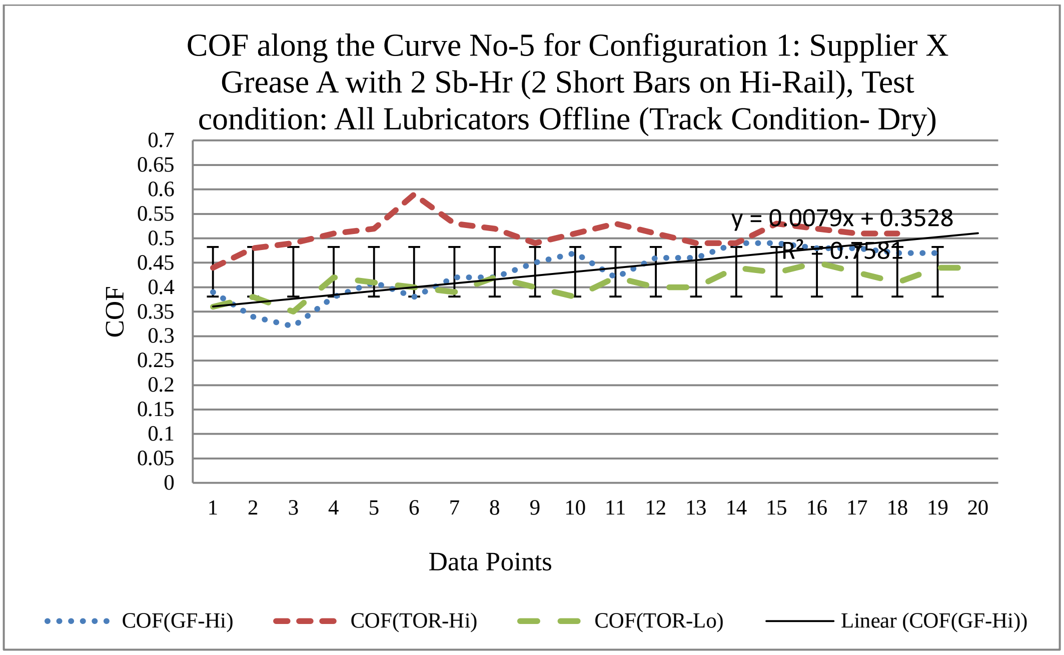



Figure 4.3: COF data distribution along Curve No-5 for configuration 1 while the track condition is dry

In Figure 4.3, collected tribometer data along the Curve No-5 indicated that the track was completely dry and the rail steel at gauge corner was observed to be completely shiny. Another qualitative method for testing lubricity called the “finger test” was also conducted by rubbing the finger on the surface of the rail curve; no grease or stickiness was observed on the finger, neither was there any change of colour or sign of heat burn (burnishing, scuffing) observed in the contact area. After the passage of trains over the dry rail curves, all remnants of the lubricant were burnt due to high heat generated due to friction and there was clear evidence of flanging with a squealing noise from wheel/rail contact being heard from a great distance. The tribometer data collection revealed that:

- The gauge face of the high rail was completely dry as the COF values were 0.35 and above.
- The standard deviation of the COF values indicate the data was close to the average COF for the curve and liner trend shows it has $R^2$ over 0.75 which strongly claims it linear trend.
- The TOR of the high rail was completely dry as the COF values were 0.4 and above.
- The TOR of the low rail was completely dry as the COF values were 0.4 and above.
- Both high and low rails were completely dry.

### 4.4.2 Configuration 1: Supplier X Grease A with 2 SB-HR (2 Short Bars on High Rail), Test Condition: Only Test Lubricators Online (Track Condition- Lubricated)

For all configurations with short bars on each rail, two lubricators were in place, one to lubricate left hand curves and another lubricator to lubricate right hand curves. The COF data was collected by hand pushed tribometer while the test lubricators were online (i.e. in operation) and were distributing Grease A. Data was collected on the body of the different curves. Tribometer data showed that, except for the COF on the gauge face of Curve 2, all the data exceeded the limits and showed the curves are either dry or starving for grease from very ineffective lubricant application. Also the data showed the top of high rail coefficient of friction is significantly low (too greasy below 0.30) for traction and this presents excessive risk to train movement and results in excessive sanding. TOR data shows the rail is too dry to operate and the TOR COF value reaches up to 0.65. Grease A disappeared very quickly from the gauge face and moved toward both the top-of-rail head (caused excessive contamination) and the bottom of the gauge face (caused excessive loss of grease from the gauge face contact area). Figure 4.4 shows the trends of COF data on Curve 2. Curves further along the track had ineffective lubrication as the gauge face COF values were higher than 0.25 and this indicates that this configuration limits the effective lubrication to within only one curve.

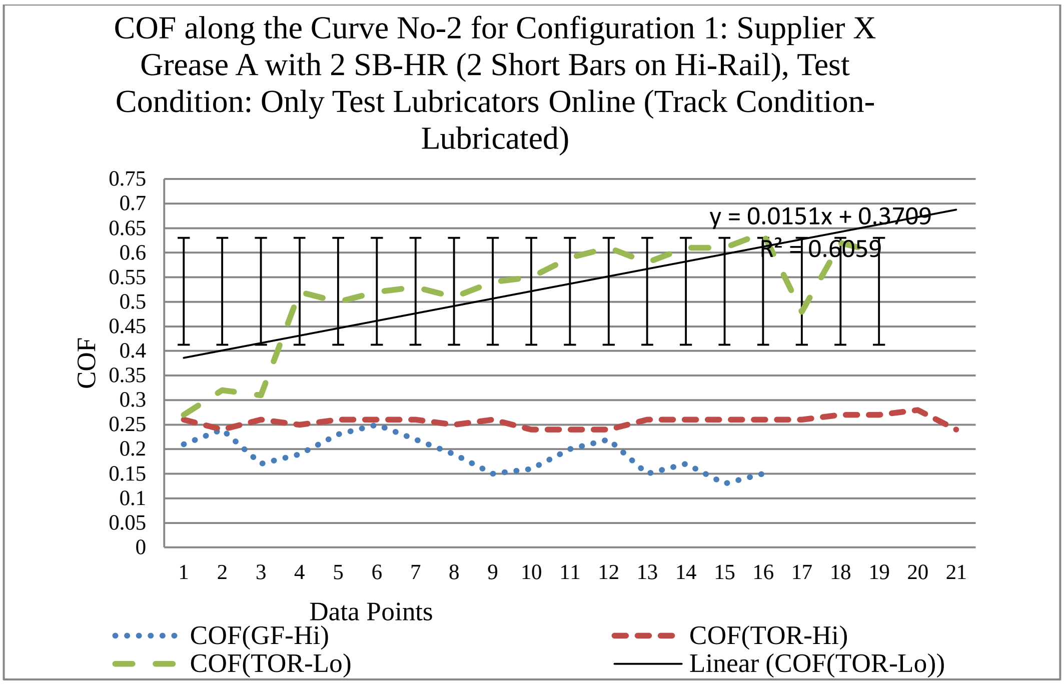



Figure 4.4: COF data distribution along Curve No-2 for configuration 1 while the track condition is lubricated with online test lubricator

This demonstrates the overall condition of the curve did not have a stable gauge face coefficient of friction varying around the expected value of (effective gauge face lubrication) less than or equal to 0.25. It confirms that the gauge face lubrication was effective only for Curve 2. The top of the high rail was highly contaminated and the top of the low rail was dry.

Data has been collected on the body of Curves 5, 6, 8, and 9 and found even worse conditions of lubrication. Hence this test configuration achieved only one curve with effective lubrication and the rest of the curves remained unlubricated. Figure 4.5 shows all the COF data collected on various curves and shows that the COF at the gauge face rises above the target value of 0.25 within a very few measurements within the first curve. There is no steady trend in the data points.  The rest of the data points are above 0.25 which shows lubrication was no longer effective. There is a similar trend in top of high rail and top of low rail data. When the GF COF goes up, TOR data goes up as well. That may be indicative of a loss of grease from the GF and an initially dry TOR, with some grease from the GF then travelling towards the TOR. TOR contamination must be avoided to maintain necessary traction and train braking capacity. In the case of TOR contamination, excessive sanding is applied by train drivers to achieve required traction even though it causes severe wear. The TOR data for the high rail shows a COF mostly below 0.35 which indicates some degree of TOR contamination occurring during the test. TOR data for lo rail shows that the standard deviation is significantly high and $R^2$ value is around 0.60 which does not strongly support its linear trend as the data values have significant variation from average or mean.

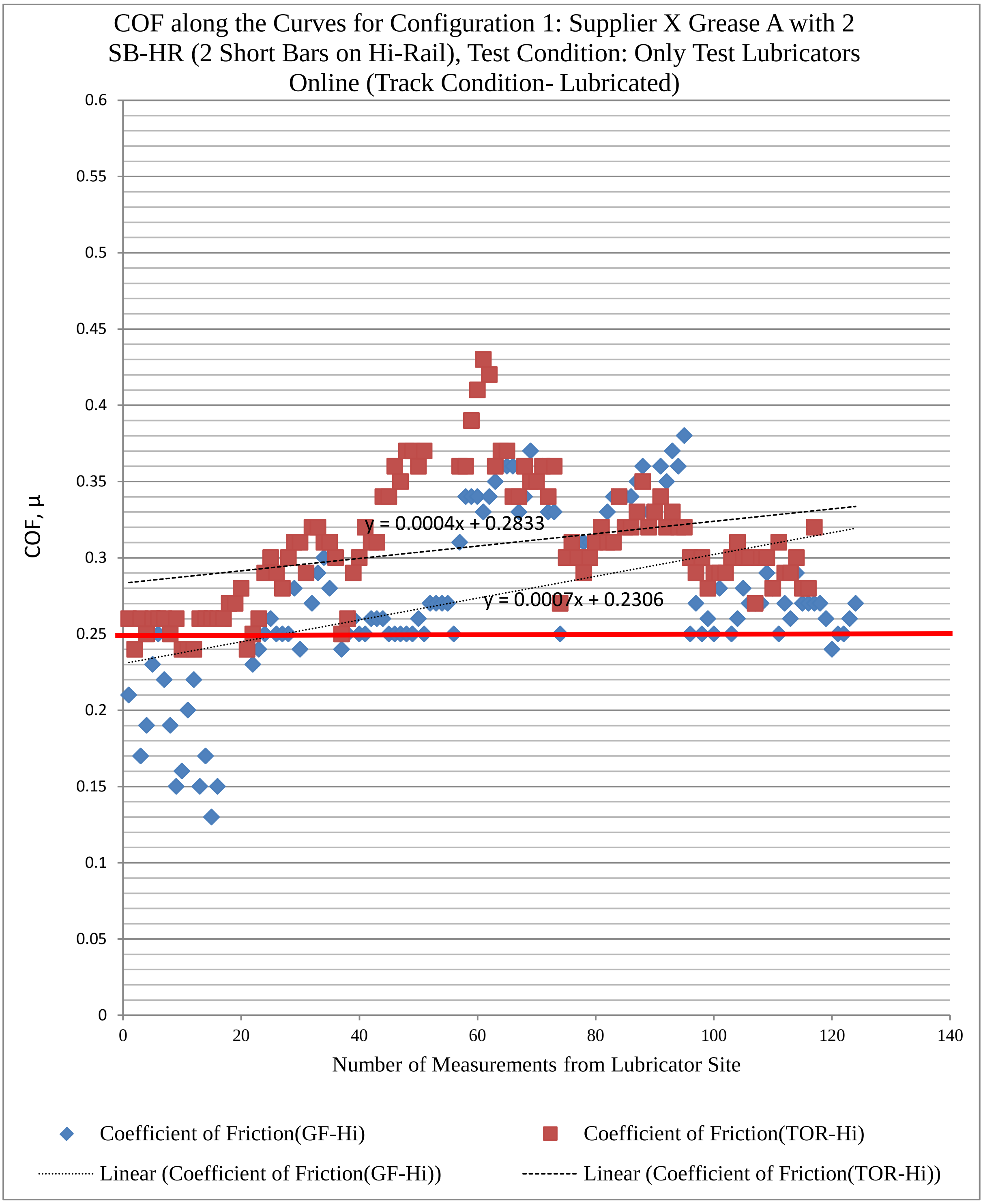



Figure 4.5: COF data distribution for configuration 1 while the track condition is lubricated with online test lubricator

### 4.4.3 Configuration 2: Supplier X Grease A with 1LB-ER (1 Long Bar on Each Rail), Test Condition: Only Test Lubricators Online (Track Condition- Lubricated)

Figure 4.6 displays the COF data for Curve 4, the last curve having stable GF COF values below 0.25 which confirms the gauge face lubrication was within the expected level. On the other hand, due to the excessive grease movement upwards from the gauge face, the TOR COF values were lower than the recommended range of 0.35 to 0.40 which is not acceptable as it may not generate enough traction/braking.

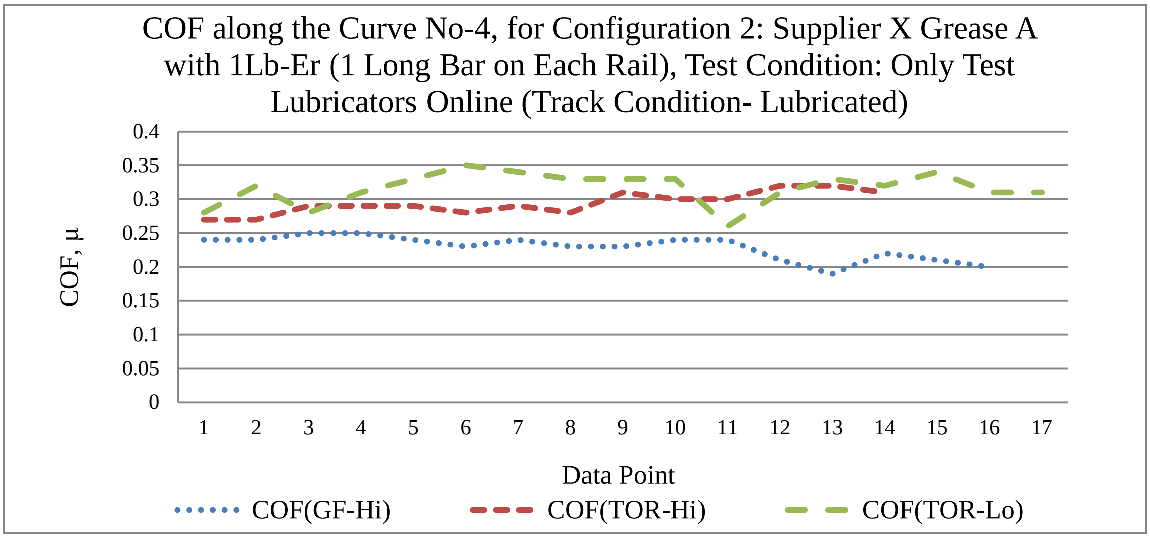



Figure 4.6: COF data distribution along Curve No- 4 for configuration 2 while the track condition is lubricated with online test lubricator

### 4.4.4 Configuration 3: Grease A with 2 LB-ER (2 Long Bars on Each Rail), Test Condition: Test Lubricator Online (Track Condition- Lubricated)

For configuration 3, only one curve (Curve 2) has been achieved the expected GF COF values, indicating that the carry distace only reached up to Curve 2 which is a very short distance. Further curves had ineffective lubrication. Figure 4.7 demonstrates and supports this conclusion and shows all the data points exceed the expected COF value of 0.25 and went much above it. There is no steady trend in the data points. Grease has significantly splashed above the whole rail head at the lubricator site and first curve. Then grease has been completely burnt out within the second curve. No grease survived in the GF contact area and was found to have fallen off the rail head as burnt lumps of grease.

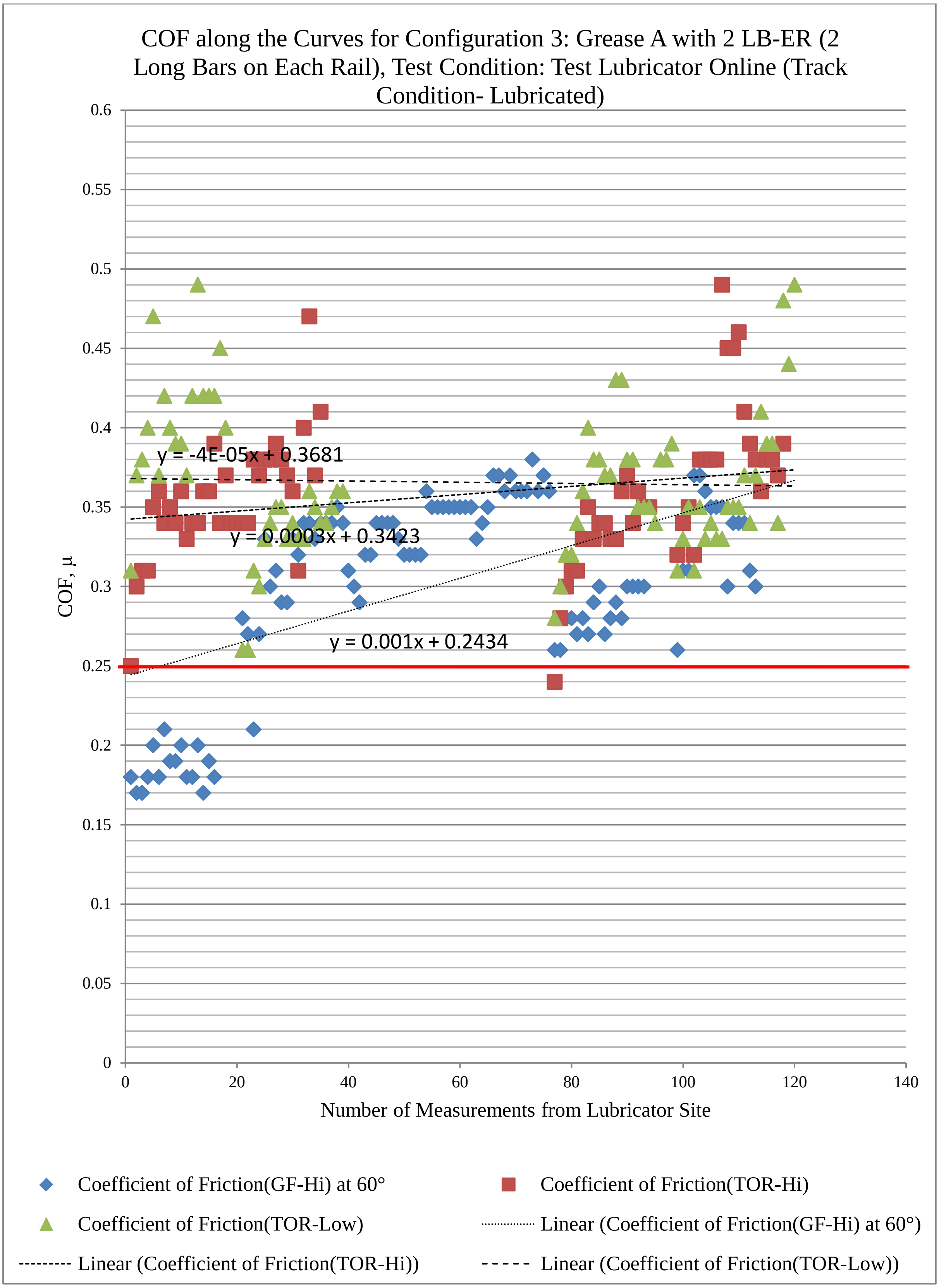



Figure 4.7: COF data distribution along the data collecting curves for configuration 3 while the track condition is lubricated with online test lubricator

### 4.4.5 Configuration 4: Grease C with 2 LB-ER (Supplier X), Test Condition: Test Lubricator Online (Track Condition- Lubricated)

Configuration 4 achieved effective rail curve lubrication. Grease C with 2 long bars on each rail in the tangent track distributed the grease up to Curve 14 which is 4.6km away from the test lubricator site. A total of 14 curves (both left hand and right hand) has been effectively lubricated by this configuration. The COF values in each curve show the established level of friction and confirms the grease has remained in the expected location on the GF. Grease C did not move to the rail head as many other greases did. It remained in the contact band of the GF where it has designed to be. Therefore is shown by the fact there is no significant TOR drop of COF. Figures 4.8 and 4.9 show the COF values in Curves No-2 and Curve No-14 respectively, and found that grease was sustained around the GF contact area and did not cause TOR contamination.

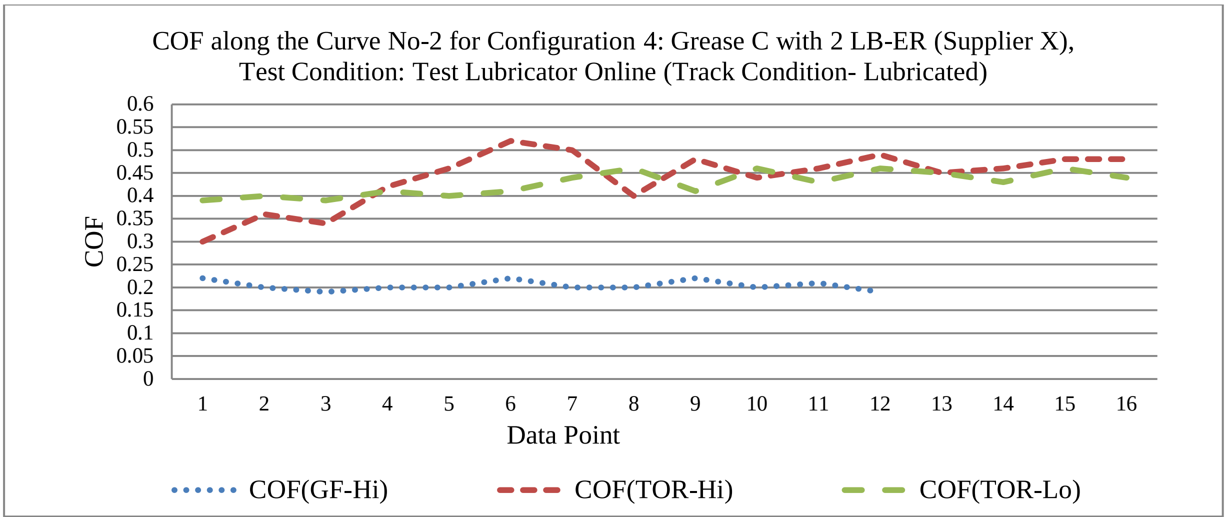



Figure 4.8: COF data distribution along Curve No-2 for configuration 4 while the track condition is lubricated with online test lubricator

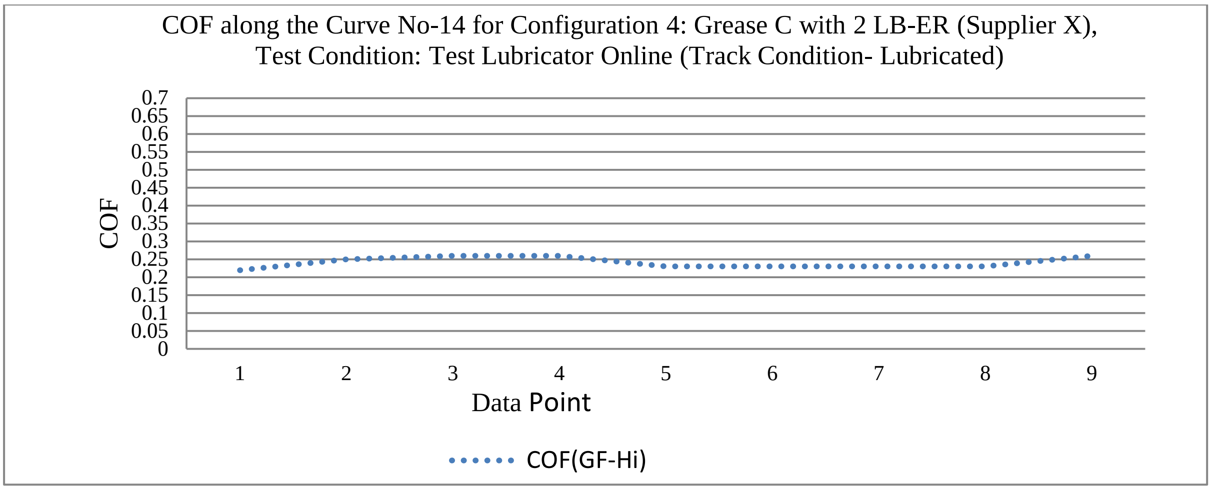



Figure 4.9: COF data distribution along Curve No-14 for configuration 4 while the track condition is lubricated with online test lubricator

The data in Figure 4.10 shows that (with some minor exceptions discussed below) all of the GF and TOR COFs were maintained within the relevant target values by this most effective lubrication at the farthest distance. Investigation showed that grease had not been splashed onto the TOR at the lubricator site and along the curves. The chemical and mechanical stability of Grease C gave it enough strength to remain on the gauge face from the application site and into the subsequent curves. No symptoms of burnt grease falling off from the GF were noticed. Interestingly, as the GF COF values increased slowly, the high rail TOR COF values decreased slowly. All the values were within their target ranges for TOR except for some low rail TOR values. Given the properties of Grease C and the field test results, it can reasonably be concluded that, Grease C existed at the gauge face contact area after multiple loaded trains passed through the line for nearly a week, in part presumably due to its structural strength and tackiness with EP additives. The grease settled down firmly at the gauge face contact area and may have delivered a little trace of grease towards the high rail TOR and helped to reduce the COF there.

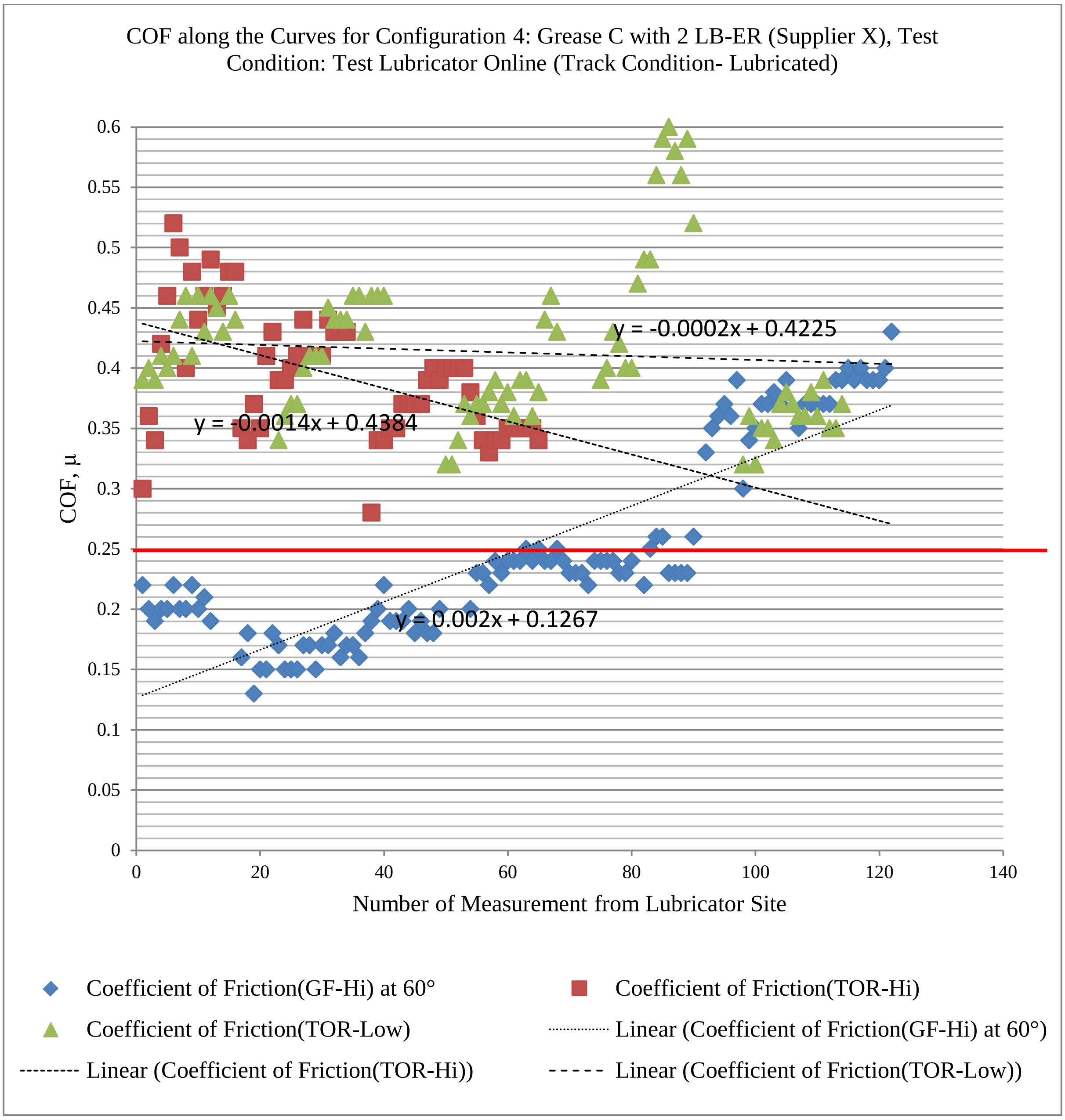



Figure 4.10: COF data distribution along the data collecting curves for configuration 4 while the track condition is lubricated with online test lubricator

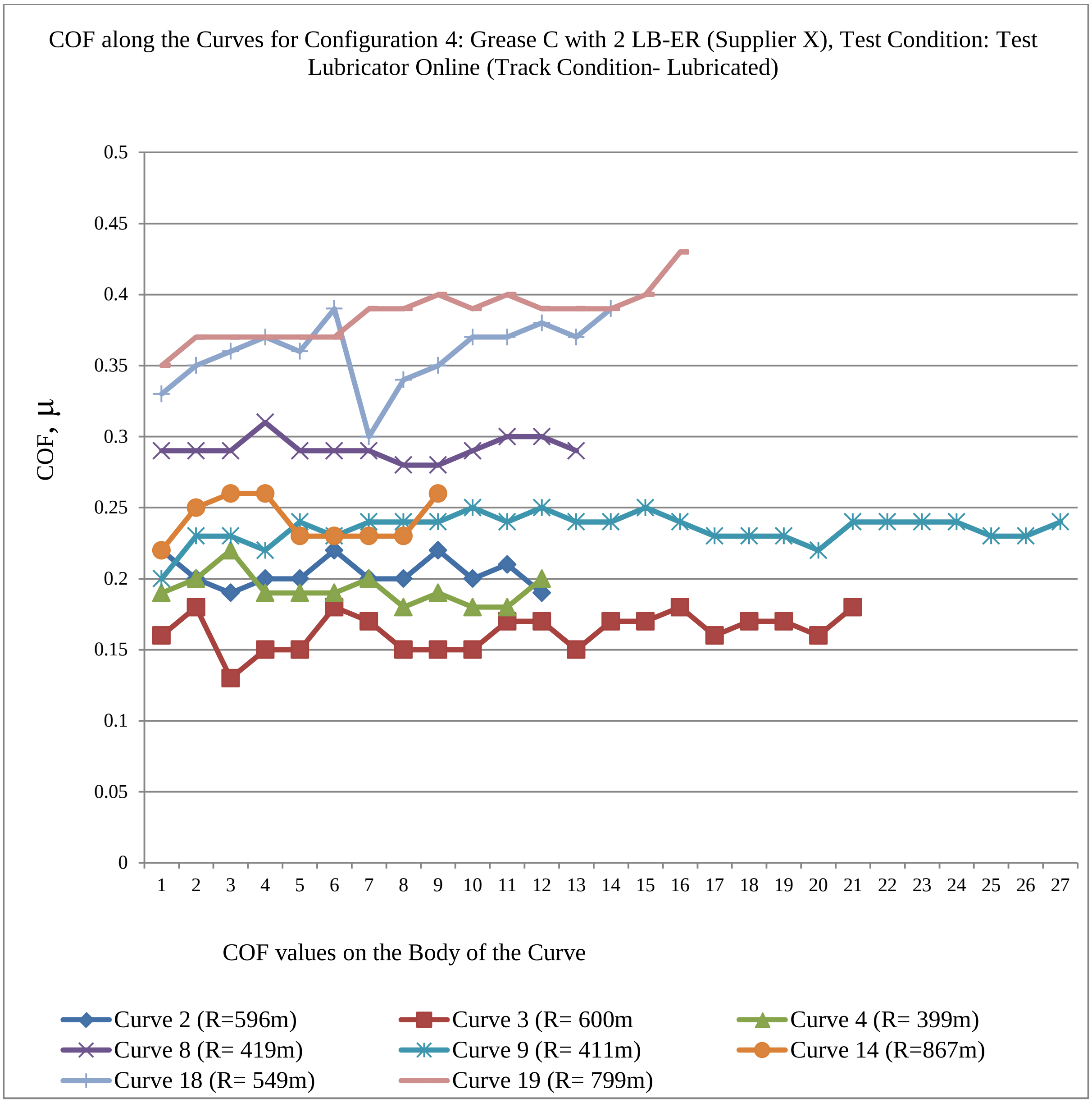



Figure 4.11: GF COF values on body of different curves with 2 long bars (Supplier X) on each rail and Grease C

Figure 4.11 shows only the GF COF values in various curves; the level of friction was sustained below 0.25 up to Curve 14. Only Curves 18 and 19 have a rise of COF values above 0.25. Grease C with 2 long bars (Supplier X) on each rail has the longest grease carry distance of all of the configurations in the field trials. Even after Curve 14, there is no sharp rise in COF; rather, it increased relatively slowly and smoothly to as high as 0.4. Grease C was observed to provide an established layer of lubricant on each curve with no sign of burnt grease anywhere on the rail foot.

### 4.4.6 Configuration 5: Grease D with 2 LB-ER (Supplier X), Test Condition: Test Lubricator Online (Track Condition- Lubricated)

Configuration 5 achieved seven curves with effective lubrication. Figure 4.12 presents the data for the last curve (Curve 7) with a COF below 0.25. Investigation showed that the grease provided good GF coverage with no grease contamination of the TOR as occurred with the existing Grease A. The total carry distance achieved was 2.64km.

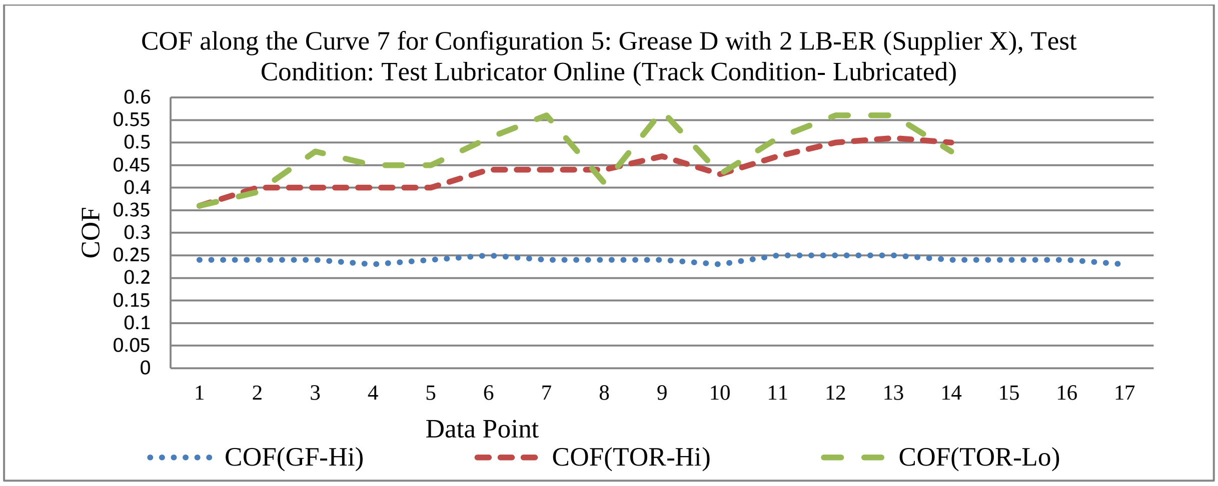



Figure 4.12: COF data distribution along Curve 7 for configuration 5 while the track condition is lubricated with online test lubricator

Figure 4.13 below shows the total distribution of the COF values collected for this configuration. It shows that, although a few data points went above the limit of 0.25 it returns below 0.25 very quickly and maintained the limit for a while. The TOR of both rails was either dry or within the expected range which confirms the established grease cover on rail GF. Grease was not splashed above the TOR at the lubricator site or along the test curves. The chemical and mechanical stability of the grease gave it enough strength to remain on the GF at the application site and along the subsequent curves. No symptoms of burnt grease falling off from the GF were noticed.

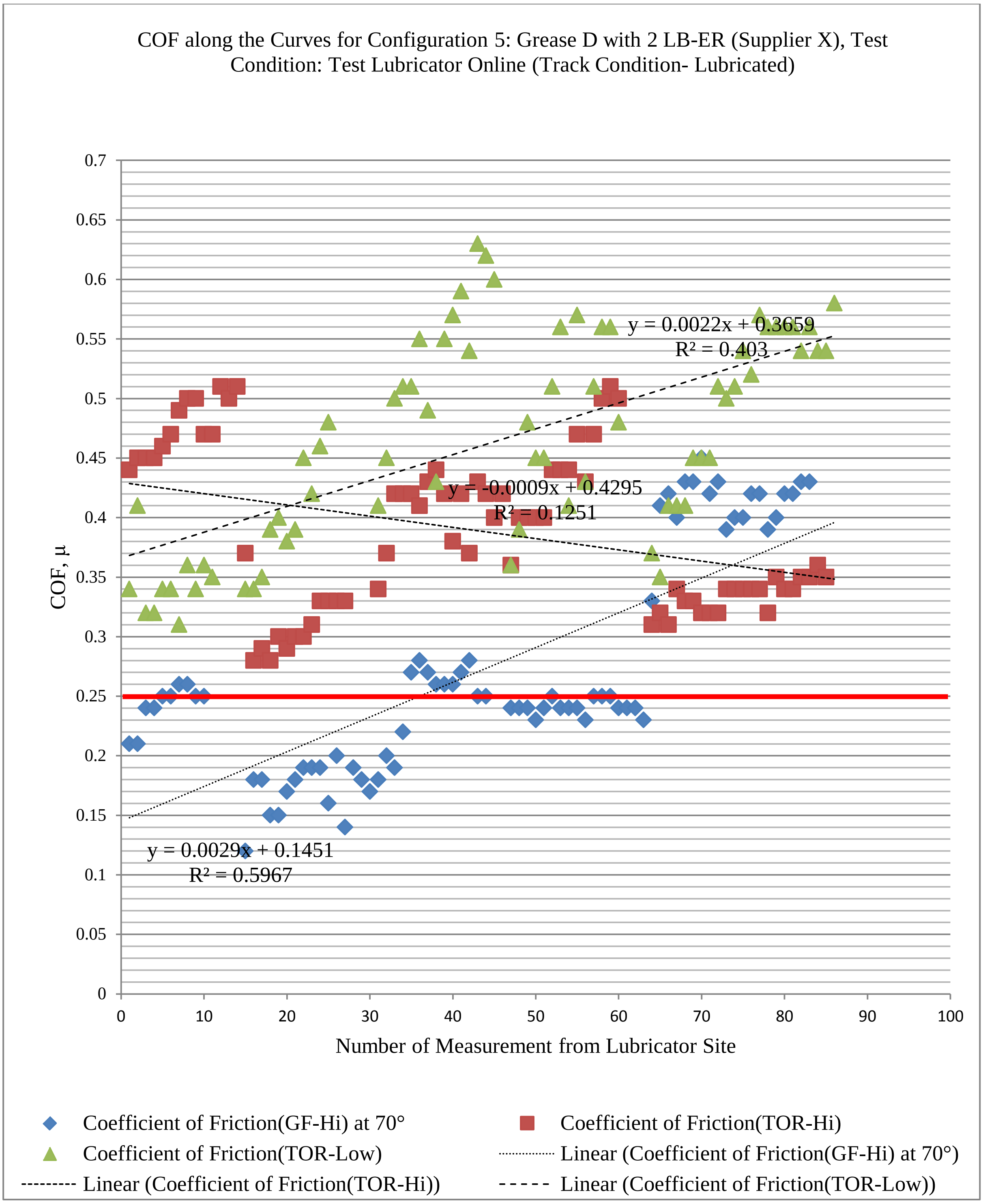



Figure 4.13: COF data distribution along the data collecting curves for configuration 5 while the track condition is lubricated with online test lubricator

### 4.4.7 Configuration 6: Grease B with 2 LB-ER, Test Condition: Test Lubricator Online (Rail Condition- Lubricated)

Figures 4.14, 4.15 and 4.16 show that configuration 6 (Grease B with 2 long bars on each rail from Supplier X) achieved effective GF lubrication from Curve 2 to Curve 9. This section of the test site showed a steady and established level of COF below 0.25. The TOR COFs were above the relevant target values, indicating no movement of grease from the GF towards the TOR.  Investigation shows that grease was not splashed above the TOR at the lubricator site or along the test curves. The chemical and mechanical stability of the grease gave it enough strength to remain on the GF at the application site and in the subsequent curves. No symptoms of burnt grease falling off from the GF were noticed.

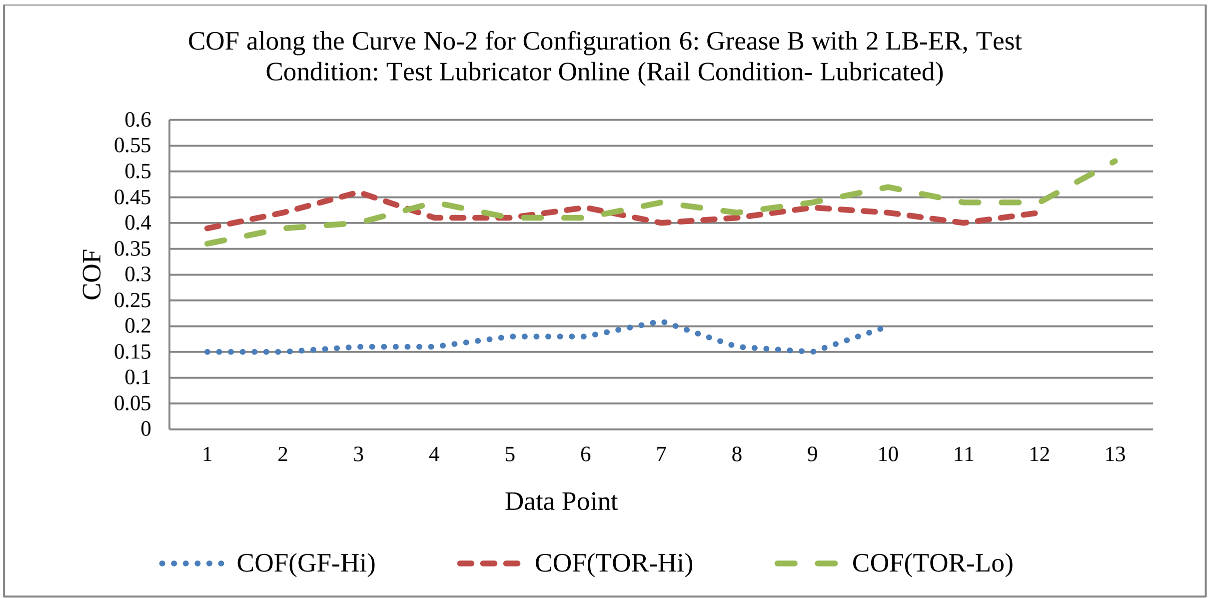



Figure 4.14: COF data distribution along Curve No-2 for configuration 6 while the track condition is lubricated with online test lubricator

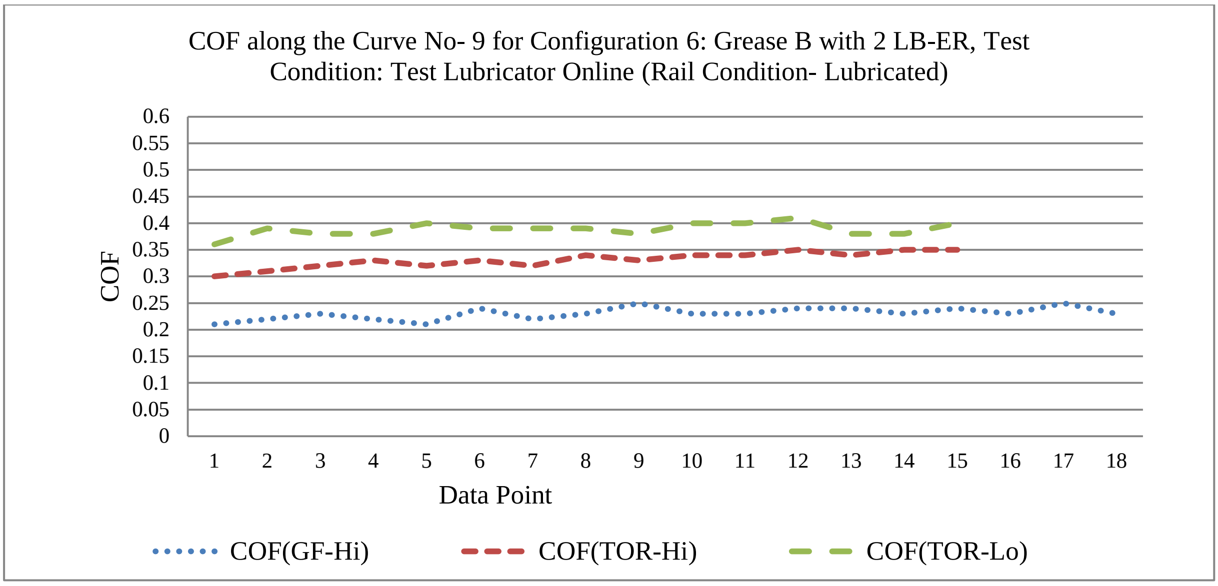



Figure 4.15: COF data distribution along Curve No-9 for configuration 6 while the track condition is lubricated with online test lubricator

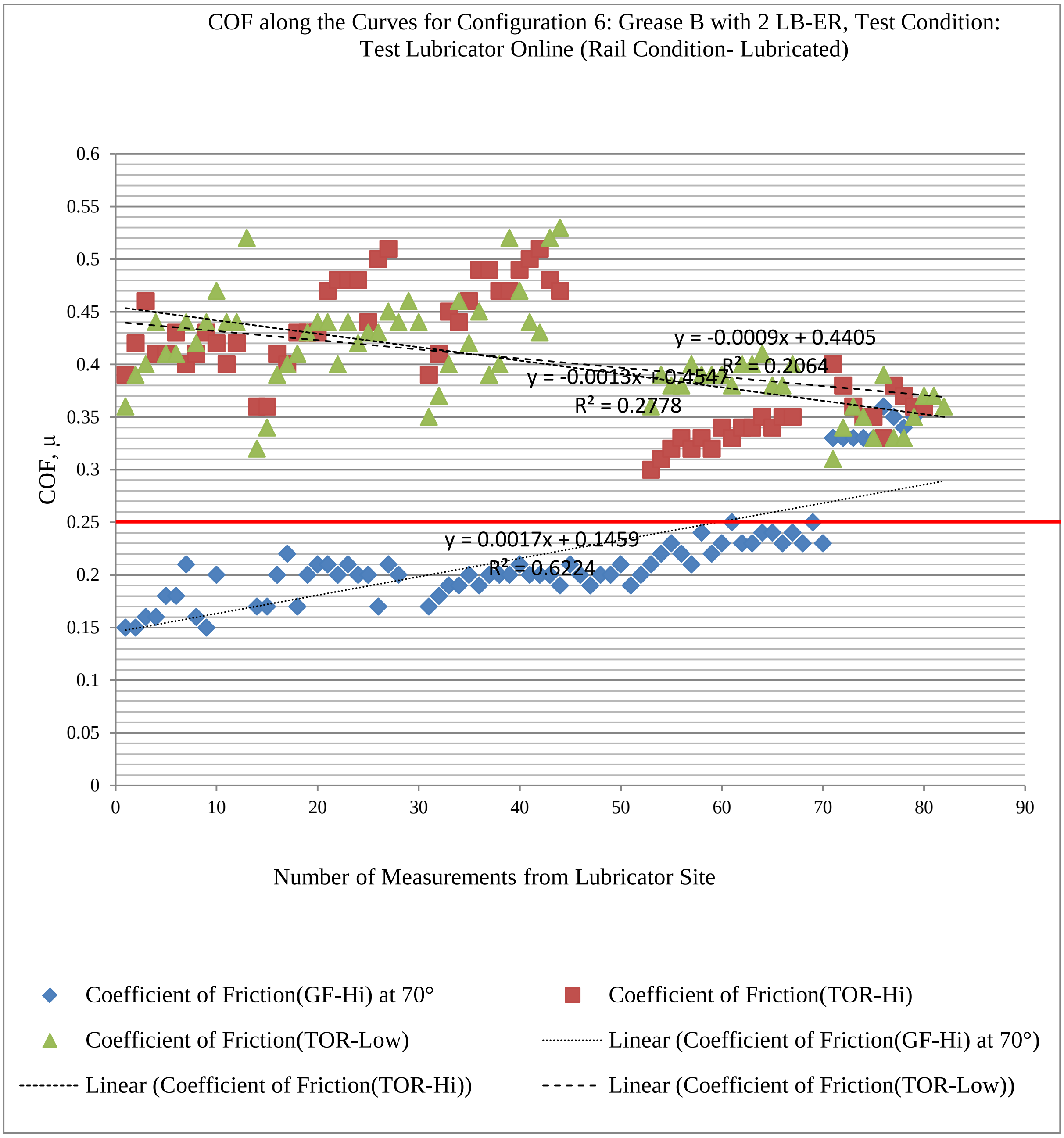



Figure 4.16: COF data distribution along the data collecting curves for configuration 6 while the track condition is lubricated with online test lubricator

### 4.4.8 Configuration 7: Grease E with 2 LB-ER, Test Condition: Test Lubricator Online (Rail Condition- Lubricated)

Configuration 7, Grease E with 2 long bars on each rail, only achieved effective GF lubrication of Curves 2 and 3 with COF values below 0.25. Figures 4.17 and 4.18 show the COF distributions. Grease was splashed above the TOR at the lubricator site and along the early curves. The chemical and mechanical stability of this grease limits its capability to sustain GF lubrication and, after just a few curves, there was no sign of grease at the gauge face. No symptoms of burnt grease falling off from the GF were noticed.

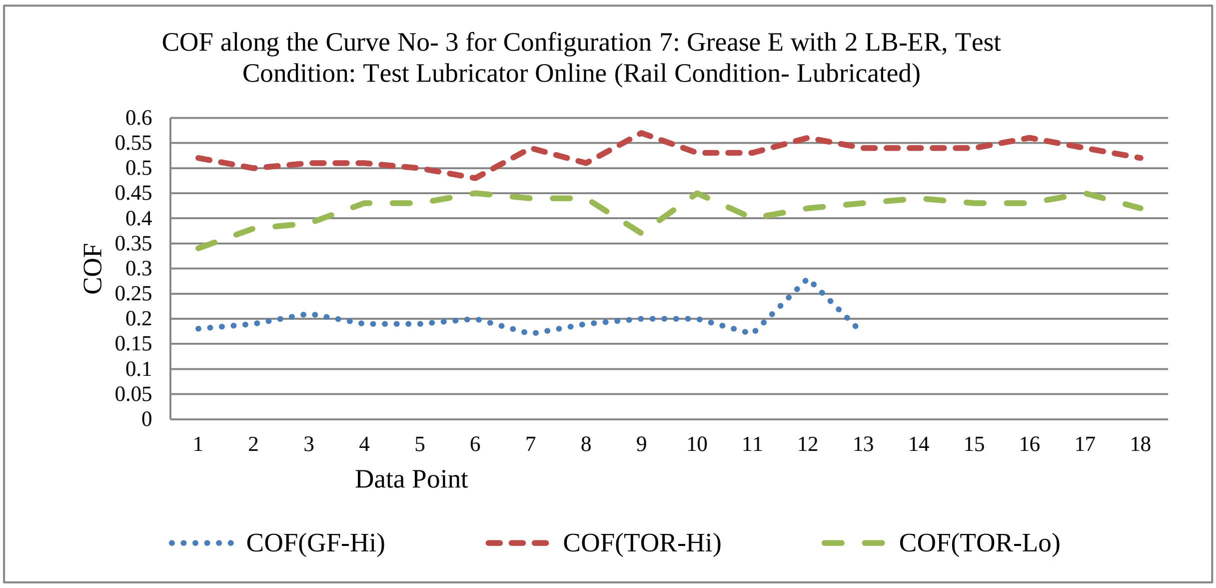



Figure 4.17: COF data distribution along Curve No-3 for configuration 7 while the track condition is lubricated with online test lubricator

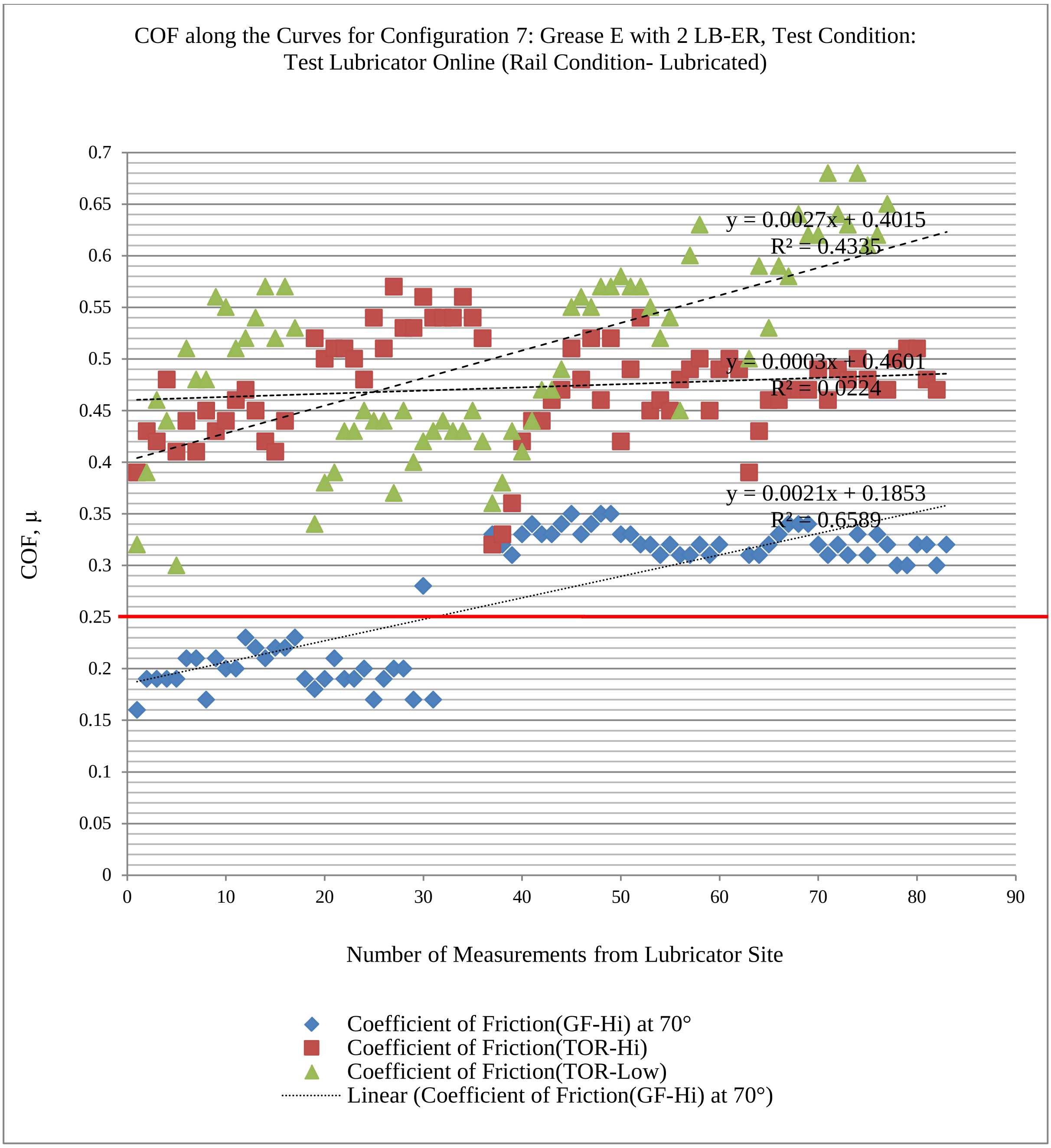



Figure 4.18: COF data distribution along the data collecting curves for configuration 7 while the track condition is lubricated with online test lubricator

### 4.4.9 Configuration 13: Grease C with 2 LB-ER (Supplier Y), Test Condition: Test Lubricator Online (Track Condition- Lubricated)

The best performed grease with Supplier X equipment was subsequently tested with Supplier Y equipment. Supplier Y equipment achieved effective lubrication only from Curve 2 to Curve 5 with this high performed grease. Though it achieved 4.62km carry distance with Supplier X equipment, Supplier Y equipment (2LB-ER) did not do well and provided a carry distance of just 1.55km (Table 4.4). Figures 4.19 and 4.20 show that GF COF was maintained below or equal to the accepted value of 0.25 for the first few curves and then there was a significant rise above that level, demonstrating the ineffective lubrication with the GF friction no longer under control. The data points show no steady trend, with a sharp rise and then a sharp decline in the values. The low values of GF COF after higher values in the middle indicate a significant shift in the grease distribution through the curves. Grease was observed on the high rail TOR. This is not acceptable due to the potential for loss of necessary traction and braking resulting in excess sanding through the curves.

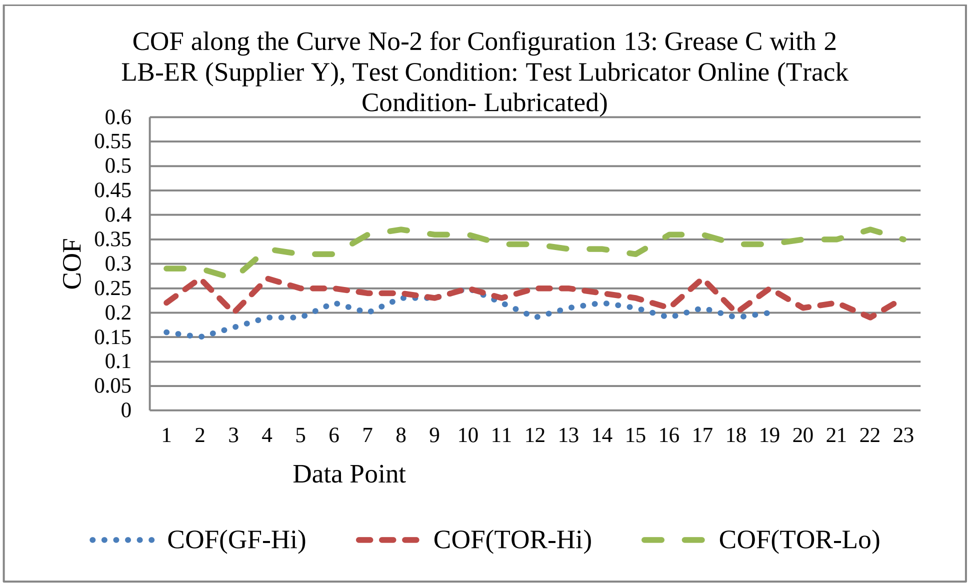



Figure 4.19: COF data distribution along Curve No-2 for configuration 13 while the track condition is lubricated with online test lubricator

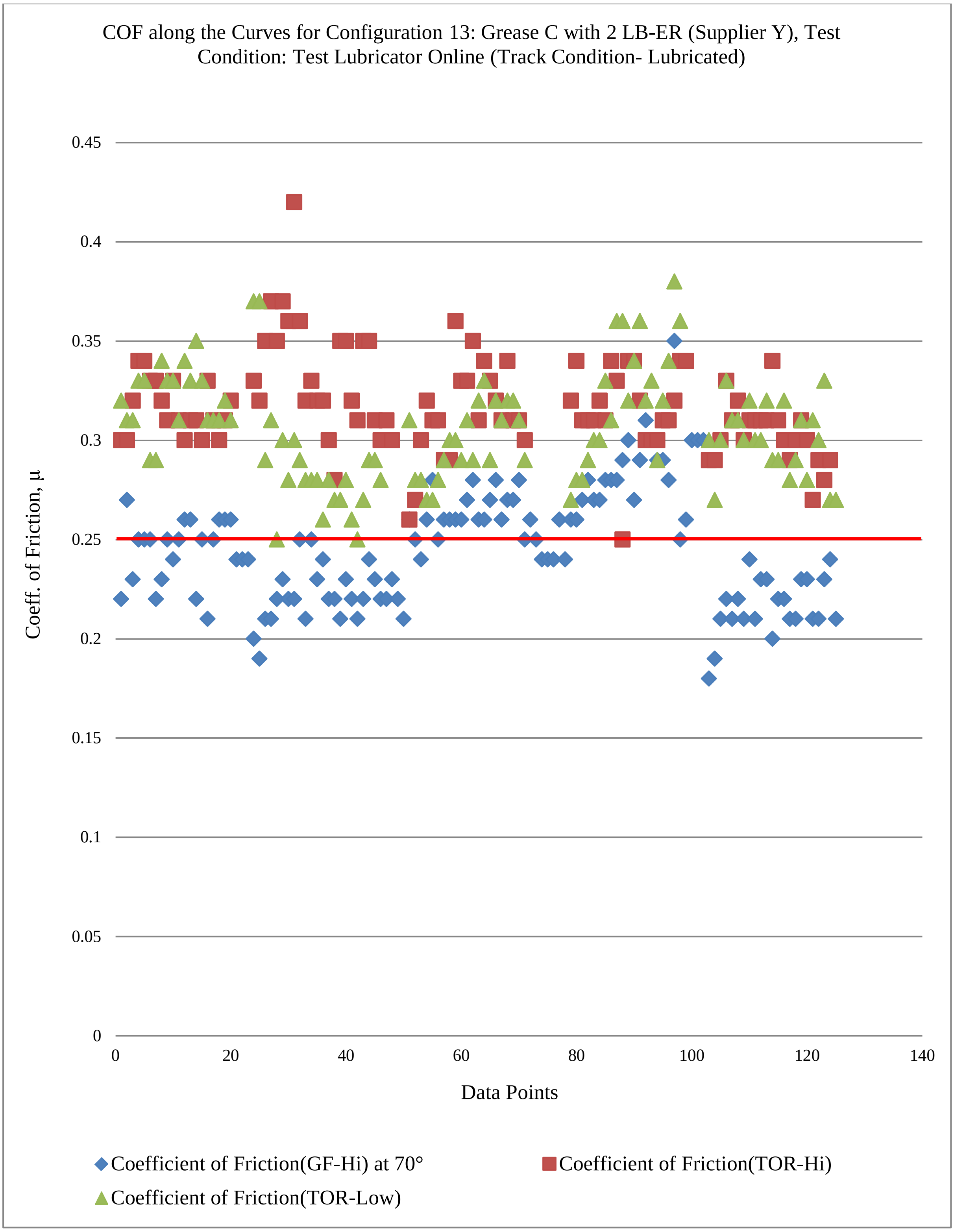



Figure 4.20: COF data distribution along the data collecting curves for configuration 13 while the track condition is lubricated with online test lubricator

## 4.5 Grease Coverage on Rail Gauge Face

Presence of grease at the GF is expected to be effective over a substantial distance. Achieving this depends on the capability of grease to be transported by the surface of the wheel for the furthest distance possible along the rail. Different configurations of equipment and grease combinations have shown substantially different distance coverage along the GF; in other words, grease carry distance varies for the same grease in different equipment configurations as well as from one grease to another in the same equipment configuration. Tackiness and bi-polar properties of a grease are very important to the achievement of quality lubrication. Many types of grease lose their properties within a few wheel passes and end up at the rail foot as burnt grease without any lubricity, just like a lump of burnt hydrocarbon. Visual inspection and automatic data collection with a digital hand pushed tribometer measurement (Measurement of COF) has unambiguously shown the difference between different grease quality and performance under the movement of loaded heavy haul trains. Inspection revealed that bulk amounts of grease were present along the bottom of the GF, but with no grease at the contact interface. Figure 4.21shows evidence of the variable quality of grease coverage in the worst (Figure 4.21(b) and best (Figure 4.21(c) test configurations.


(a) (b) (c)

Figure 4.21: GF coverage with long bars and different greases in field tests: (a) 2+2 Long bars and Grease A, (b) 2+2 Long bars and Grease B, (c) 2+2 Long bars and Grease C

## 4.6 Discussion

The mechanism of lubricant supply along the rails at the contact interface of the GF and wheel flange is simple. Grease is ejected by the lubricator applicator bar onto the GF and is picked up by passing wheels and applied by transferring from the wheel to the GF during the forward rotary motion of the wheel. Supply of grease by the wheels onto the gauge face is obviously desired for a substantial distance along the rail curves. Effective lubrication is expected to be sustained at the wheel/rail interface contact band up to this substantial distance, and the length of track for which this is achieved is referred to as the carry distance. In other words we can say a good lubricant must possess a property that a wheel carries it the furthest distance possible. Different configurations of equipment and grease show different coverage at the gauge face for different carry distances. Grease carry distance varies significantly with each configuration of equipment and with each grease. Due to poor film strength under the combined effects of load, speed and environmental parameters, many greases fail to form a film at the contact surface between the flange and the GF. The cause of this could be two-fold: one that they fail to achieve the required carry distance, and secondly that their physio-chemical properties cannot resist the combined effects of load, speed and environmental conditions. The experimental study revealed that many rail lubrication greases were ineffective within a small number of wheels passes and ended up at the rail foot as a puddle of burnt grease without any lubricity, much like a lump of burnt hydrocarbon. Visual inspections and automatic data collection with a digital hand pushed tribometer measurement showed the difference between the varieties of greases under loaded heavy haul train movements. In other cases, inspections revealed grease at the bottom of the gauge face, but no grease at the actual contact interface.

Each of the greases has its own film strength and hence, for the same operating conditions such as load, speed and environmental factors, distinct grease has a different carry distance. Load carrying capacity of the lubricant film also varies with the constituents of its additive package. Under boundary lubrication conditions, those greases that have bi-polar additives have a greater film strength and can withstand higher loads and speeds. Similarly, grease that has higher tackiness can carry for longer distances than those that have lower tackiness. Thus tackiness and boundary lubrication properties play a vital role in achieving an appropriate quality of lubrication. The constituents of the additive package are confidential and highly secret; it is difficult to ascertain the chemical properties responsible for low friction and higher grease carry distances.

### 4.7 Comparative Study of Grease and Equipment Performance

The data collected for various configurations for five types of greases has been further analysed for their effectiveness with different equipment set-ups to ascertain the best practice lubrication.

### 4.7.1 Performance of Greases with Same Equipment

Measurements of grease carry distance are presented in Table 4.3 which shows the difference of grease in their performance with the same applicator bar set-up in the field tests. The performance of each of the five greases in the tests shows significantly different levels of achievement in terms of the carry distance that provided effective lubrication on the gauge face. With two long applicator bars from the same supplier (Supplier X) installed on each rail, Grease C achieved the maximum effective lubrication carry distance of 4.623km. Table 4.3 shows the difference of the greases in their field trial performance with this same applicator bar set-up. The optimum grease application rate setting for lubricator type ‘X (2LB-ER) with five different grease was 0.25 second of pumping at every 12 wheel passing.

Table 4-3: Grease application rate settings and achieved carry distances by each of the 5 test greases with Supplier X equipment (two long bars on each rail on tangent track)

| Grease | Travel | Lubricator | Setting | Axles |
|---|---|---|---|---|
| | Distance (km) | Type | Pump/Wheel | Passing |
| A | 0.33 | X (2 LB-ER) | 0.25x12 | 42627 |
| E | 1.28 | X (2 LB-ER) | 0.25x12 | 42377 |
| B | 2.97 | X (2 LB-ER) | 0.25x12 | 57682 |
| D | 2.65 | X (2 LB-ER) | 0.25x12 | 57379 |
| C | 4.623 | X (2 LB-ER) | 0.25x12 | 38591 |

The same lubricator and grease dispensing equipment has achieved different levels of GF lubrication effectiveness when different greases were applied. It can be observed from Table 4.3 that:

- All five greases were tested with the same equipment set-up (Supplier X equipment with 2 long bars on each rai).
- All five greases had the same pump and wheel setting for grease delivery (pump activation for 0.25 seconds after every 12 wheel pass).
- Grease C achieved the highest carry distance of 4.623km.
- Grease B recorded the second highest carry distance of 2.97km.
- Grease A produced the lowest carry distance of 0.33km.
- Grease C was accepted as the best grease and grease A accepted as the worst grease.

- Grease performance plays the most significant role in both economic gain and operational performance.

### 4.7.2 Performance of One Grease with Different Equipment

Different equipment set-ups performed at very different levels when the same grease was applied on the rail gauge face.

Table 4.4 compares the performance of various equipment set-ups from both Supplier X and Supplier Y. It shows how two different applicator bar arrangements used in the field tests performed in carry distance when the same grease (Grease C) was applied.

Five different lubricators were tested with different applicator bar combinations where only Grease C was used. It can be observed from Table 4.4 that:

- All five of the lubricators achieved different levels of GF lubrication effectiveness with the same Grease C.
- Lubricator 5 and Lubricator 3 supplied by Supplier X had the highest grease consumption rate and Lubricator 5 achieved the longest carry distance of 4.623km. Excess grease consumption of Lubricator 5 is 2.23 grams per 1000 axles as compared to Lubricator 3 which is negligible. Any of the Lubricators in Table 4.4 will not be viable from economic point of view over the benefits of largest length of effective rail protection with target value of friction with effective lubrication. Rest of the Lubricators may have less grease consumption but they have also achieved very low carry distance or effective rail protection with target value of friction. We optimise the grease consumption rate to avoid rail head contamination from splash. Performance lubricator is evaluated based on the longest carry distance not based on the consumption of grease. If we consider the cost of grease consumption and cost of rail/wheel maintenance and replacement, extra grease cost for Lubricator 5 will be very low or nearly negligible.
- Lubricator 5 achieved the best carry distance using long applicator bars; Lubricator 3 achieved the equal best carry distance using short applicator bars.
- Lubricators 4, 2 and 1 supplied by Supplier Y had lower or equal carry distances compared to the equivalent configurations of Lubricators 5 and 3, though all Supplier Y set-ups showed significantly lower grease consumption rates compared to Supplier X set-ups.

Table 4-4: Grease application rate settings and achieved carry distances by each of the 5 test greases with

| Lubricators | Different Applicator Bars | Test Grease | Distance Travel | Pump Setting | Axle Setting in control box | Passing Axles | Grease/pump | | Grams/1000 axles |
|---|---|---|---|---|---|---|---|---|---|
| | | | km | Seconds | Axles | | Dispensed grams (based on pump setting in seconds) | | |
| Lubricator 1 | Y (2LB-ER-WB) | C | 0 | 2 | 24 | 76805 | 2.4 | | 100.00 |
| Lubricator 2 | Y (2LB-ER) | C | 1.5 | 2 | 24 | 122673 | 2.4 | 2 curves | 100.00 |
| Lubricator 3 | X (SB-HR) | C | 1.56 | 0.2 | 18 | 68160 | 10.16 | | 564.44 |
| Lubricator 4 | Y (2SB-HR-WB) | C | 1.56 | 2 | 32 | 60721 | 4.8 | 2 curves | 150.00 |
| Lubricator 5 | X (2 LB-ER) | C | 4.623 | 0.25 | 12 | 38591 | 6.8 | | 566.67 |

## 4.8 Comparative Study of Friction Level from Curve to Curve and Grease Carry Distance

To quantify the overall friction level achieved in each individual curve, an average coefficient of friction (ACOF) has been calculated from the sum of the total values of all GF COF values. If the average COF value is less than or equal to the target value of 0.25, it is considered that the curve has been effectively lubricated. The grease carry distance is considered to be up to the end of the last curve having an ACOF less than or equal to 0.25. The colour code in Table 4.5 shows 'ACOF GF-Hi (2 LB-ER, Supplier X, Grease C)' has the accepted level of friction up to Curve 14 and this configuration demonstrated the longest carry distance of 4.623km, whereas the 'ACOF GF-Hi (2 SB-HR, Supplier X, Grease A)' has the accepted level of friction up to Curve 2 and demonstrated the shortest carry distance of 0.34km.

Figure 4.22 shows the rise of ACOF from the lubricator site up to the last curve measured for each of the configurations tested. The rate of change of ACOF GF-Hi (2 LB-ER, Grease A), dy/dx = 0.049, which shows that Grease A is the worst grease. Rate of change of ACOF GF-Hi (2 LB-ER, Grease C), dy/dx = 0.0107, which shows that Grease C is the best grease.

The worst grease has a 4.579 times faster rate of change of average gauge face coefficient of friction than the best grease. The worst grease, Grease A, lost the expected level of friction within 0.33km which covers up to Curve No-2 and no trace of grease at the gauge face was noticed. It was completely dry.

The best grease, Grease C, has the expected level of friction up to 4.623km which covers up to Curve 14 while still maintaining the ACOF at a steady rate just above 0.25.

Table 4-5: Performance of different configurations of lubricator units from different suppliers with different greases

| Curve Direction | Curve No From Lubricator | ACOF GF-Hi (2 SB-HR, Supplier X, Grease A) **0.34km** | ACOF GF-Hi (1 LB-ER, Supplier X, Grease A) **1.39km** | ACOF GF-Hi (2 LB-ER, Supplier X, Grease A) **0.33km** | ACOF GF-Hi (2 LB-ER, Supplier X, Grease C) **4.623km** | ACOF GF-Hi (2 LB-ER, Supplier X, Grease D) **2.65km** | ACOF GF-Hi (2 LB-ER, Supplier X, Grease B) **2.96km** | ACOF GF-Hi (2 LB-ER, Supplier X, Grease E) **1.28km** | ACOF GF-Hi (2 LB-ER, Supplier Y, Grease C) **1.55km** | ACOF GF-Hi (2 SB-HR, Supplier X, Grease C) **2.87km** | ACOF GF-Hi (2 SB-HR, Supplier Y, Grease C) **0.72km** |
|---|---|---|---|---|---|---|---|---|---|---|---|
| R | 2 | 0.19 | 0.19 | 0.18 | 0.2 | 0.24 | 0.17 | 0.2 | 0.24 | 0.2 | 0.22 |
| L | 3 | 0.26 | 0.21 | 0.31 | 0.16 | | 0.2 | 0.2 | 0.22 | 0.16 | 0.23 |
| R | 4 | | 0.23 | 0.34 | 0.19 | 0.17 | 0.2 | 0.33 | 0.26 | | 0.25 |
| R | 5 | 0.26 | 0.25 | 0.28 | | 0.24 | | 0.32 | | 0.2 | |
| R | 6 | 0.34 | | | | | | | | | |
| R | 7 | | | | | 0.24 | | | | | 0.27 |
| L | 8 | 0.33 | 0.3 | 0.33 | | | | | 0.22 | 0.24 | 0.25 |
| R | 9 | | 0.28 | | 0.23 | 0.41 | 0.23 | | | 0.2 | 0.3 |
| R | 12 | | 0.26 | | | | 0.34 | | | | |
| R | 14 | | | | 0.24 | | | | | | |
| R | 18 | | | | 0.35 | | | | | 0.35 | |
| R | 19 | | | | 0.38 | | | | | 0.29 | |

Note: The carry distance for each configuration is the km figure at the bottom of the column heading. Numbers in those columns are the average GF COF values recorded for the relevant curves.

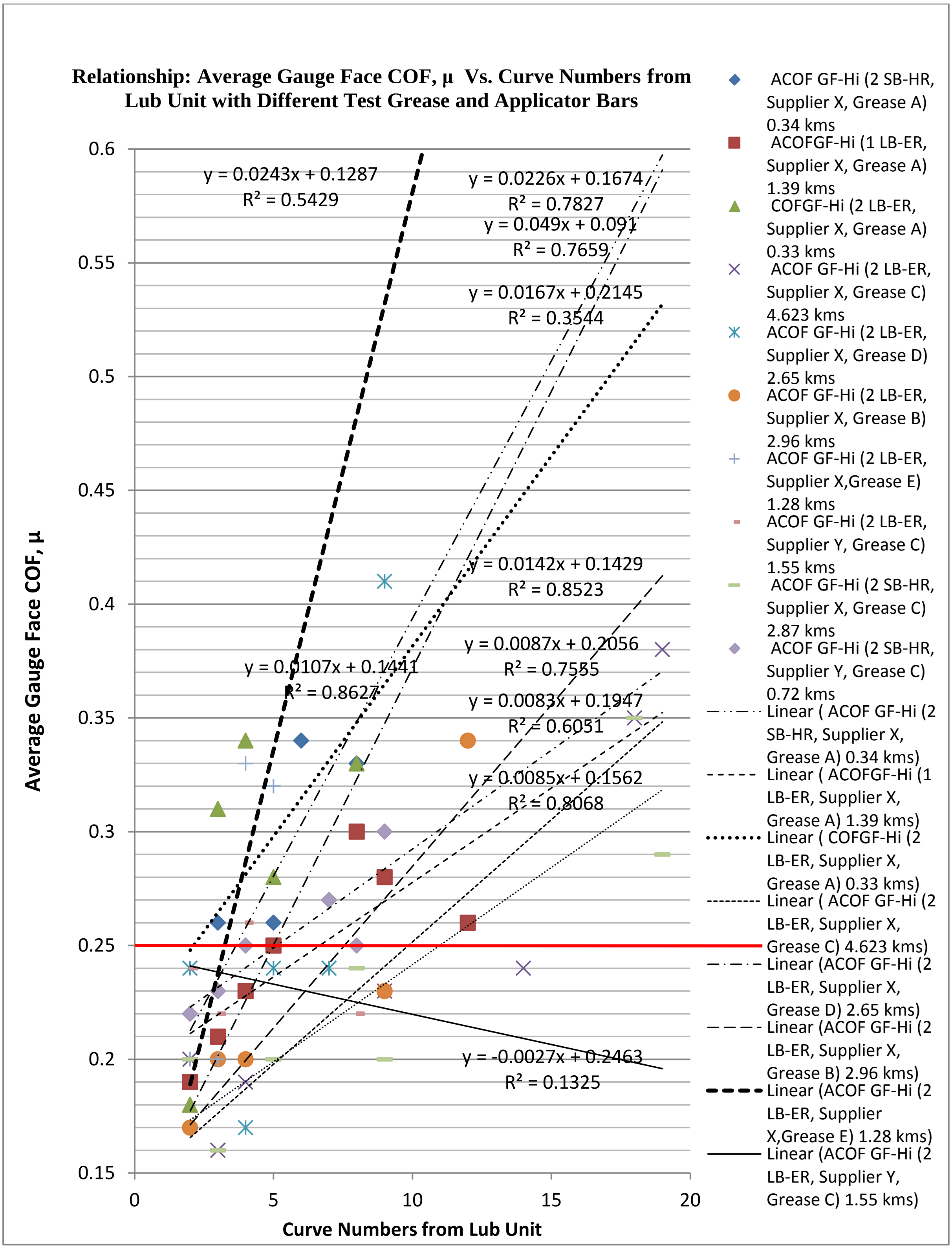



Figure 4.22: Change in Average Gauge Face COF, µ with respect to Curve Numbers

In rail curve lubrication it is highly important to have consistent values of coefficient of friction below the target gauge face friction value with maximum achievable carry distance.

Average Coefficient of Friction GF-Hi (2+2 Long Bars, Supplier X with Grease C) demonstrated the best consistency over the longest carry distance of 4.62km. This configuration has a very low rate of change of gauge face COF compared to other tested configuration.

### 4.9 Conclusions

The technical data analysis gives us an in depth understanding and decision making power to determine appropriate selection of lubrication equipment and grease for best performance. The following conclusions and recommendations can be made:

- Field tests must be carried out to find the best configuration of applicator bars and grease.
- Electric lubricators with long bar applicators installed in tangent track are the most effective system configuration for wayside lubrication.
- Electric lubricators with long bar applicators utilising the best EP grease can create most effective lubrication with longest carry distance.
- In addition to achieving an effective friction level, the longest carry distance are also desired from the wayside lubrication system.
- In the field tests, the electric units with long bars from Supplier X using the best grease (Grease C) achieved the longest carry distance.
- Many other greases, including the current practice, did not perform well with any equipment set-up.
- Combinations of the appropriate applicator bar and grease configuration can provide the best performance with longest grease carry distance.
- Technological enhancement of the equipment and the resulting higher capability should be adopted by the rail operators because they can generate enormous savings and flexibility in resource allocation and motivation.

# Chapter 5

# RAIL CURVE LUBRICATION PLACEMENT MODEL

## 5.1 Introduction

Wayside lubrication methods are used to apply grease to the gauge corner of the rails to be picked up by train wheels while passing through the lubricator site. This method was first introduced to Australian railway systems in the 1950s when it became apparent that lubrication improves the rail and wheel life few times compared to unlubricated systems. Different methods and placement locations are currently being used in different rail networks to lubricate the wheel/rail contact interface without understanding the science behind it. Under current practices, lubricators are installed either too close together or too far apart without proper investigation and research into the effects of grease or equipment performance. Field investigations show that an effective lubrication level ceases within a few metres of wayside lubricator sites in many curves. This makes it necessary to investigate the effect of placement location, equipment, quality of grease and wheel/rail contact surface conditions. Field trials and laboratory tests show a huge variation in grease loss and carry distance due to the influencing factors discussed above, hence there is a need to analyse this data and develop a model.

A limited number of studies are available in the literature on wayside lubricator placement models, and evidence of studies based on the impacts of the parameters mentioned above is rare.

A study by Ishida and Aoki (2004) investigated gauge corner wear in Japanese railways. In this research some important parameters that need to be considered in wayside lubrication programs were identified. These included equipment technology, selection of the placement location of lubricators, selection of appropriate applicator bars (either long bars in tangent track or short bars in curves), selection of suitable grease, rate of grease application and other necessary track and traffic parameters of a placement model to design the system development for total lubrication planning of any corridor. Gauge side wear on the high rails of curves is a common problem (Turner 2008) where the correct placement of a lubricator plays a vital role.

Thus field and lab test data is analysed rigorously and a mathematical model is developed for rail curve lubrication taking into consideration a comprehensive approach for understanding the effect of wayside lubricator placement in this study. This research will enhance the understanding of the lubrication mechanism and the effective grease carrying capacity at the gauge corner area of the wheel/rail interface. This work has also been reported in detail by the author in the CRC for Rail Innovation Project Report (CRC Australia 2014).

### 5.2 Existing Placement Models for Wayside Lubrication

Wayside lubrication placement practice is a highly complicated issue. Generally, lubricator placement is based on assumptions and visual inspection is a common practice. There is no standard method available for determining the placement of lubricators. Recent studies in Europe (INNOTRACK 2009) show that reasons to use wayside lubricators include reduction of gauge face wear, traction coefficient, controlling rolling contact fatigue (RCF) at the gauge corner, and minimising the risk of potential derailment due to the combination of dry gauge face/wheel flange. This study indicates that, in Europe, selection of wayside lubricator placement location in curves is based on curves with a history of excessive gauge face wear, RCF, radius less than 1500m, and cant deficiency greater than 50mm. In Australia, decisions about wayside lubricator placement locations based on field studies are rare; rather, they are mostly based on the experience and understanding of local maintenance personnel.

Though the basic principles of lubrication are well known, very limited publications are available on the real life procedures of lubricant application and measures of success in the field. The placement and maintenance of the lubricators are mainly the responsibilities of experienced lubrication personnel (Marich et al. 2000). Those authors suggested that lubrication was not needed for curves with radii greater than 500-600m, and that very efficient lubrication could be achieved by positioning lubricators at the end of the shallower curves (600-1000m radii), While the positioning of lubricators near sharper curves lead to excessive lubrication. Sroba et al. (2001) extended and modified the formula of de Koker (1994) by including a bogie factor. Table 5.1 shows the factors considered in the formulae developed by de Koker and Sroba et al.

The factors considered by these researchers have been critically reviewed in this study and found to have some shortcomings; other factors that could influence the quality of lubrication that should also be considered have been identified.

Table 5-1: Factors considered in current wayside lubricator placement models

| Track Related Factors | | Traffic Related Factors | |
|---|---|---|---|
| de Koker (1994) formula | Sroba et al. (2001) model | de Koker (1994) formula | Sroba et al. (2001) model |
| (C+S) | (C+S) | T (Traffic factor) | T |
| P (Applicator factor) | P | L (Locomotive factor) | L |
| G (Grease factor) | G | A (Axle load factor) | A |
| R (Radius factor) | R | V (Velocity factor) | V |
| | | M (Misalignment factor) | M |
| | | B (Braking factor) | $B_R$ (Braking factor) |
| | | | $B_G$ (Bogie factor) |

where,

C = Original length of a curve

C+S = $C_{eq}$ = Equivalent curve length for each curve

S = 2.5% of Tangent Length before Curve + 2.5% of Tangent Length after Curve

The following shortcomings have been identified in the currently available models:

- Location and placement are based on arbitrary assumptions.
- Evaluation or ranking of lubricator technology from various perspectives is rare.
- Ranking of grease based on performance in field study does not match Ranking of applicator bars based on grease distribution.
- Grease carry distance is not defined.
- Cost-benefit analysis is missing.
- Performance and economic benefits of lubricator and applicator bars have not been compared.
- Factors such as rail and wheel profiles have not been taken into consideration.

The above issues have been critically examined and have been taken into account in the proposed model as detailed in the following sections.

## 5.3 Wayside Lubricator Placement Model

In this research, the wayside lubricator placement model is a part of the total lubrication practice model which is a conceptual model that initiates from idea generation of lubrication to the expected effectiveness evaluation. The conceptual total model of lubrication is the operating strategy that is based on currently adopted methods and their cost-benefit analysis or economic modelling over a period of practice time. Figure 5.1 shows the structure of a total lubrication practice model which would be suitable for any method of lubrication either by wayside, on-board or hi-rail application.

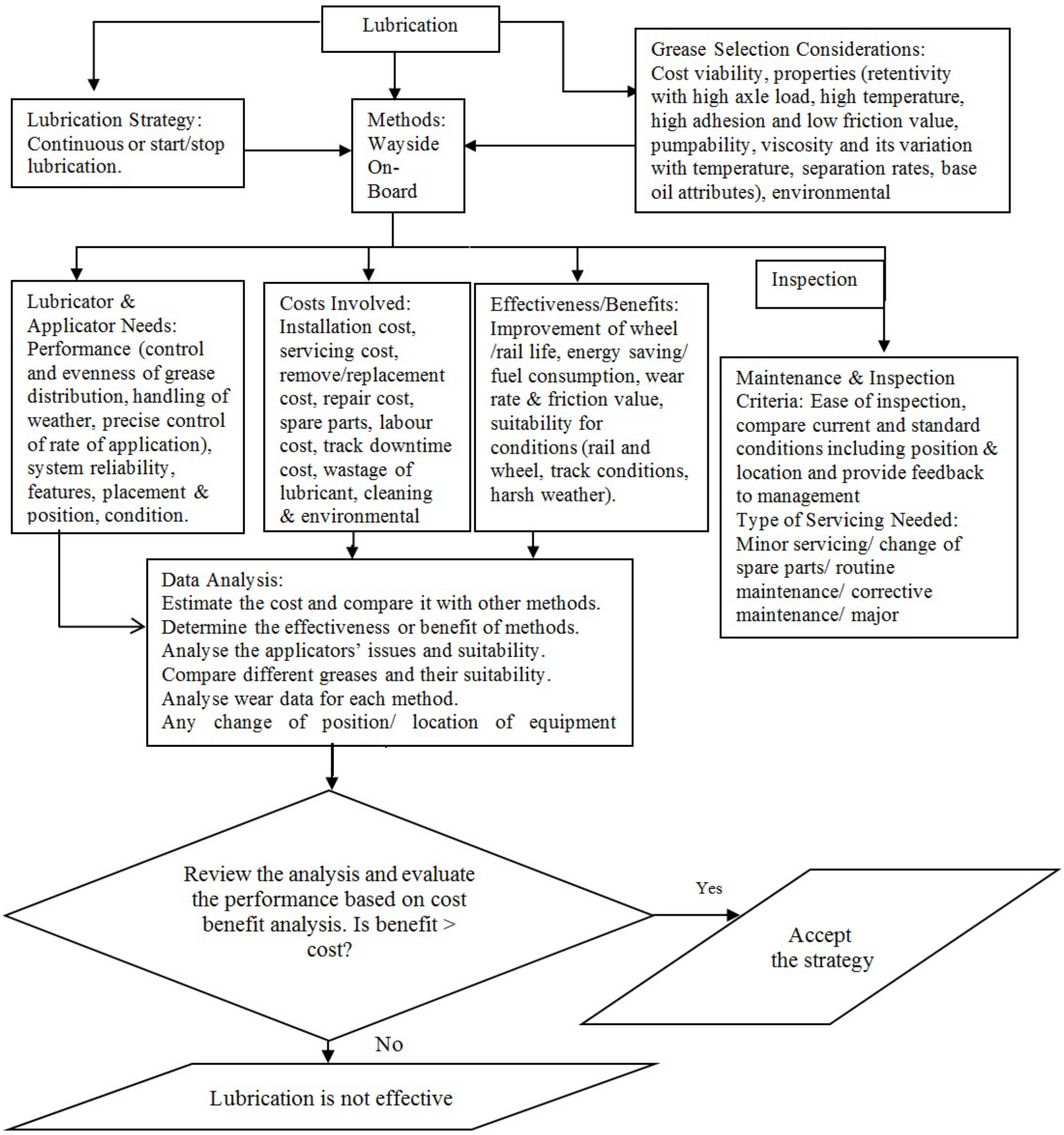



Figure 5.1: Total lubrication model structure for overall lubrication practice (Uddin & Chattopadhyay 2009)

The proposed Wayside Lubrication Placement Model is basically an extension of the model developed by de Koker that can be represented mathematically as stated below:

$$\text{de Koker's final formula, } \frac{\frac{(C+S)*G*R}{P}}{T*L*A*V*M*B} \tag{5.1}$$

This formula is a conceptual theoretical model where relevant factors need to be adjusted to a workable formula and extensive field trials and development needed to give it a practical approach. This formula has been rearranged by Sroba et al. (2001) to implement in practical application. Further improvement and appropriate quantification of various factors existing factors and new factors has been considered in this research to give a better application accuracy of Sroba et al (2001) model. In the above de Koker equation the applicator or greasing plate factor was considered as length of their standard applicator bar divided by the length of applicator bar used in specific application. In this research applicator bar factor was defined based on the achieved carry distance data.

In de Koker (1994), this formula was used to calculate the value of the “de Koker number” for each track segment (tangents and curves). The “de Koker number” has units of length times degree of curvature, and thus does not represent the actual distance along the track as measured from the lubricator. As originally applied, the COF of the gauge face of each high rail was measured, starting at the first curve after the lubricator, until it rose above 0.25. The “de Koker number” was calculated for each curve and tangent between the lubricator and the curve where the COF first rose to 0.25. These numbers were then summed to yield the total “de Koker number” between lubricators. The next lubricator would be positioned in the tangent following the curve where the COF reached 0.25, and all subsequent lubricators would be positioned in tangent segments further along the track such that the total “de Koker number” for the tangents and curves between lubricators equalled the value of the de Koker number originally determined for that specific railway system. In the proposed model, an index called the Lubrication Effectiveness Index (LEI) has been derived. The LEI is defined as a representative number for each rail curve that represents the lubrication quality taking into account a variety of parameters including placement of the lubricators. It is determined by multiplication of all the considered values of track factors divided by the product of all the considered values of traffic factors and is expressed as:

$$Lubrication\ Effectiveness\ Index, LEI = \frac{(C+S)*G*R*P*L_{perf}*P_{profile}}{T*L*A*V*M*B_R*B_G} \tag{5.2}$$

### 5.3.1 Derivation of Equation

Equation 5.2 clearly shows that both the track and traffic factors affect rail curve lubrication for each curve. Each curve has its own curve length, transition length, and rail profile condition; similarly, the traffic parameters such as speed, traffic direction, axle load, braking conditions, wheel profile condition, bogies and their misalignment have an impact on the LEI. All the above factors may have potential influence on the distribution of effective lubrication on each curve. Due to the limitations of the scope of this research, only salient factors have been investigated in this modelling process.

It is observed that the lubricator location, applicator bar set-up and the type of grease show very different behaviour for each of the curves. To define a representative number for each curve, a simplistic approach was taken so that it is possible to calculate LEI for as many curves as are needed. The LEI can be calculated for each curve separately, then a cumulative value of LEI for the overall network can be calculated by summing the LEI for each curve up to the last curve that achieved an acceptable COF. The tangent (for long bars)/transition (short bars) after the end of this curve will be the next lubricator installation site. Subsequent lubricators should be installed at the end of the same cumulative LEI as for the first lubricator.

The flow diagram in Figure 5.2 shows the process for determining the LEI for a chosen track.

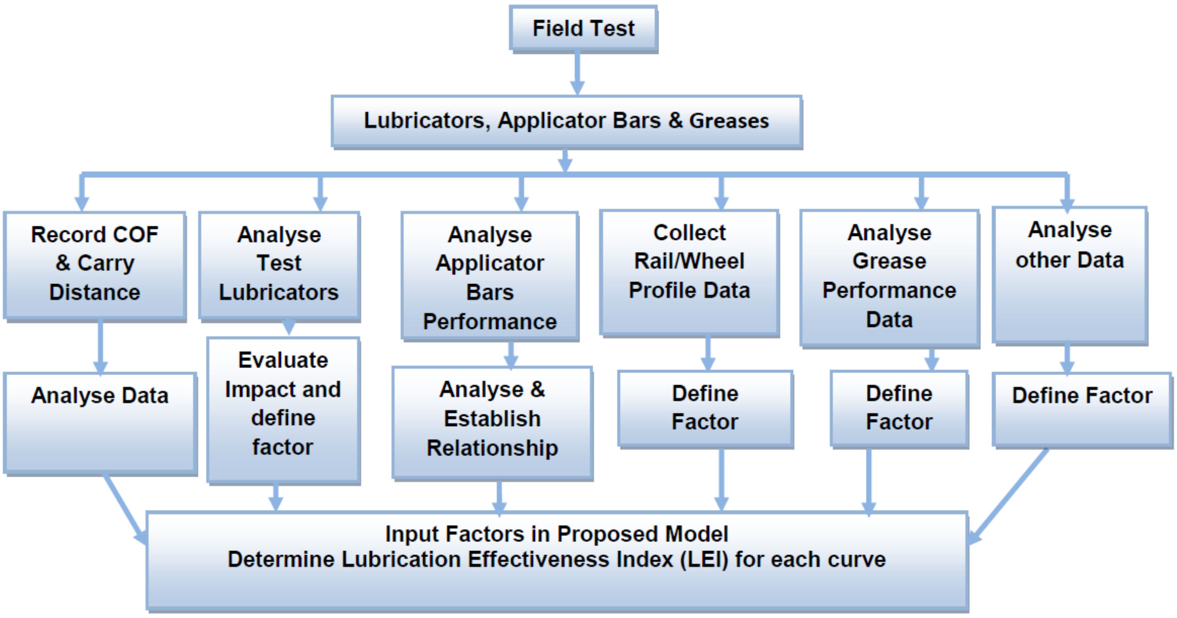



Figure 5.2: Steps to determine Lubrication Effectiveness Index (LEI) (Uddin et al. 2010a; Uddin et al. 2011b)

### 5.3.2 Assumptions in the Modelling

The model is based on the assumptions listed below:

1. The model is considered only for defining effective wayside gauge face lubrication practice.
2. Consider that all lubricators and applicator bars of the same type from the same source are identical in performance except for any manufacturing or assembly defects.
3. Inability to fulfil any conditions of effective lubrication must be considered as the failure of the unit. Potential failure modes and effects are considered in this regard.
4. A series of both left hand curves and right hand curves must be considered for lubrication using long applicator bars placed in tangent track; lengthy sections with only left hand curves or only right hand curves should be considered for lubrication using short bars placed in the transition spirals of curves. The achieved carry distance should be applied to determine the location of subsequent units.
5. The placement locations may be deviated from due to road access inaccessibility of the determined placement location.

6. 2.5% of tangent length before the curve and 2.5% of tangent length after the curve are added to the curve length C as S to determine equivalent curve length to consider wheel flange- rail gauge contact around curves due to considerable hunting (body sway) effect that is apparent when trains are entering and exiting from curves. According to Sroba et al. (2001) & IHHA (2001) tangent track in the Thompson Subdivision has been found with confirmed film of lubrication which confirms the lateral movement of trains on the tangent section of the track. Based on the experience of bogie hunting equivalent curve length is considered as the original curve length plus the effected length of the before and after tangent of each curve. Spoornet in South Africa and Canadian pacific Railway (CPR) have considered 5% of the length of the tangents (Sroba et al. 2001) as extended curve length. Same assumption has been considered in this research to compensate bogie hunting. As this 2.5% is considered as a rule of thumb in this industry to determine equivalent curve length. Average curvature may differ from track geometry, because spirals have been included in the curve length.
7. Impact of grease is considered based on assessed performance in field tests.
8. Effects of reverse curves (when a left hand curve and a right hand curve are situated one after another) are considered to be zero for appropriate long bar application in tangent. If there is any unexpected value of coefficient of friction monitored on any such curves, the applicator bars on the relevant rail should be reinvestigated and checked for the effectiveness of lubrication. For short bar applications, each rail should be considered separately to locate the relevant lubricator location on left hand and right hand curves.
9. Degree of curvature factor is based on the average degree of curvature including the spiral of the curve.
10. Braking factor is based on the severity of the braking force on different downward grades. Braking factor, B = 1, for level track and B = 0.8 for grade (this track was measured using 1 lubricator on grade after a loaded train. More grease burns off on grade).
11. All traffic follows the speed limit.
12. Specified traffic follows the appropriate maximum allowable axle load.
13. Selected grease should follow the performance criteria throughout the year with no variation in carry distance due to weather conditions. Otherwise the lubricators location needs to be reset or determine necessary location for say like winter

condition and summer condition. It is not practical to change lubricator locations twice a year due to seasonal conditions. Therefore, selection of a suitable grease for the full annual range of weather conditions experienced by each specific Australian railway was considered as the only practical solution.

14. Decisions regarding unidirectional and bidirectional traffic depend on the running mode of trains. If the great majority of trains run in one direction, then it is considered as unidirectional. And if a significant number of trains run in both directions, then it is considered as bidirectional.

## 5.4 Factors in the Model

The field trials for this model development are based on a Partial Factorial Experiment (PFE) which does not cover all the parameters that contribute to effective gauge face lubrication. Only a subset of combinations of parameters contributing to the outcome of lubrication practice has been considered. This saves computation time, cost and excessive resources allocation. In this model, two new factors have been added and several factors modified of those used in the previous formula. The models that have been available previously needed improvement; hence new factors have been proposed in developing a new model as are shown in Table 5.2.

Table 5-2: Proposed new factors and modified existing factors

| Introducing New Factors (identified in this research) | Modified Factors (others have used) |
|---|---|
| Lubricator Performance Factor, $L_{perf}$ | Applicator Factor, P |
| Wheel/Rail Profile Factor, $P_{profile}$ | Grease Factor, G |
| | Braking Factor, $B_R$ |
| | Traffic Factor, T |
| | Axle Load Factor, L |

A number of factors considered by de Koker needed to be adjusted to generate a workable formula. As de Koker (1994) suggested, field trials are needed to properly derive parameters. Thus, based on extensive field trials, an improved concept of quantifying and defining parameters and their impact on the effectiveness of wayside lubrication has been considered in this research.

The effects of wheel/rail temperature, wheel/rail profiles, gauge width, surface contamination, track irregularities, lubricator performance, environmental conditions and management factors have not been considered in the existing lubricator placement model. Factors like applicator factor, traffic factor, grease factor, bogie factor and braking factor need to be reinvestigated. Table 5.1 exhibits the track and traffic related factors considered in the de Koker (1994) formula and the Sroba et al. (2001) model.

Existing placement practices can be improved through implementation of new factors and the revision of existing factors. Each factor considered in this research is critically reviewed in the following sections.

### 5.4.1 Lubricator Performance Factor, $L_{perf}$

The technology of wayside gauge face lubrication has changed dramatically over the recent years (Sroba et al. 2001). Wayside lubricators are now electric to better maintain reliability and performance, and the new technology replaces the aging mechanical and hydraulic systems.

For evaluation of lubricator performance, key intake variables considered are their reliability, rate of grease application, and modes of failure. Similarly, the response variables are steady state coefficient of friction on gauge face and achieved grease carry distance. The reliability analysis of hydraulic and electric lubricators with remote performance monitoring showed that the reliability of hydraulic units is significantly lower than the electric units. As detailed below, the reliability evaluation shows that probability of failure of hydraulic units is over 95% and reliability below 5%, whereas probability of failure of electric units is below 30% and reliability is over 70%. Field investigations show that there are a significant number of failure modes such as grease pump, grease tank leaking, delivery hose and applicator bar breakdowns are quite common and failure takes place in hydraulic and mechanical unit sites. Advantages of electric lubricators and common problems on mechanical/hydraulic lubricator sites are given in Table 5.3.

Table 5-3: Comparison of electric and hydraulic lubricators

| Advantages of Electric Lubricators | Disadvantages of Mechanical/Hydraulic Lubricators |
|---|---|
| • Highly reliable, efficient operation, high service life<br>• Electronic and electromagnetic sensors improve reliability and reduce failure rate<br>• Grease application based on precise control of pumping and wheel counts<br>• Less surface contamination and grease wastage<br>• Longer grease carry distance with appropriate amount of grease<br>• Fewer units are required compared to other types<br>• Improved reliability even in harsh climates (extreme cold and heat)<br>• Fewer failures and less maintenance time<br>• Remote Performance Monitoring system can reduce the down time of remote units by generating statistical data, reports, graphical presentations, alarms and warnings<br>• Continuous power is available from solar energy or power grid with a rechargeable battery for emergency back up | • Broken plunger springs<br>• Worn plunger height adjustment cams<br>• Mushroomed plungers<br>• Grease distribution bar backing shims not aligning to manifold holes<br>• Grease delivery hoses cracked<br>• Tanks leaking<br>• Empty tank or run out of grease<br>• Rain water or precipitation mixing with grease in the tank<br>• Extreme build–up of grease on site cause maintenance work difficulties<br>• Assembly points failure<br>• Insufficient bar height adjustment; very much limited to specific rails<br>• Not suitable for high speed or high MGT lines<br>• Too many mechanical components and easy to fail<br>• Significantly low service life compared to advanced electric lubricators<br>• No precise or constant grease delivery rate |

Hydraulic lubricators are commonly used in Australian heavy haul railways. The main features of hydraulic lubricators are the grease tank, pump, and hydraulic plunger/actuator assembly on the gauge side of the rail, hydraulic hose connected to pump externally mounted beside the grease tank, two applicator bars and grease delivery hoses. Figure 5.3 shows a typical hydraulic lubricator.

Figure 5.3: Hydraulic lubricator and huge wastage of grease (Uddin et al. 2010a; CRC Australia 2014)

Hydraulic lubricators are very simple in construction, but a lot of maintenance needed. It delivers grease with every passing wheel without precise control which causes huge grease wastage and severe TOR contamination. Failure in hydraulic lubricator components is common which causes serious interruption to grease application, causes whole system failure and results in the rail remaining poorly lubricated or dry.

#### *5.4.1.1 Reliability Analysis of Lubricators*

Reliability analysis of hydraulic and electric lubricators has been conducted based on failure data collected from field investigations and feedback from lubricator maintainers over the various heavy haul railway networks in Australia. Reliability block modelling has been applied in the reliability calculation of hydraulic and electric lubricators, where reliability of a system which has few components in series is defined as being equal to the multiple of the reliabilities of components, i.e., $R_{sys} = R_1*R_2*R_3*\ldots\ldots R_n$.

In this analysis, each function of the lubricator is considered as a system. Therefore, each function number has a specific number of components.

Overall results of reliability calculation for hydraulic lubricators are presented in Table 5. 4.

Table 5-4: Reliability evaluation of hydraulic lubricators (CRC Australia 2014)

| Hydraulic Lubricator | | | | | |
|---|---|---|---|---|---|
| Component | Function | Time period | Probability of failure | Reliability | Function number |
| | | **weeks** | | | |
| Plunger | Energy | 2 | 0.100 | 0.900 | 1 |
| Hydraulic hose | Energy | 2 | 0.400 | 0.600 | 1 |
| Hydraulic actuator | Energy | 2 | 0.200 | 0.800 | 1 |
| Grease pump | Grease delivery | 2 | 0.500 | 0.500 | 3 |
| Grease tank | Grease delivery | 2 | 0.050 | 0.950 | 3 |
| Grease delivery hose | Grease transport | 2 | 0.500 | 0.500 | 4 |
| Applicator bars | Grease distribution | 2 | 0.800 | 0.200 | 5 |
| | Energy | | 0.568 | 0.432 | |
| | Grease delivery | | 0.525 | 0.475 | |
| | Grease transport | | 0.500 | 0.500 | |
| | Grease distribution | | 0.800 | 0.200 | |
| | System | | 0.979 | 0.021 | |

Failure of electric units can be detected easily if the unit has a Remote Performance Monitoring (RPM) system installed. By contrast, a quick response to failures to minimise the severity of the consequences is not possible in hydraulic units. Table 5.5 shows the live data base generated by RPM of a test electric lubricator; RPM records various operating variables of the unit such as end time of train transit through the site, product/grease level percentage in the tank, pumping cycles, motor volts, motor peak volts while pumping the grease, ambient temperature, total wheel counts, total pump time, wheel setting for each lubrication cycle, pumping time, power status, direction of train through the site, etc., when installed. From the failure analysis and reliability evaluation of electric lubricators in Table 5.6, it is evident that the components do not fail frequently. Field investigation and trials have shown that failure modes are dominated by cavitation in grease pump inlet, damage of applicator bars, clogging of ports, breakdown of grease delivery hose in joints and manifolds. Probability of complete failure of electric lubricators is very low.

Performance ranking of electric lubricators is much higher as compared to hydraulic systems. Based on reliability, the lubricator performance factor for electric units is derived as 1 compared to 0.03 for hydraulic units.

Table 5-5: Live data update from RPM installed electric lubricator on a test site

| Train end time [local_unit_time] | Grease Level % | Pump Cycles | Volts | Volts min | Motor Peak Amp | T ambient ($^0C$) | Total wheels | Total pump time | Wheels setting for each lubrication cycle | Pump time (sec) | Power Status | Power Button Pushed | Direction |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 7/11/2010 23:44 | 22.17 | 23 | 12.89 | 11.38 | 48.42 | 15.09 | 84440 | 967.7 | 18 | 0.2 | TRUE | TRUE | A or B |
| 7/11/2010 23:03 | 22.56 | 7 | 12.93 | 11.46 | 48.42 | 15.8 | 84024 | 963.1 | 18 | 0.2 | TRUE | TRUE | A or B |
| 7/11/2010 21:38 | 22.56 | 20 | 12.92 | 11.43 | 48.65 | 17.12 | 83898 | 961.7 | 18 | 0.2 | TRUE | TRUE | A or B |
| 7/11/2010 20:44 | 22.56 | 1 | 12.96 | 11.47 | 48 | 17.55 | 83530 | 957.7 | 18 | 0.2 | TRUE | TRUE | A or B |
| 7/11/2010 20:38 | 22.46 | 23 | 12.93 | 11.43 | 48.65 | 17.64 | 83518 | 957.5 | 18 | 0.2 | TRUE | TRUE | A or B |

Table 5-6: Reliability evaluation of components and whole system for electric lubricators (CRC Australia 2014)

| Electric Lubricator | | Time period | 52 weeks | | | |
|---|---|---|---|---|---|---|
| Component | Function | Time period start | Time period end | Probability of failure | Reliability | Function number |
| | | weeks | | | | |
| Solar panel | Energy | 250 | 302 | 0.00001 | 0.99999 | 1 |
| Battery | Energy | 250 | 302 | 0.00050 | 0.9995 | 1 |
| Pump motor | Energy | 250 | 302 | 0.00010 | 0.9999 | 1 |
| Controller | Control | 250 | 302 | 0.00001 | 0.99999 | 2 |
| Axle detector | Control | 250 | 302 | 0.00001 | 0.99999 | 2 |
| Axle detector cable | Control | 250 | 302 | 0.00001 | 0.99999 | 2 |
| Grease pump | Grease delivery | 250 | 302 | 0.01000 | 0.99000 | 2 |
| Grease tank | Grease delivery | 250 | 302 | 0.05000 | 0.95000 | 3 |
| Grease delivery hose | Grease transport | 250 | 302 | 0.05000 | 0.95000 | 3 |
| Applicator bars | Grease distribution | 250 | 302 | 0.10000 | 0.9000 | 4 |
| Grease level | Warning indicators | 250 | 302 | 0.00100 | 0.99900 | 5 |
| Tank lid open | Warning indicators | 250 | 302 | 0.00100 | 0.99900 | 6 |
| Temperature sensor | Warning indicators | 250 | 302 | 0.00100 | 0.99900 | 6 |
| Train counter | Monitoring | 250 | 302 | 0.00001 | 0.99999 | 6 |
| Maintenance management system | Condition based maintenance | 250 | 302 | 0.05000 | 0.95000 | 7 |
| Maintenance / servicing | Condition based maintenance | 250 | 302 | 0.05000 | 0.95000 | 8 |
| Remote performance monitoring | Condition based maintenance | 250 | 302 | 0.00100 | 0.99900 | 8 |
| Energy | | | | 0.00061 | 0.99939 | |
| Control | | | | 0.00003 | 0.99997 | |
| Grease delivery | | | | 0.05950 | 0.94050 | |
| Grease transport | | | | 0.05000 | 0.95000 | |

| | | | | | | |
|---|---|---|---|---|---|---|
| Grease distribution | | | | 0.10000 | 0.90000 | |
| Condition based maintenance | | | | 0.09840 | 0.90160 | |
| Programmed maintenance | | | | 0.20000 | 0.80000 | |
| Basic System | | | | 0.35711 | 0.64289 | |
| System with RPM | | | | 0.27546 | 0.72454 | |

#### *5.4.1.2 Lubricator Criticality Assessment*

Consideration of lubricator criticality may contribute significantly to the development of an effective asset management strategy for the whole rail network. This practice and realisation of its benefits are very rare in the heavy haul rail industry. Lubricator criticality assessment was conducted in this research to prioritise the areas where lubricator condition monitoring, maintenance planning and spare parts requirements need to be developed. In this regard, every lubricator is considered as a productive unit which conducts a complete process of grease delivery to applicator bars. Each lubricator consists of a number of maintainable components. Each maintainable item is individually replaceable, for example the grease delivery hose or the solar panels in the lubricators. To assess criticality consequence of failure, the following relevant events need to be determined:

- Grease delivery failure.
- Cost of failure.
- Type of functional failure.
- Failure modes.
- Appropriate control measures for failure.
- Cost/ Benefit analysis.
- Implementation plan for actions.
- Failure and cost data.

The procedure used for the lubricator criticality assessment process can be described as:

1. Use a top/ down approach from the lubricator level, through function level down to component level.

2. Assign a criticality score from 1-5 (where 1 is for the lowest criticality and 5 is for the highest criticality) for effective grease delivery, and associated safety and environmental factors.

3. Develop a risk assessment model for loss of grease delivery, and associated safety and environmental factors.

4. Any score (for grease delivery failure, safety & environment) for a component is the lowest score in the criticality score. The score hierarchy is quantified as follows:

- Lubricator, 5
- Function, 4
- Components, 3 to 1

To determine the consequence of failure, the following questions need to be answered:

- If grease delivery failure occurs, what is the consequence on the lubricator's functionality?
- If this failure occurs, what is the consequence on associated site safety factors?
- If this failure occurs, what is the consequence on associated environmental factors?

Scores for grease delivery failure, and associated safety and environmental factors were kept the same as for the specific item, as follows:

1. No impact on grease delivery.
2. Failure causes little immediate failure of grease delivery; might be a stand by component to activate.
3. Failure causes immediate partial loss (<30% loss of grease delivery).
4. Failure causes immediate partial grease delivery failure until the unit is repaired (=50% loss of grease delivery).
5. Failure causes complete grease delivery failure of the lubricator until it is repaired (no back up, standby, surge capacity or bypass capability).

Based on the criticality score levels, an analysis can be defined to develop the appropriate asset management strategy.

Electric lubricators have precise electronic control of grease application rates based on wheel counts and pump delivery cycle times.

Recent investigations on Australian heavy haul rail networks revealed very few ports of lubricator applicator bars were working due to flattened plungers or grease clogging and the amount of grease picked up by wheels covered only a tiny part of the wheel's circumference. A major portion of the grease was being wasted and the grease carry distance ended within a short distance from the lubricators. Table 5.7 shows the tribometer readings on two hydraulic lubricator unit sites where grease ran out within few metres of the lubricator and the rail gauge face was dry.

Table 5-7: Average coefficient of friction (ACOF) values

| Location (km) | Type of Applicator bars | Type of Lubricator | Grease | Average COF (GF-Hi) | Average COF (TOR-Hi) | Average COF (TOR-Low) |
|---|---|---|---|---|---|---|
| 79.4 | 2 short bars | Hydraulic | Grease A | 0.32 | 0.36 | 0.37 |
| 79.4 | 2 short bars | Hydraulic | Grease A | 0.35 | 0.34 | 0.41 |

The Lubricator Performance Factor $L_{perf}$ was considered as a driving parameter for reliability, maintainability and availability of the equipment. It is rare to find any study that has been conducted on the lubricator reliability, Failure Mode and Effect Analysis (FMEA) and lubricator's asset criticality analysis. In this research reliability analysis of different existing lubricators and trialled lubricators has been studied and found that as compared to the old style hydraulic and mechanical lubricators, all electric lubricators are extensively ahead in reliable lubrication system performance. Reliability analysis shows that the overall reliability of electric lubricators without remote condition monitoring is about 64% and with remote condition monitoring is about 72% whereas reliability of hydraulic lubricator is below about 2%, i.e. electric units are extensively reliable compare to hydraulic lubricators. The Lubricator performance Factor $L_{perf}$ is considered as overall reliability rating of whole equipment as a system. It would be more robust model if integrated well practiced reliability, maintainability and asset criticality data are available to implement and analyse the asset performance in wayside lubrication practice.

### 5.4.2 Applicator Bar Factor (P)

Applicator bars are mounted to the lower part of the rail head to deliver grease to the wheels. Determination of appropriate positioning of applicator bars is undertaken by visual inspection, rail profile measurement, and wheel/rail contact pattern evaluation based on wheel gauge, and track gauge width as measured with a bar gauge.

#### *5.4.2.1 Applicator Bar Set up*

Wheels pick up grease from the applicator bar site, then transport and apply it to subsequent curves. It is a challenge to transfer grease from the wheel to the rail by picking an optimal quantity and carry it for the maximum possible distance. This study reveals that it depends on the type of bars and their appropriate placement. Applicator bars play a significant role in the grease transport mechanism, but they are not effective unless located properly or not suitable for precise grease application, no matter what type of metering equipment is in place. Currently available applicator bars are generally long bars (1400 mm in length) and short bars (600 mm in length). Two short bars are placed in the transition spiral of curves, whereas four long bars are placed in the tangent track before curves. It was found that long bars in tangent track have advantages over short bars in the transition spiral of curves, namely:

- Long bars do not need removal during the curve grinding cycle which reduces the work requirement for the lubricator maintainer.
- Long bars deliver grease to the rail gauge face area over nearly the full length of the wheel circumference. Therefore, the long bar set-up is capable of delivering significantly more grease to the wheel throat without splash occurring from the grease bead, and hence they can lubricate a greater track length when compared to short bars. The length of each long bar used in this test was 1400 mm (Supplier X), therefore two long bars cover a total 2800mm, whereas the circumference of a wheel at the throat is about 3000 mm.
- Long bars use less grease compared to short bars.
- As long bars are installed on both rails in tangent track, both left hand curves and right hand curves are lubricated by one unit. Therefore the number of long bar units required to lubricate the track is significantly lower compared to short bars units.

Long bars should be placed in tangent track before mild curves to ensure the milder curve helps to distribute the grease around the throat area of the wheels (Figure 5.4). The figure shows (a) the positioning of the bars, and (b) the picked up grease on a train wheel when the train was stopped for inspection to assess the grease distribution on the wheel circumference. Long bars with a support mechanism (below the grease application ports in the form of foam or brushes) in the trough area next to the grease ports spread grease around the wheel flange for distribution on both left and right hand curves.

(a) (b)

Figure 5.4: (a) Long bars on tangent track (CRC Australia 2014), and (b) Grease on the wheel of a train which stopped for in situ inspection

Short bars (Figure 5.5) rely on their placement in the transition spirals of mild curves of either left hand or right hand curves. Two short bars are used in each transition and their placement is around the point where the wheel throat starts to contact the gauge corner. Mild curves are used to properly distribute the grease around the throat area of the wheels to ensure more effective lubrication in sharper curves along the track.

Figure 5.5: Short applicator bars (CRC Australia 2014)

### *5.4.2.2 Grease Carry Distance for Different Bars*

The contribution of different bar set-ups in the grease transport mechanism and the achieved carry distances plays an important role. Evaluation should be based on field test results and analysis of their performance. Field trials have been conducted in an Australian heavy haul network and it was found that a 1.6 times longer carry distance has been achieved with long bars compared to short bars even with a lesser amount of grease application. Figure 5.6 shows the performance (in carry distance) of electric lubricators with long bars in tangent track and short bars in the transition spiral of curves with the same grease, Grease C.

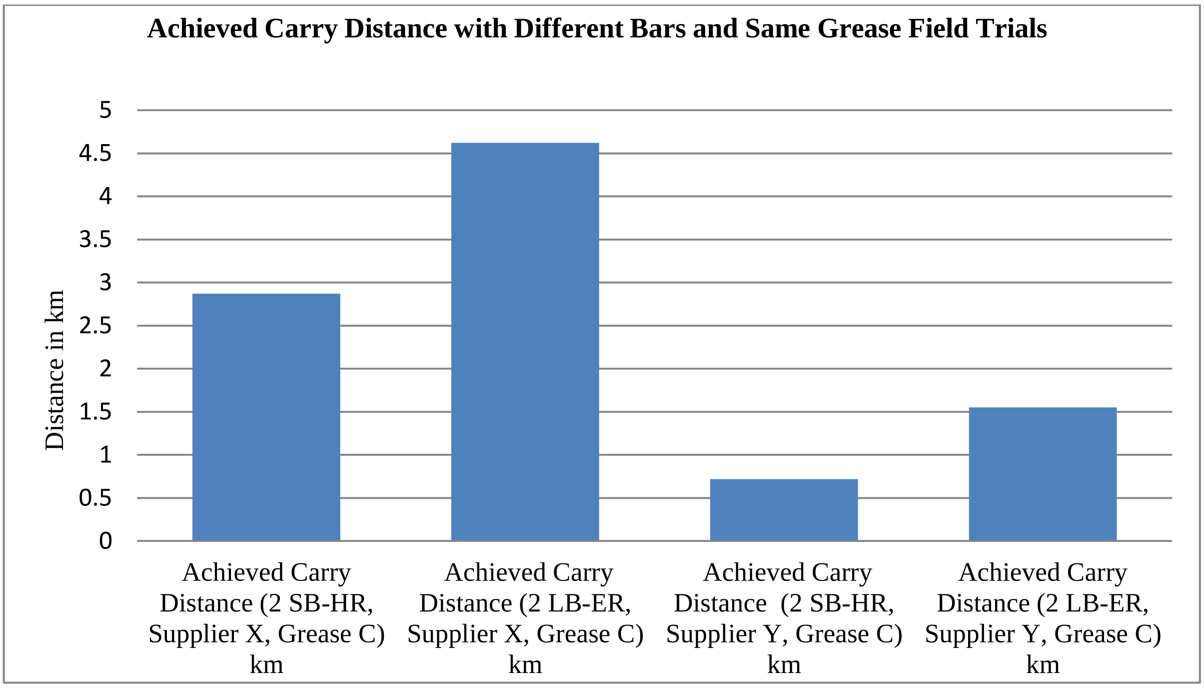



Figure 5.6: Grease carry distance performance of long bars and short bars (CRC Australia 2014)

The grease carry distance performance has also been compared for different combinations of long bars and short bars as shown in Table 5.8.

Table 5-8: Grease carry distance performance for different lubricating bar combinations (Uddin et al. 2011b; CRC Australia 2014)

| Curve Direction | Curve No From Lubricator | ACOF GF-Hi (2 SB-ER, X Grease A)0.34 kms | ACOF GF-Hi (2 LB-ER, X Grease A) 0.33 kms | ACOF GF-Hi (2 LB-ER, X Grease C) 4.623 kms | ACOF GF-Hi (2 LB-ER, X Grease D) 2.65 kms | ACOF GF-Hi (2 LB-ER, X Grease B) 2.96 kms | ACOF GF-Hi (2 LB-ER, X Grease E) 1.28 kms | ACOF GF-Hi (2 LB-ER, Y Grease C) 1.55 kms | ACOF GF-Hi (2 SB-HR, X Grease C) 2.87 km |
|---|---|---|---|---|---|---|---|---|---|
| R | 2 | 0.19 | 0.18 | 0.2 | 0.24 | 0.17 | 0.2 | 0.24 | 0.2 |
| L | 3 | 0.26 | 0.31 | 0.16 | | 0.2 | 0.2 | 0.22 | 0.16 |
| R | 4 | | 0.34 | 0.19 | 0.17 | 0.2 | 0.33 | 0.26 | |
| R | 5 | 0.26 | 0.28 | | 0.24 | | 0.32 | | 0.2 |
| R | 6 | 0.34 | | | | | | | |
| R | 7 | | | | 0.24 | | | | |
| L | 8 | 0.33 | 0.33 | | | | | 0.22 | 0.24 |
| R | 9 | | | 0.23 | 0.41 | 0.23 | | | 0.2 |
| R | 12 | | | | | 0.34 | | | |
| R | 14 | | | 0.24 | | | | | |
| R | 18 | | | 0.35 | | | | | 0.35 |
| R | 19 | | | 0.38 | | | | | 0.29 |

The wheel flange and its contact with the rail are a critical component of the lubricant transport mechanism. The success of an effective lubrication strategy depends on the transport mechanism (Thelen & Lovette 1996). Location and height of applicator bars were highly important to the success of the wheel pick up of grease and its carry distance down the track. The worn wheel gauge shown in Figure 5.7 is used to determine the precise bar height so that the wheels can pick up the maximum amount of grease from the grease beads.

Figure 5.7: Bar height adjustment (CRC Australia 2014)

#### *5.4.2.3 Grease Application Rate*

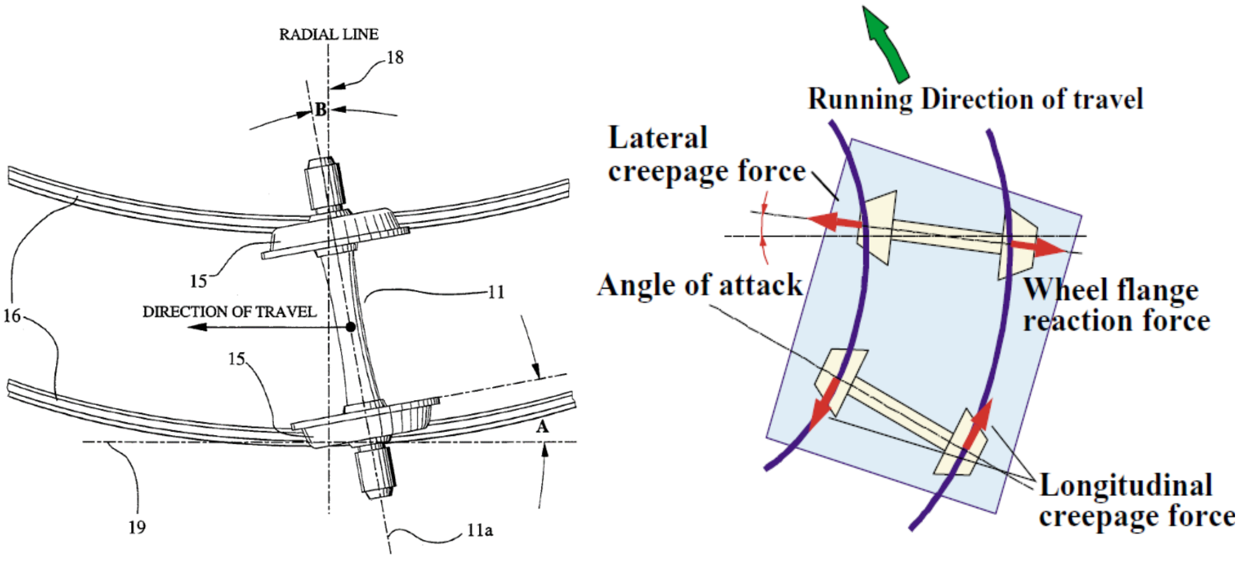
The application rate and interval between applications of grease is controlled by the electronic control box. The optimal setting for grease application rate is achieved by a ‘splash test’ as described in Chapter 3 “Design of Experiment & Methodology”. It starts with the manufacturer’s recommended settings and then reaches an optimal setting after a trial and error process based on the conditions of splash and head contamination with grease. Once the optimal setting is achieved, a waiting period of around a week (depending on traffic density) is applied to reach a steady state of uniform grease distribution at the gauge face. Then tribometer testing needs to be conducted to determine if a stable level of coefficient of friction has been achieved so that the performance of the applicator bars can be properly evaluated. Table 5.9 shows the recommended pump settings for each applicator bar set-up and grease type. Within optimal settings, each lubrication configuration was observed for the duration of more than 50000 wheels passing through the site for each test segment and it was found that very little grease has been scattered away from the rail gauge face.

Table 5-9: Optimal grease application rate/pump for different bars and types of grease, achieved by splash test (CRC Australia 2014)

| Type of Bars | Name of Test Grease | Pump Activation | | Wheel Count | Application Rate/Pump (gm) (as per pump activation parameters) |
|---|---|---|---|---|---|
| | | For Seconds | Wheel Frequency | | |
| Short | A | 0.2 | 18 | 25195 | 5.08 |
| Long | A | 0.25 | 12 | 124500 | 6.80 |
| Long | C | 0.25 | 12 | 359497 | 6.80 |
| Long | D | 0.25 | 12 | 360489 | 6.80 |
| Long | B | 0.25 | 12 | 418981 | 6.80 |
| Long | E | 0.25 | 12 | 518109 | 6.80 |
| Short | C | 0.20 | 18 | 500956 | 5.08 |

When a wheel passes through the short bar applicator sites in the transition spiral of the curve, the grease transport mechanism was severely affected by wheel motion and alignment with the rail gauge face. The planetary dual motion of wheels causes severe sliding or flanging between rail and wheel. The rotary motion of wheels in curves takes place at an angle with the rail's longitudinal axis (angle of attack), whereas the rotary motion of wheels at tangent lubricator sites is parallel to the rail axis and does not cause severe sliding and flanging as occurs at short bar sites.

Due to the angle of attack and the bogie steering capability in curving, wheels, instead of travelling parallel to the rail, travel obliquely at an angle with the rail. The angle between the tangent of the rail and the direction of wheel rotation is equivalent to the angle of attack or, according to McGuire et al. (2014), the angle of attack is the angle taken by the axle relative to the direction of motion as shown in Figure 5.8.




(a) (b)

Figure 5.8: (a) Angle-of-attack in the curve (Izbinsky et al. 1994), (b) Angle-of-attack with active forces in the wheel/rail interface (Ishida & Aoki 2004)

Hanson et al. (2014) reported that the angle-of-attack of a leading wheelset with good steering performance can be calculated as follows:

Angle-of-Attack (AoA) = Wheelbase/ Curve Radius

This angle of wheel rotation causes the grease to be scattered away from the longitudinal axis of the rail towards the field side and gauge side of the track. Another issue is that, when a wheel travel fast with an angle of attack, it squeezes the grease bead out and wastes grease around the short bars. Figure 5.9 demonstrates the clear indication of the direction of grease when the wheels strike the grease beads when passing over the applicator bars at the short bar applicator site. The direction of grease shows that there is no tendency of grease to travel towards the gauge corner, but rather it travels toward the centre of the track and even reaches to the low rail side of the track.

Figure 5.9: Short bars applicator site on the transition spiral of curve, the direction of grease movement when wheels strike on grease beads and the bead shape (CRC Australia 2014)

If there is no hunting proper site selection, positioning can significantly reduce the amount of grease wastage and improve the amount of grease being transferred and transported from the long bar sites. Figure 5.10 shows that the wheel travel direction was closely parallel with the rail tangential axis and least amount of grease wastage has been recorded.

Figure 5.10: Long bar applicator site showing grease bead and its shape (CRC Australia 2014)

If a significant portion of grease is lost on site, very little remains for friction control. Grease fling off may vary based on the grease bead size and quality. Short bars have a bigger bead size than long bars, which causes more wastage of grease compared to long bar units.

Evaluation of applicator bars can be undertaken with a properly designed experiment and real life field trials. For a proper evaluation, an experiment design with a matrix of lubricant types and lubricating bar applicator types can be used to obtain an assessment of various combinations. These can be categorised as discussed below.

There could be several combinations tested such as a specific type of bar used for different greases and observe the results, then use another type of bar and the same greases. These tests can be performed under controlled operating parameters and their specific impacts can be recorded.

Using informed judgements, it is then possible to quantify the decision parameters for each type of applicator bar based on the carry distance achieved under the standard placement model. In the field tests, the best combination of 2 long bars on each rail with Grease C has achieved the longest grease carry distance and an applicator bar factor of 1 was achieved. In this research any other combination of the applicator bars and grease has achieved shorter grease carry distance as compared to combination of 2 long bars on each rail and grease C. Therefore for all other combinations, the applicator bar factors were less than 1. The applicator bar factor, P for any configuration of applicator bars is considered as grease carry distance of the specific configuration divided by the achieved longest grease carry distance.

### 5.4.3 Grease Performance Factor, G

Grease quality has a significant influence on successful lubrication and distribution along the rail. It should have the necessary properties to prevent it from running off the gauge face within a short distance or after a small number of wheel load cycles. Lubricant should be sustained at the wheel/rail interface irrespective of load and weather conditions.

If grease liquefies and migrates away from the contact area, then the likelihood of lubrication system failure is high. The service life of grease is often determined by the eventual loss of its semi-solid consistency to become either a liquid or a hard deposit. Metallic soap thickeners are responsible for grease retentivity, and heat, water and extreme load resistance which are some of the properties that determine the quality of the grease.

Sroba et al. (2001) reported that field testing would be required to rank prospective lubricants in terms of their effectiveness. Based on field data analysis and the study of grease properties, it has been shown that laboratory tests can provide information on grease rheology, flow characteristics and behaviours, but to determine the best performances and evaluate trade-offs, grease selection field trials are inevitably necessary. Final evaluation cannot be achieved without field test results. Table 5.10 shows the most properties of the test greases (A, B, C, D and E) based on their technical data sheet parameters and their achieved carry distances with two long bar applicators on each rail of tangent track. Table 5.10 shows that some data is missing, because it was not available in the supplier’s product data sheet from where it was collected. As stated earlier it was also out of the scope of this research to investigate the detailed properties of grease or their standard test.

Table 5-10: Test grease specifications and carry distances with two long bars on each rail of tangent track

| Technical Data | | | |
|---|---|---|---|
| Grease A | | | |
| Properties | Data | Test Method Standard | Achieved Carry Distance |
| Appearance | Greyish black grease | | 0.33km, with 2 long bars on each rail |
| Lubricating Solids | Graphite | | |
| Thickener | Lithium | | |
| NLGI Classification | 1 | | |
| Base Oil | Mineral Oil | | |
| Base Oil Viscosity at $40^0$C (cSt) | 150 | ASTM D445 | |
| Specific Gravity | 0.9 | | |
| Drop Point in $^0$C | 190 | ASTM D-566(IP132) | |
| 4 Ball Weld Load in kg | 550 | ASTM D2596(IP 239) | |
| Flash point in $^0$C | Greater than $200^0$ C | ASTM92 | |
| Flammability | Non Flammable | | |
| Consistency | 310-340 $mm^{-1}$ | ASTM D217 | |
| Temperature Range (continuous) | $-10^0$C to $150^0$C | | |
| Grease C | | | |
| Appearance | Dark Grey, Tacky | | 4.623km with 2 long bars on each rail |
| Lubricating Solids | Molybdenum disulphide (wt %) 3 | | |
| Thickener | Lithium soap | | |

| | | | |
|---|---|---|---|
| NLGI Classification | 2 | | |
| Base Oil | Highly refined base oils, a special EP additives package and 3% molybdenum disulphide | | |
| Base Oil Viscosity at $40^0$C (cSt) | 220 | D 445 | |
| Base Oil Viscosity at 100 $^0$C (cSt) | 15 | D 445 | |
| Specific Gravity | | | |
| Drop Point in $^0$C | 176.66 | Mettler | |
| 4 Ball Weld Load in kg | | | |
| Flash point in $^0$C | | | |
| Flammability | | | |
| Consistency | | | |
| Temperature Range (Continuous Service) | 121 $^0$C | | |
| Temperature Range (Short Exposure) | 176.66 $^0$C | | |
| Penetration Worked, 60X | 265-295 | | |
| Rust Protection | Pass | D 1743 | |
| Copper Corrosion | lb | D 4048 | |
| Timken, OK loads, lbs | 30 | D 2509 | |
| Four-ball EP Load Wear index, kgf | 46 | D 2596 | |
| Four-ball Weld Point, kgf | 250 | D 2596 | |
| Four-ball Wear, mm (1 hr, $75^0$C, 1200 rpm, 40 kgf) | 0.4 | D2266 | |
| Grease D | | | |
| Appearance | Black | | 2.65km with 2 long bars on each rail |
| Lubricating Solids | Graphite | | |
| Thickener | Microgel | | |
| NLGI Classification | 1 | | |
| Base Oil | | | |

| | | | |
|---|---|---|---|
| Base Oil Viscosity at $40^0$C (cSt) | 220 | IP 50/ ASTM D445 | |
| Specific Gravity | | | |
| Drop Point in $^0$C | 260 | IP 396 | |
| 4 Ball Weld Load in kg | | | |
| Flash point in $^0$ C | | | |
| Flammability | | | |
| Consistency | | | |
| Temperature Range (continuous) | | $-35^0$C to $80^0$C | |
| Penetration Worked, 60X | 340 | IP 50/ ASTM D217 | |
| Low Temperature Pumpability (2) kPa/m (psi/ft) @ $-40^0$C | 3800 (168) | | |
| Grease E | | | |
| Appearance | Smooth grey grease | | |
| Lubricating Solids | | | |
| Thickener | Lithium | | |
| NLGI Classification | 2 | | |
| Base Oil | | | |
| Base Oil Kinematic Viscosity at $40^0$C (cSt) | 680 | | |
| Specific Gravity | | | |
| Penetration Worked, mm/10 | 290 | ASTM D 217 | 1.28km with 2 long bars on each rail |
| Drop Point in $^0$C | Greater than $200^0$C | ASTM D 5661 | |
| Timken, OK loads, kg | 20.5 | | |
| 4 Ball Weld Load in kg | 420 | IP 239 | |
| Flash point in $^0$C | Not available | | |
| Flammability | Not available | | |
| Consistency | Not available | | |
| Temperature Range (continuous) | Not available | | |
| Rust Protection | Pass | ASTM D 1743 | |

| Grease B | | | |
|---|---|---|---|
| Appearance | Amber | | 2.96km with 2 long bars on each rail |
| Lubricating Solids | | | |
| Thickener | Lithium 12-Hydroxystearate | | |
| NLGI Classification | 2 | | |
| Base Oil | | | |
| Base Oil Kinematic Viscosity at $40^0$C (cSt) | 205-235 | D445 | |
| Base Oil Kinematic Viscosity at $100^0$C (cSt) | 15.6 min | D445 | |
| Specific Gravity | | | |
| Penetration @77.F W 60 | 280 | | |
| Drop Point in $^0$C | 182 | | |
| Timken, OK loads, kg | 45 | D 2509 | |
| 4 Ball Weld Load in kg | | | |
| Flash point in $^0$C | 226 | | |
| Flammability | | | |
| Consistency | | | |
| Temperature Range (continuous) | | | |
| Rust Protection | Pass | D- 1743 | |
| Copper Strip, 24 hr | 1B | D- 130 | |

### *5.4.3.1 Test Grease Performance Evaluation based on Grease Properties*

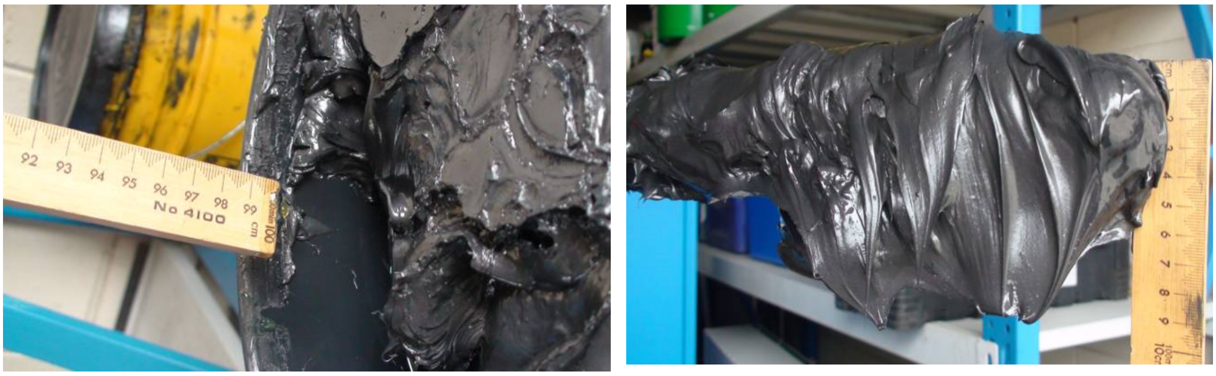
There are several properties that assist with grease selection for a desired application. These properties are not only based on meeting the operating parameter requirements of an application, but also meeting other characteristic requirements. Most grease properties can be measured using standard ASTM test procedures.

##### *5.4.3.1.1 NLGI Classification and Consistency of Test Greases*

Consistency of grease is measured by its penetration number, popularly known as the NLGI grade. Out of the five test greases, four were NLGI Grade 2, and only one was NLGI Grade 1. Throughout the test observations and data collection, it was shown that the NLGI Grade 1 grease (Grease A) demonstrated extremely poor performance and did not achieve a significant carry distance with any combination of equipment, either with long bar or short bar applicators. Chapter 3 and Chapter 4 have elaborated on the evidence in the field as to how the grease had burnt out and/or moved towards the rail head and had fallen off from the rail gauge face. Due to the absence of grease and the lack of lubricity, the coefficient of friction exceeded the accepted limit within a few hundred metres past the lubricator and left the rest of the track dry/unprotected. With reference to Table 5.10, the National Lubricating Grease Institute has established the NLGI numbering system for grease consistency, ranging from 000 to 6 which corresponds to a specified range of penetration numbers (Obiedo 2012). Grease A has the lowest consistency number amongst all the tested greases.

On the other hand, NLGI Grade 2 greases, having a higher consistency, have shown a significantly better performance in effective lubrication in terms of achieved carry distance. These greases contain additives to give resistance to washout due to rain, and other additives similarly give resistance to bleeding/leaking, making them more effective at high operating pressures and temperatures; these characteristics are referred to as Extreme Pressure (EP) properties of the grease. It is evident that the NLGI Grade 1 grease has a low viscosity base oil which gives poor retention and consistency, thus giving poor lubrication performance at high loads and hence being judged as not suitable for heavy haul rail applications. Such greases could perform better in cold climates because the viscosity of the base oil is higher at low temperatures.

Excessive oil separation or bleeding (where lubricating oil separates from thickeners) causes the loss of lubricity of the grease. The continuous existence of grease at the gauge corner is a basic need of effective lubrication. Field tests revealed the tendency of low consistency grease to simply migrate from the gauge corner towards the bottom of the gauge face within a very short distance from the lubricators. Figure 5.11 shows that, when grease is stored at ambient temperatures for a few days, this causes significantly greater oil separation in Grease A compared to the higher consistency Grease C.




(a)

(b)

Figure 5.11: Oil separation and tackiness comparison of Grease A and Grease C used in field tests: (a) Severe oil separation and low tackiness of Grease A, (b) Negligible oil separation and high tackiness of Grease C (CRC Australia 2014)

##### *5.4.3.1.2 Viscosity and Dropping Point of Test Greases*

According to ASTM D-566 "This is the temperature at which a drop of fluid forms and falls from grease under test conditions established by the ASTM standard. The limiting temperature for prolonged exposure is well below the Dropping Point" (Booser 1983). In Table 5.10, The Base Oil of Grease A has the lowest viscosity at $40^0$C (150cSt) and the grease has the lowest dropping point as compared to all other NLGI Grade 2 greases tested. No data was available for the viscosity of Base Oil of Grease A at $100^0$C. The Base Oil of other test greases have significantly higher viscosity at $40^0$C. Test observations showed that NLGI Grade 2 greases with higher viscosity and higher dropping point achieved remarkably better carry distances compared to the NLGI Grade 1 grease with low viscosity and low dropping point. Low viscosity and lower dropping point may significantly contribute to the disappearance of the grease in the curves, and the quick burn out of lubricant to dry graphite that ends up at the foot of the rail. Though the tested NLGI Grade 2 greases showed considerable variation in the achieved carry distances, they significantly did stick to the rail gauge corner contact surface and provided remarkable protection of the rails with effective lubrication. The trials also showed that grease with higher viscosity base oil (Grease E, base oil kinematic viscosity 680cSt at $40^0$C) achieved a lower carry distance with effective lubrication compared to the best grease (Grease C, base oil viscosity 220cSt at $40^0$C). Field trials showed that optimal grease selection should be done based on grease properties and their field test performance. The data showed that an optimum viscosity of the base oil in the grease is important. If viscosity is too low such as the NLGI Grade 1 test grease, the oil is not retained at the contact interface surface and, due to high contact pressures and temperatures, viscosity drops to a limit such that the lubricant disappears quickly and no protective film is formed. In the case of too high a viscosity, lubricant does not flow with the wheel motion and hence distribution along the track is not possible causing a dry wheel/rail interface.

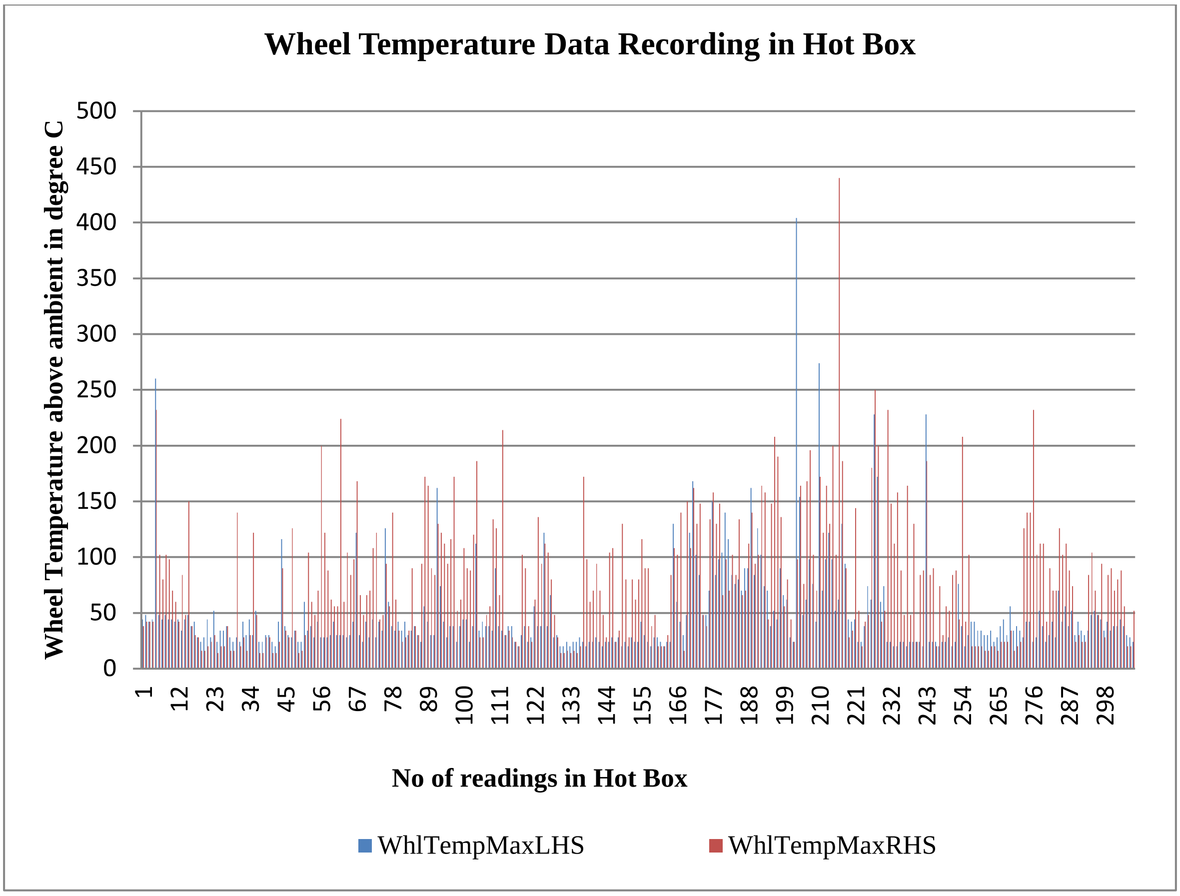



Figure 5.12: Wheel temperature data from hot wheel detector readings on QR network

Figure 5.12 shows the temperature data of heavy haul train wheels operating on the QR network as detected by wayside hot box sensors. The chart shows the wheel temperature data as the rise above ambient temperature. Therefore, the actual temperature of the wheels is around $35^0$C plus the temperature shown in the chart. The chart clearly indicates that there are significant numbers of wheels that have temperatures 100 to $300^0$C above the ambient temperature i.e. actual temperatures of 135 to 335$^0$C. All of the trains have 50 to 110 wagons and operate at speeds up to 110km/hr. It was also observed that grease with highest viscosity base oil sustained in the rail wheel interface and might not achieved the longest carry distance, though it appears to give a longer carry distance. These observations indicate a full scale field trial in normal operation is necessary to characterise grease for its lubrication performance.

#### *5.4.3.1.3 Mechanical Stability of Test Greases*

The trial also showed that Grease A has very low mechanical stability compared to other test greases. Due to pumping pressure and resistance from the grease delivery hose and the internal flow path of the applicator bars, the grease is weakened physically. As soon as Grease A has been delivered through the grease applicator ports of the applicator bars, the grease beads collapsed from the rail gauge face and spread on the fall protection bar (Figure 5.10). It was observed that most of the grease was wastage from the wheel strike and contaminated the tracks and ballasts. Also found was very little carry distance achieved due to the very little picked amount of grease and also amount of fallen off the wheel flange and the rail gauge face. A detailed investigation on the grease properties and its detail standard methodology of test performed by suppliers was not established. The term Mechanical Stability was focused on the grease physical stability on the grease port and sustaining in the gauge corner along the rail. The best grease, Grease C, and the other NLGI Grade 2 test greases have very high mechanical stability and maintained a strong grease column at the delivery ports of the applicator bars, hence very little had fallen off.

#### *5.4.3.1.4 Comparison of Greases*

The comparison of the best and worst grease has been carried out in this analysis. Grease C is type of lithium soap greases enhanced with a specifically formulated package to meet the pumpability, adhesion and load carrying capacity of rail curve grease. It has improved lubricity and durability to maintain performance under extreme conditions of a hot and humid climate, high pressures and high operating temperatures at the wheel/rail contact surfaces. It also offers excellent resistance to rust and corrosion. This lithium soap thickened grease is made of highly refined base oils, a special extreme pressure (EP) additive package and 3% molybdenum disulphide. The molybdenum disulphide acts to enhance anti-wear and load carrying properties, which are critical for the heavy haul railway application. A highly shear stable tackifier improves grease adhesion to rail and wheel surfaces and prevents oil bleeding.

It also provides good resistance to the mechanical shear which takes place within lubricator operations.

Grease C demonstrated the following characteristics with all test equipment configurations:

- Suitable tackiness.
- Good pumpability.
- High retentivity.
- Mechanical stability.
- High dropping point.
- Outstanding lubricity.
- Strong water washout resistance.
- No oil separation’
- Excellent carry distance.
- Biodegradability and non-toxicity.
- Inclusion of extreme pressure (EP) and anti-wear resistance additives.

On the other hand, Grease A contains solid graphite in mineral oil with lithium as a thickener. No information regarding any special additives is available on this grease. Regardless of the technical specifications in the Technical Data Sheet for this grease, the field tests found it to be the worst performed grease. It never produced an effective friction level on the rail gauge face and also did not get carried along the rail gauge face for any long distance. Chapter 4 has demonstrated the performance data available from this grease. Grease A showed the worst performance with all of the configurations of equipment and lubricator bars in the tests.

#### *5.4.3.1.5 Grease Application Rate & Grease Carry Distance*

Currently there are no specifications or guidelines for selection of the best grease for heavy haul railway application. Tests were conducted using both long and short applicator bar technology with necessary control box settings for equal distribution to the wheels. The track was allowed to dry for 3 days (approximately 30,000 wheel passes) and measurements were made with the tribometer to ensure that curves adjacent to the test location were dry. The grease tank was then filled with the test grease and the unit was run for approximately

40,000 to 60,000 wheels with the splash test set up. The tribometer was then run from the lubricator test site to the location where the COF was greater than 0.25 on the gauge corner.

Appropriate grease application rate settings were decided based on:

- Proper priming of the grease through the delivery hose to make sure there was no cavitation or blockage in the suction and discharge of the pump and hose. It has been observed that at the very low level of grease in the grease reservoir tank pump run dry and significant drops in the motor current. Ultimately the blockage to the grease flow in the pump suction or the low level of grease in the tank cause pumping air and with reduced amount of grease or no grease and cause air bubbles in the discharge side of the pump and up to the grease delivery port of the applicator bars.
- Severity of splash of the grease on the test site from splash test.
- Ensuring effective bar height to get the grease onto the wheel; there are limitations to being able to achieve the exact bar height for effective set-up.
- Splash observation and grease volume measurements for each test. The volume of grease was observed and reviewed based on the splash condition and was not same for all the grease from Grease A to Grease E.
- Changing the bar height if needed and then again conducting the splash test and measuring the grease volume needed.
- Measuring coefficient of friction with tribometer and determining the grease carry distance.
- Review the application settings again.

It can be concluded that different greases may have different performance levels in achieving carry distance when they have been tested with the same configurations of equipment. Table 5.11 shows the performance of the different greases in field trials with the same lubricator and applicator bars combination.

Table 5-11: Carry distance performance of greases using the same applicator bars

| Bar Combination | Grease | Achieved Carry Distance (km) |
|---|---|---|
| (2+2) Long bars, Supplier X | A | 0.330 |
| (2+2) Long bars, Supplier X | B | 2.968 |
| (2+2) Long bars, Supplier X | C | 4.623 |
| (2+2) Long bars, Supplier X | D | 1.650 |
| (2+2) Long bars, Supplier X | E | 1.280 |

Figure 5.13 (a) shows the performance of different greases with the same applicator bars in field trials. Grease carry distance of each grease is different for the same applicator bars and operating conditions. Achieved carry distances show that it is extremely important to identify the best grease for effective lubrication using field trials. Figure 5.13 (b) shows that different greases have different rates of change of the level of friction within the achieved carry distance and beyond. The COF for the worst grease, Grease A, jumped above the acceptable value of 0.25 after the first curve from the lubricator site. However, the best grease, Grease C, showed a steady rate of change of COF up to Curve 14 while staying below the acceptable value of 0.25. The achieved carry distance was 4.63km and the grease coverage on the rail surface was highly effective and well established. No sign of burn or fallen grease was identified at the rail foot. Excellent thick, black grease coverage was clearly visible up to Curve 14 which confirms the performance of the grease. This type of grease should be acceptable for economic and effective lubrication which can save an enormous amount of rail and wheel maintenance costs and rail renewal costs.

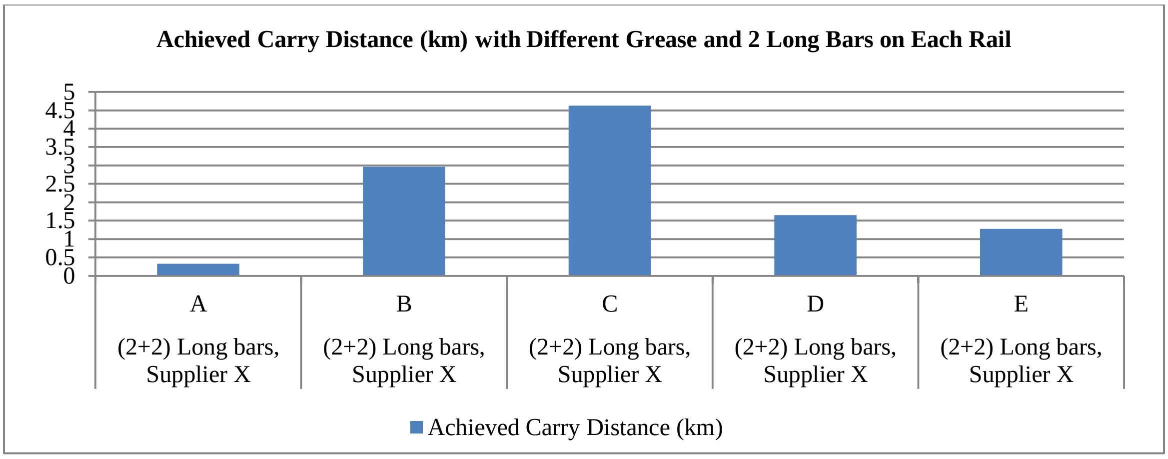



Figure 5.13 (a)

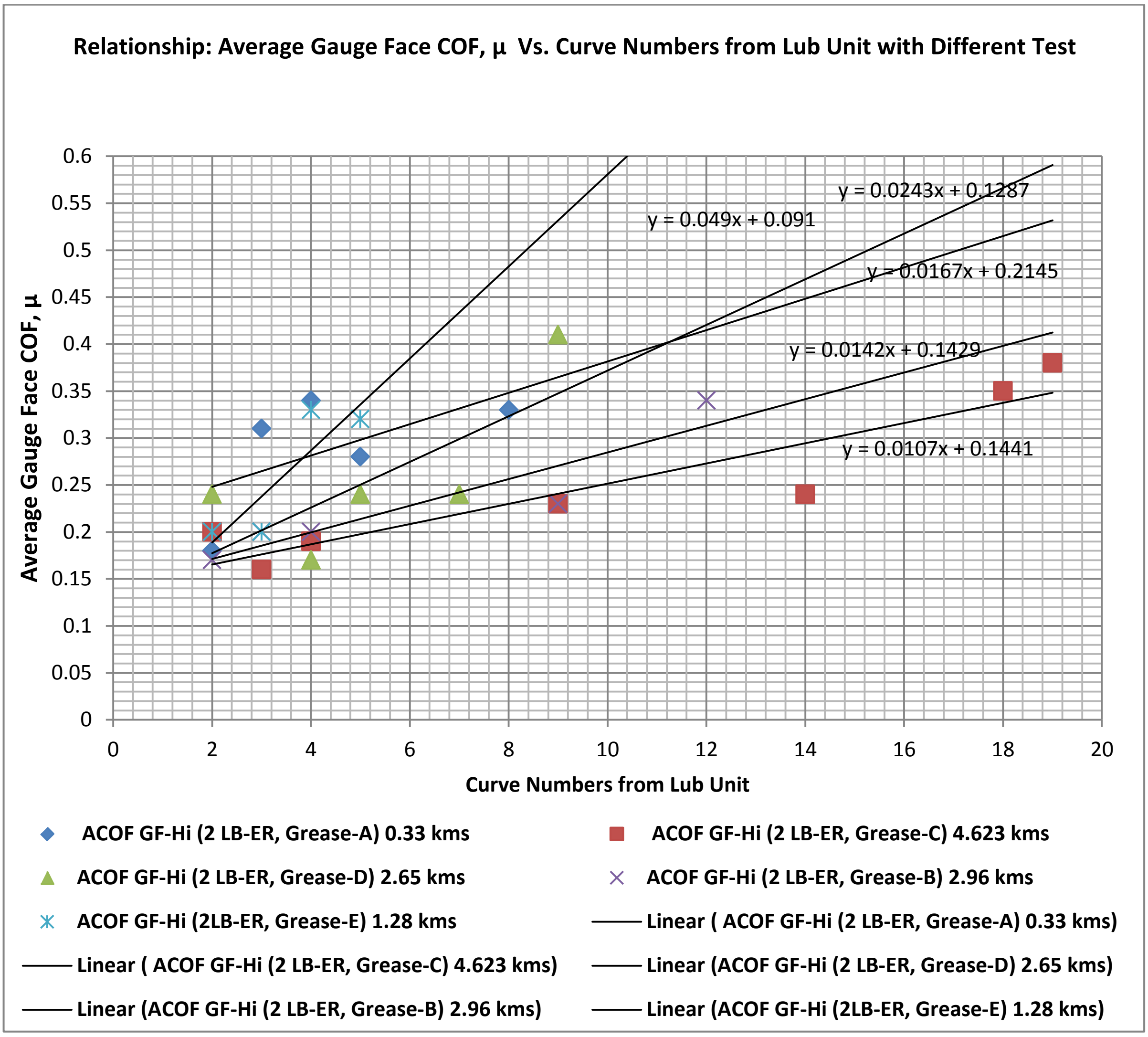



Figure 5.13 (b)

Figure 5.13: (a) Grease carry distance with two long applicator bars (CRC Australia 2014), (b) Change in coefficient of friction using two long applicator bars (Uddin et al. 2011b; CRC Australia 2014)

The quality of grease and applicator bars plays a significant role in achieving acceptable carry distances. A matrix has been developed as shown in Figure 5.14 to define the configurations of grease and applicator bars which show the combined effect in achieving carry distance.

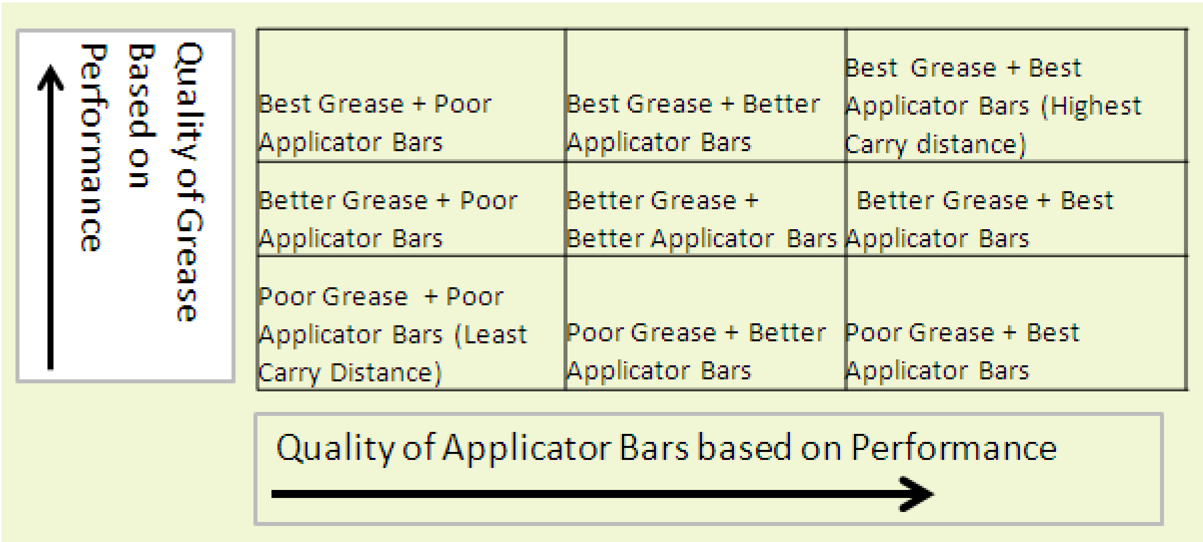



Figure 5.14: Grease and applicator bars matrix

Best applicator bars refer to the applicators bars which have achieved the longest carry distance with each of the test greases. In this research the best configuration of equipment was two long bars on each rail from Supplier X. The long bars' placement location was on the tangent track just before a rail curve.

Grease factors need to be determined based on the achieved carry distance which should be evaluated from field trials. The combination of applicator bars and grease identified as the best performer and their data is required to be used for developing the lubricator placement model for a specific track section or network.

In this research, based on all field trials and results, Grease C shows the highest carry distance in combination with 2 long applicator bars on each rail. Therefore, the grease factor for Grease C has been considered to be 1 for the configuration of any grease with two long bars on each rail (2LB-ER) as this configuration achieved the longest carry distance compare to any grease with the same applicator bar configuration of two long bars on each rail (2LB-ER). For the evaluation of grease factor it is considered to compare the different grease performance with the same configuration of applicator bars.

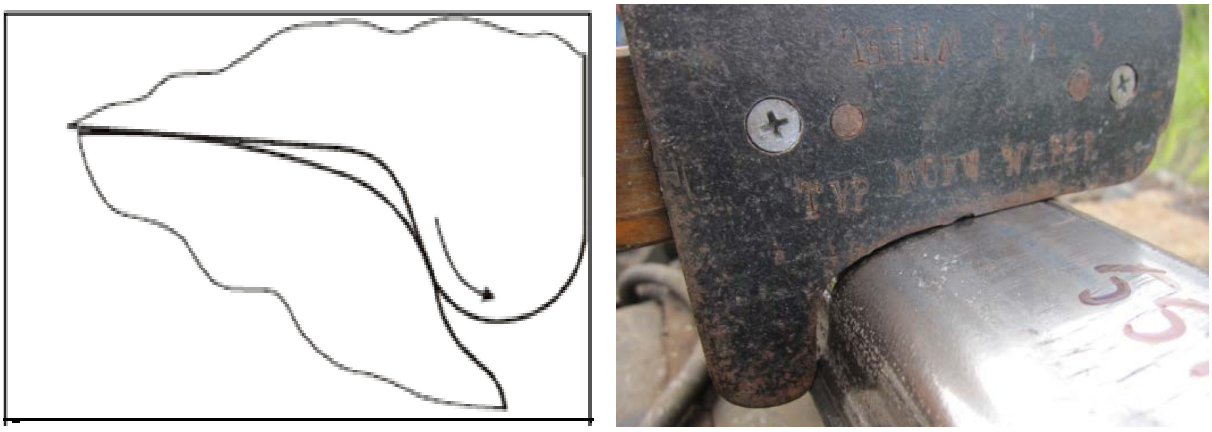
Grease factor G, can be defined as carry distance of the selected grease divided by the carry distance of the best grease when using the same configuration of equipment. To cite an example; Grease factor, G for Grease A with two long bars on each rail (2LB-ER) is (0.33/4.623) or 0.07.

### 5.4.4 Wheel/Rail Profile Factor, $P_{prof}$

Wheel/rail contact patterns play a significant role in grease transport from curve to curve. Depending on the material, age and wear conditions, rails and wheels may have a different contact pattern from curve to curve. Contact between the gauge corner of the rail and the wheel flange is typically considered to be two-point, single-point or conformal contact. Two-point contact causes severe rail gauge face and wheel flange wear and single-point contact is associated with rolling contact fatigue. Conformal contact ensures the largest possible contact area and thus decreases the contact stress (Frohling 2007). Conformal contact is an optimum condition for non-steering vehicles and supports lubrication, whereas two-point contact often cuts any lubricating film applied to the contact zone (IHHA 2001). The impact of the type of contact occurring at the wheel/rail interface needs to be well understood in relation to lubrication effectiveness.

#### *5.4.4.1 Type of Wheel/Rail Contact*

According to McGuire (2014), the wheel/rail interface COF is the genesis of wheel and rail wear and energy consumption. Furthermore, contact geometry keeps changing as wear progresses. From the rail lubrication point of view, two-point contact (Figure 5.15) applies grease from wheel flanges to the gauge face contact points. Therefore, the gauge corner remains unprotected. If a significant proportion of wheels have two-point contact with the rail, then there may not be a sufficient amount of grease to provide effective gauge face lubrication. In this case, grease remains in the wheel flange throat and may travel towards the flange root where it may be wasted.



(a) Two point contact (Frohling 2007) (b) On test curves

Figure 5.15: Two point contact

Two-point contact only has scope to lubricate just a thin slice of the rail surface at the contact point on the lower gauge face. During curving, lateral creep forces and flange forces at the gauge face contact points causes burning of applied grease and the rest goes to waste. After being burnt off, there is no more grease left to be carried further along the rail as in Figure 5.15b. It is anticipated that, at these points, extreme pressure occurs and there is little chance of boundary lubrication being sustained, resulting in direct surface to surface contact.

In the case of Grease A, field tests show that there is only dry graphite left on the gauge face contact band instead of grease, and lumps of grease were observed at the bottom of the rail head. Sharp two-point and single-point contact both have extremely adverse effects for lubrication effectiveness. As the contact area is far smaller than under conformal contact, the contact stress in two-point and single point contact is extremely high, to the extent that temperature rise is also high and only burnt graphite is left with no base oil remaining. In both cases grease is burnt off faster than any other contact condition. Single-point contact is probably the most damaging to both rail vehicles and track. The high contact stresses occurring under high creepage result in gauge corner fatigue. It is associated with high longitudinal creepage which causes rail material flow that leads to vehicle instability (IHHA 2001). Therefore, in both single and two-point contact, not enough grease is applied on the gauge face in the subsequent curves and the gauge corner remains dry.

In conformal contact as shown in Figure 5.16, grease is spread on a larger area at the gauge corner where lubrication is necessary; the contact stress is lower than any other type of contact and this supports the grease retentivity at the gauge corner and reduces the burning of grease.


Figure 5.16: Conformal contacts on test curves

The relationship between the contact forces in different types of contact and grease sustainability can determine the effectiveness of lubrication under the extreme conditions at the wheel/rail interface at the gauge corner. Based on the observation in the field trials the conformal contact shows the most effective grease distribution along the curve and has a more effective grease coverage on the rail profile compare to single point or two points contacts grease distribution. No evidence of burnt grease has been found in the curves where conformal contact was observed for most of the trialled grease though some particular grease might burnt out even in the conformal contact. It has commonly been observed burnt grease or lump of grease on the rail feet in the curves where the single point or two point contacts was observed. These contact pattern reduce the grease exchange in between rail and wheel and may significantly reduce the contribution to transport grease towards the further curves. Wheel rail profile factor $P_{Prof}$ for conformal contact, single point contact and two point contact needs further study to quantify appropriately.

### 5.4.5 Equivalent Length of Curve, (C+S)

Coefficient of friction values in each curve have a linear relationship with a low standard deviation. Therefore, with good quality grease the COF within each curve length remains almost the same.

However, COF values for different curves have significant variation. COF and ACOF values within different curves show that the values have an upward linear trend.

Equivalent curve length for each curve can be given by:

$$Ceq = C + S \tag{5.3}$$

where

$C_{eq}$ = Equivalent Curve Length

S = 2.5% of Tangent Length before Curve + 2.5% of Tangent Length after Curve

Travelling from tangent track into a curve through the transition spiral and from the curve towards tangent track, wheels travel on differential radii which causes dynamic condition changes for the vehicle. While a train is exiting from a curve, it regularly generates a pattern of lateral movement which depends on various track and traffic conditions such as speed, friction, rail and wheel condition, bogie type, load condition, track alignment, gauge width and curve and tangent length. This effect is known as hunting. Therefore, rail wear has been identified in a portion of the tangent track segment which is considered as an extension of the curve length. Therefore, based on the field experience and literature from other heavy haul networks, a specific percentage of tangent tracks on each side of the curve are added to the original curve length to quantify an 'Equivalent Curve Length', $C_{eq.}$

During hunting, grease on the wheel flanges is wiped off or falls off, which causes adverse effects on effective lubrication. S has been considered as the sum of 2.5% of Tangent Length before Curve and 2.5% of Tangent Length after Curve (de Koker 1994; Sroba et al. 2001). This extension of the actual curve length helps account for the loss of lubrication caused by hunting on tangent track.

### 5.4.6 Radius Factor, R

Radius of curvature has a significant impact on wheel/rail wear and rail degradation. Field investigation shows that unlubricated sharper curves wear out faster than shallower curves. The retentivity of grease is subject to a dramatic reduction in sharper curves compared to shallower curves due to the higher lateral forces and more aggressive contact pattern. Gauge face wear is highly visible in sharp curves compared to shallow curves when the gauge corner is dry or unlubricated. During the field tests, it was observed that sharp curves have more grease burnt and fallen off the gauge corner compared to shallow curves. According to Table 5.12 and Figures 5.17 and 5.18, wear data and corresponding radius data from field investigations show that there is a non-linear relationship between curve radius and wear rate.

Table 5-12: Radius of curve r, wear rate W and degree of curvature $D_c$

| Radius of Curve, r (m) | Wear Rate, W (sq. mm/yr) | Degree of curvature, $D_c$ (°) |
|---|---|---|
| 504 | 56.70 | 3.46 |
| 298 | 60.42 | 5.86 |
| 808 | 52.09 | 2.16 |
| 900 | | 1.94 |
| 1000 | | 1.75 |
| 1200 | | 1.46 |

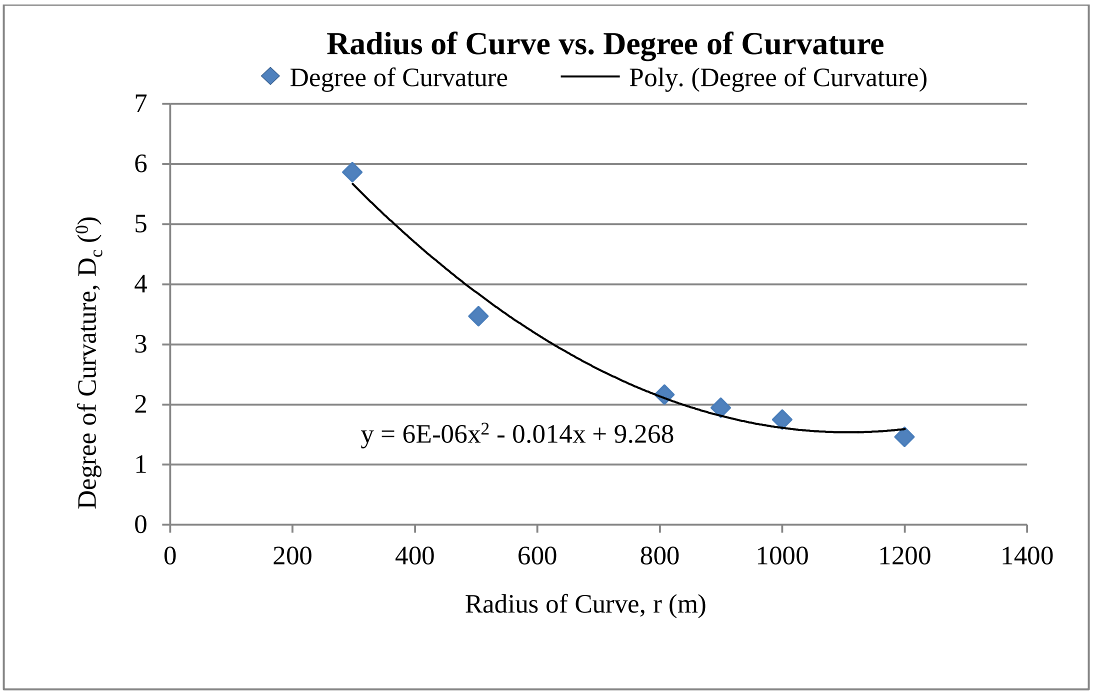



Figure 5.17: Radius of curve, r (m) versus degree of curvature, $D_c$ (°)

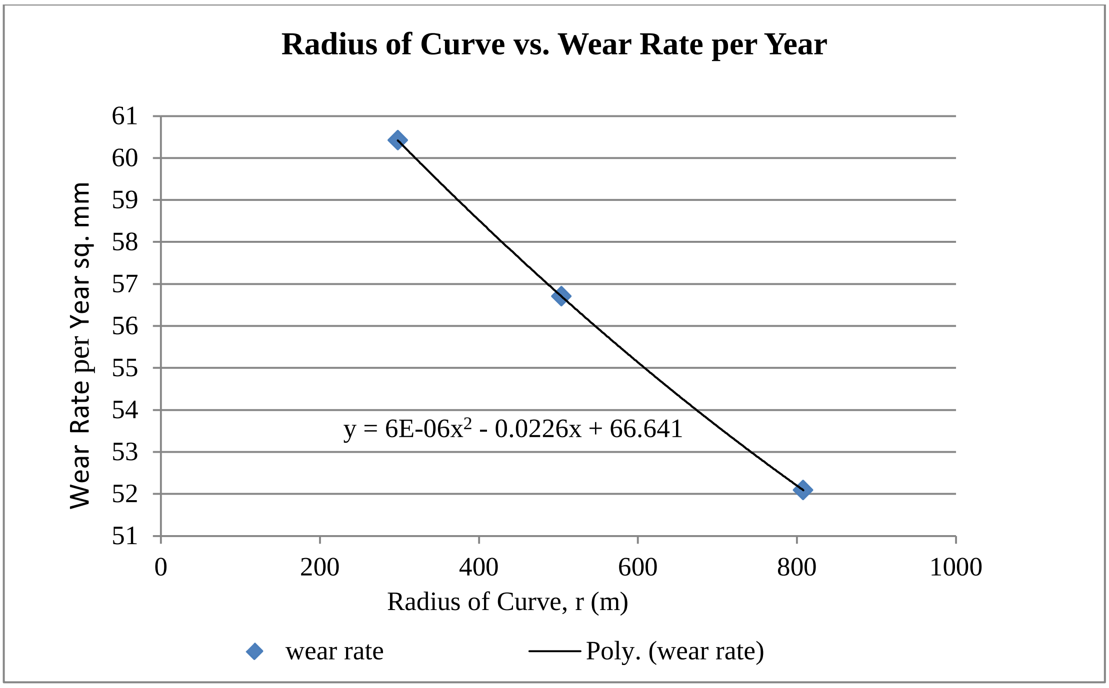



Figure 5.18: Radius of curve, r (m) versus wear rate per year (sq. mm/yr)

Both the Figure 5.17 and Figure 5.18 show that the following variables are significantly related to each other:

- Radius of curves and wear rate on relevant curves.
- Radius of curves and degree of curvature of relevant curves.
- The rate of change of the degree of curvature with respect to radius of curves is very close to the rate of change of wear rate with respect to radius of curves.
- Equations of both the rate of changes follow the same gradient approximately.

IHHA (2001) reports that lateral displacement increases non-linearly when curve radius decreases, which causes high lateral movements in sharper curves and might contribute to increased wear and creep forces.

Wang et al. (2010) reported that, with decreasing curve radius, the wear of rail increased non-linearly, especially for curves below 1200m radius where the rail wear increased rapidly with the decreasing curve radius. Friction in wheel/rail interaction decreased non-linearly with increasing curve radius.

Considering collected field data and the literature available, the degree of curvature of a curve is considered as the Radius factor R. In the definition of degree of curvature $D_c$ = $(36000/2\pi r) = 5729.6/r$, is the central angel subtended by a 100-foot arc. If the SI units are used then the central angel subtended by a 30.5-metre arc. Radius will be in metres. Therefore $D_c = (36000/2\pi r) = (5729.6/3)/r = 1746/r$

Sroba et al. (2001), used the above formula to define the degree of curvature as:

$$D_c = 1746/r \tag{5.4}$$

where:

$D_c$ = Degree of curvature, Arc definition

r = Radius of curve in metres

R is a term to include the effect of curve radius. It has been taken as the average degree of curvature of the curve, including the spirals. QR defines separate curves with no tangent in between as compound curves. However, if the curve is a compound curve with no distinction between the different curvatures, then all circular curve sections and their transition spirals are combined to determine the average degree of curvature.

### 5.4.7 Traffic Factor, T

Bi-directional and unidirectional traffic has a direct effect on carry distance. Rail lubricator wheel sensors sense traffic from both directions and the pumps operate in conjunction with wheel passes. Therefore, for bi-directional traffic, grease carry distance measurements need to be conducted in both Up and Down directions of traffic and one single lubricator unit covers around double the length of track compared to unidirectional traffic.

The suggested traffic factor developed by de Koker (1994) was T = 1 for bi-directional track and 0.5 for unidirectional.

However, Sroba et al. (2001) considered that, if the track has unidirectional traffic, the factor is unity, and for bi-directional traffic, the factor is 2. Rail operators frequently run five or six trains in the same direction before allowing traffic to move in the opposite direction because, after three to four loaded trains, the coefficient of friction on the gauge face may rise above the acceptable limit. In this case, traffic should be considered as unidirectional and the lubricators spaced accordingly. Otherwise, these blocks of unidirectional traffic could cause rapid wear. Field tests were conducted on the Up track of the double track line where all the loaded trains travel through the Up track and the empty trains use the Down track where trains travel towards the mine sites. Limited numbers of empty trains or freight trains may travel on the Up track as per network needs and some grease accumulation on the rail crossings in the opposite direction from the test curves was therefore observed. In this research, the traffic factor as considered by Sroba et al. (2001) was adopted.

### 5.4.8 Bogie Factor, $B_G$

The bogie factor is important where mixed traffic predominantly shares the track, such as where freight and passenger traffic make up a significant percentage of total traffic volume. Different bogies have different steering capacity depending on curve radius and bogie compatibility to the specific curve. Wear of rails and wheels varies significantly due to the combined effect of bogies and curves. There was no scope in this research to study on the individual bogie performance and behaviour in the rail networks as there was limited resource and time frame to complete the field trials. It is considered that further study should be conducted to determine the effect of this issue. The bogie factor in this research has been considered the same as by Sroba et al. (2001), being 1 for tangent track, 1.5 for curves between 2 to 5 degrees, and 2 for curves greater than 5 degrees of Degree of Curvature.

### 5.4.9 Axle Load Factor, A

The effectiveness of rail lubrication potentially varies and needs to be rearranged based on the type of traffic such as heavy haul, mixed traffic, light passenger traffic as each of these traffic types have very different axle loads and the stress generated on the wheel/rail interface during curving is therefore totally different. Friction coefficient would increase when contact stresses increase and cause wheel/rail wear. Investigations revealed that heavy haul wheel wear was higher compared to rail wear. Due to mixed traffic, the axle load factor should be determined based on the fraction of each traffic type sharing the same track. The impact of all the different axle loads applied on the track should be reflected in the axle load factor. Detrimental impact on grease retentivity and distribution was found to be less for low axle loads. Quantification of the impact of different axle loads on wheel/rail wear and lubrication effectiveness is necessary.

The axle load factor can be determined by:

$$A = 1 + \frac{A_p \times n_p}{A_m} + \frac{A_f \times n_f}{A_m} \tag{5.5}$$

where

$A_p$ = Axle load of passenger traffic, in tonnes

$A_f$ = Axle load of freight traffic, in tonnes

$A_m$ = Standard axle load for the considered track, in tonnes (26.5 tonnes for the test site track)

$n_p$ = Percentage of passenger traffic

$n_f$ = Percentage of freight traffic

If there is no other traffic except standard heavy haul traffic, the axle load factor will be 1 as the percentage of passenger traffic $n_p$ and percentage of freight traffic $n_f$ is zero. For mixed traffic conditions the factor is considered based on the axle loads and percentages of the individual type of traffic which will be greater than 1.

### 5.4.10 Locomotive Factor, L

Locomotive Factor, L allows for the effect of the wheelbase of different locomotives on the rail while curving. No analysis has been conducted on this factor and it is considered to be as same as used by de Koker (1994) and Sroba et al. (2001). De Koker recommends to use the most common locomotive in the network as the baseline of locomotive and scaling all other locomotives in terms of wheelbase and axle load. As in the Thompson Subdivison in the Canadian Pacific Railway (CPR) the most common units were 4400 horsepower AC locomotives and this factor was considered as unity.

### 5.4.11 Speed Factor, V

Speed has a significant impact on lubrication and wear. Significant variation has been observed in grease wastage based on speed and quality of grease. It has also been noticed that, for the same quality of grease, loss is higher for higher speeds. Sliding speed of wheels at the gauge face varies with curve radius and severe sliding causes severe wear. The speed factor is considered to be the same as used by Sroba et al. (2001) and de Koker (1994). According to Sroba et al. (2001) it was left to unity in Thompson Subdivision. More work is required to identify the actual effect of speed on effective lubrication.

### 5.4.12 Bogie Misalignment Factor, M

It is necessary to allow for the effect of misalignment of bogies on the distribution of grease over tangent track and curves as it affects overall lubrication effectiveness. However, no analysis has been conducted on this factor and it is considered to be the same as used by

Sroba et al. (2001). De Koker (1994) recommends a value of up to 1.25 and Sroba et al. (2001) reported to use 1.23 in CPR Thompson Subdivision.

### 5.4.13 Braking Factor, $B_R$

Brake applications raise the wheel temperature and this could burn the grease or cause it to flow down to the bottom of the gauge face. Hence, lubricators should not be placed within a short distance of any severe braking zone.

For severe braking, $B_R > 1$

For no severe braking, $B_R = 1$, where no sharp braking like horizontal track.

According to Sroba et al. (2001), $B_R$ is a factor used to account for the effect of train braking. If a loaded freight train descends a long grade with a continuous moderate to severe brake application, the wheels can become hot enough to burn off the lubricant, or cause it to flow down to the bottom of the gauge face. Reducing this factor below unity implies that the curve will need more de Koker units to keep the curve lubricated. Lubricators must be placed closer together because of severe downgrades. Due to traction concerns, lubricators are not normally positioned at the top or bottom of grades. By way of example, one heavy haul railway uses a factor of 0.8 for a 2% grade. Sroba et al. (2001) reported that CPR used the factor as unity in the Thompson Subdivision.

## 5.5 Extension of Lubricator Placement Model

Each factor has been evaluated in the previous section. The combined effect of different track and traffic factors has been evaluated for each curve in the proposed model.

The proposed model of wayside lubrication defines the Lubrication Effectiveness Index (LEI) for each curve based on the proposed new factors and modified existing factors. The following equation has been previously mentioned as Equation 5.2.

$$Lubrication\ Effectiveness\ Index, LEI = \frac{(C+S)*G*R*P*L_{perf}*P_{profile}}{T*L*A*V*M*B_R*B_G}$$

This model is an extended and enhanced method of lubricator placement to that of de Koker (1994) and Sroba et al. (2001). We can compare it with those models as given below:

$$Final\ de\ Koker\ (1994)\ formula,\ \frac{\frac{(C+S)*G*R}{P}}{T*L*A*V*M*B}$$

$$Final\ Sroba\ et\ al.(2001)\ formula, \frac{(C+S)*G*R*P}{T*L*A*V*M*B_R*B_G} \qquad (5.6)$$

### 5.5.1 Simplified Lubrication Effectiveness Index, LEI

In this study, a simplified lubrication effectiveness index (LEI) has been presented for the lubricator placement where,

Both (left hand and right hand curves) have been taken into account for long applicator bars. Long applicator bars serve grease to both rails in the tangent track which could be transferred to high and/or low rails in the subsequent curves. The following salient features have been considered in this model to simplify the LEI:

Only Grease C has been considered as the selected grease.

2 long applicator bars on each rail have been considered as the selected bars.

The wheel/rail contact pattern in each curve is considered as unchanged.

Locomotive wheel base is considered as unchanged.

The same axle load is considered to be in operation, though freight and passenger traffic share the track.

Bogie misalignment is unchanged.

Same bogies are in use.

Considering the above assumptions, the LEI can be further simplified as:

$$Lubrication\ Effectiveness\ Index, LEI = \frac{(C+S)*R}{T*B_R} \qquad (5.7)$$

## 5.6 Numerical Illustration of Placement Location

A track chart of the field trial site on the Queensland Rail network in the vicinity of Aldoga, Gladstone, Queensland Australia is detailed in Table 5.13. Table 5.14 show the simplified calculations for the test site lubricator placement determination.

Table 5-13: Track chart

| Track | Curve No. | To (km) | From (km) | Radius (m) | Cant (mm) | Speed (km/h) | Direction | Length (m) |
|---|---|---|---|---|---|---|---|---|
| ALDOGA | | | | | | | | |
| Lubricator | | 554 | 553.9295 | Tangent | | | N/A | 70.457 |
| UP | 2 | 553.9295 | 553.6643 | 596 | 65 | 80 | R | 265.278 |
| | | | | Tangent | | | N/A | 176.149 |
| UP | 3 | 553.4881 | 553.1746 | 599.7 | 65 | 80 | L | 313.486 |
| | | | | Tangent | | | N/A | 157.23 |
| UP | 4 | 553.0174 | 552.72 | 398.5 | 55 | 60 | R | 297.4 |
| UP | 5 | 552.72 | 552.613 | 383.527 | 55 | 60 | R | 107 |
| UP | 6 | 552.613 | 552.5 | 470.096 | 55 | 60 | R | 113.02 |
| | | | | Tangent | | | N/A | 22.927 |
| UP | 7 | 552.4771 | 552.3499 | 411 | 70 | 60 | R | 127.159 |
| | | | | Tangent | | | N/A | 40.004 |
| UP | 8 | 552.3099 | 551.4601 | 419 | 70 | 60 | L | 849.75 |
| | | | | Tangent | | | N/A | 40.017 |
| UP | 9 | 551.4201 | 551.0325 | 411 | 70 | 60 | R | 387.643 |
| | | | | Tangent | | | N/A | 183.78 |
| UP | 10 | 550.8487 | 550.629 | 584.724 | 65 | 80 | R | 219.7 |
| UP | 11 | 550.629 | 550.406 | 606 | 65 | 80 | R | 223 |
| UP | 12 | 550.406 | 550.165 | 586.533 | 65 | 80 | R | 241 |
| Adjusted | | 550.165 | 549.054 | Tangent | | | N/A | 1111 |
| UP | 13 | 549.054 | 549.286 | 1620 | 35 | 100 | R | 232 |
| UP | 14 | 549.286 | 549.377 | 866.759 | 50 | 100 | R | 91 |

Table 5-14: LEI calculation and lubricator placement location using extended model

| | | | | | | | All | Cumulative | LEI |
|---|---|---|---|---|---|---|---|---|---|
| | | 1746.0 | | | | 2% Grade | Curves | | Limit |
| Track factors | | | Traffic factors | | | | Uni-directional | LEI | Uni-directional |
| C | S+C | R | T, Uni-directional | T, Bi-directional | BR | BR | Grade Track Lube Analysis | Grade Track Lube Analysis | 13500 |
| | | radius | traffic | traffic | brake | brake | | | |
| | | average | factor | factor | factor | factor | LEI | LEI | Lube Unit |
| | | Degrees | Uni-directional | Bi-directional | Level | % grade | Individual | Cumulative | Position |
| 0 | | 0.000 | 1.0 | 2.00 | 1.00 | 0.8 | 0 | 0 | 1 |
| 265 | 271 | 2.930 | 1.0 | 2.00 | 1.00 | 0.8 | 994 | 994 | |
| 0 | | 0.000 | 1.0 | 2.00 | 1.00 | 0.8 | 0 | 994 | |
| 313 | 322 | 2.911 | 1.0 | 2.00 | 1.00 | 0.8 | 1171 | 2165 | |
| 0 | | 0.000 | 1.0 | 2.00 | 1.00 | 0.8 | 0 | 2165 | |
| 297 | 301 | 4.381 | 1.0 | 2.00 | 1.00 | 0.8 | 0 | 2165 | |
| 107 | 107 | 4.552 | 1.0 | 2.00 | 1.00 | 0.8 | 609 | 2774 | |
| 113 | 114 | 3.714 | 1.0 | 2.00 | 1.00 | 0.8 | 527 | 3301 | |
| 0 | | 0.000 | 1.0 | 2.00 | 1.00 | 0.8 | 0 | 3301 | |
| 127 | 129 | 4.248 | 1.0 | 2.00 | 1.00 | 0.8 | 684 | 3985 | |
| 0 | | 0.000 | 1.0 | 2.00 | 1.00 | 0.8 | 0 | 3985 | |
| 850 | 852 | 4.167 | 1.0 | 2.00 | 1.00 | 0.8 | 4437 | 8422 | |
| 0 | | 0.000 | 1.0 | 2.00 | 1.00 | 0.8 | 0 | 8422 | |
| 388 | 393 | 4.248 | 1.0 | 2.00 | 1.00 | 0.8 | 2088 | 10510 | |
| 0 | | 0.000 | 1.0 | 2.00 | 1.00 | 0.8 | 0 | 10510 | |
| 220 | 224 | 2.986 | 1.0 | 2.00 | 1.00 | 0.8 | 838 | 11347 | |
| 223 | 223 | 2.881 | 1.0 | 2.00 | 1.00 | 0.8 | 803 | 12151 | |

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| 241 | 269 | 2.977 | 1.0 | 2.00 | 1.00 | 0.8 | 1000 | 13151 | |
| 0 | | 0.000 | 1.0 | 2.00 | 1.00 | 0.8 | 0 | 13151 | |
| 232 | 260 | 1.078 | 1.0 | 2.00 | 1.00 | 0.8 | 350 | 13501 | |
| 91 | | 2.014 | 1.0 | 2.00 | 1.00 | 0.8 | 0 | 13501 | |
| Next lube unit in tangent track | | | | | | | | | 1 |

Applied factors in this proposed lubricator placement model were developed based on field test data analysis and the results achieved. Factors have been defined based on the effects they cause on lubrication effectiveness. This approach could significantly improve the placement of the lubricators in heavy haul rail networks. A perfect approach would be when a network owner conducts field tests on various configurations of equipment, applicator bars, various greases, rail and wheel profiles, varying axle loads in mixed traffic and changes in predominant traffic direction. They should also determine their own values of each factor before using Equation 5.2.

## 5.7 Conclusions

Wayside lubrication is a widely used practice in rail curve lubrication and need to be implemented effectively to achieve best outcomes. Extensive field testing has covered the performance of several types of grease with several combinations of long and short applicator bars. The lubricators and applicator bars from different suppliers also performed significantly differently. Long applicator bars in tangent track and short applicator bars in the transition spirals of curves were observed and found that the applicator bars have significant effects on consumption of grease, transport of grease and carry distance. It was also found that different greases with the same applicator bars showed significantly different results in carry distance under the same track and traffic conditions. It has been noticed that wheel/rail contact patterns have significant impact on grease sustainability and transport. Grease in the curves with conformal contact showed very good grease coverage compared to the curves having single point or two-point contact. Quick burn out of grease due to extreme loads in single point or two-point contact was found to occur, whereas very good coverage of grease was observed in the curves with conformal contact. The field trials also showed that long bars in the tangent track displayed superior performance compared to short bars in the transition spirals of curves with the same grease application. This research presents some recent field study data and develops a framework for measuring lubrication effectiveness on Australian heavy haul rail lines. A practical and technical approach was developed to implement an extended placement model which highly emphasised the necessity of extensive studies on each of the salient factors. The extended placement model is based on factors influencing the LEI which is a simpler approach in comparison to other researchers.

The cumulative LEI up to the last curve of carry distance is considered as the base for the selection of next lubricator location for a particular configuration of equipment and grease. LEI need to be calculated for the extended network and the subsequent lubricator placement location is selected at the distance of the cumulative LEI as it was considered for the first installation location. Due to variation in different contribution from different factors the physical distance might be varied from the lubricator to lubricator. Besides theoretical calculations; suitable accessibility for lubricator operation and maintenance aspects must also be considered for actual location of lubricators.

The individual LEI have been calculated based on the individual factors in the LEI formula. In the Table 5.14 the individual LEIs would be similar or different for each of the tested grease (if same or different grease factor) as per the simplified formula which is considered for simplest and least complicated small section of track network. But the cumulative LEI should be different for each grease with different applicator bar combination as they have different carry distance. In a complicated track network where various types of greases and/or equipment may is used, there will be difference in individual LEI based on the each individual factor is considered.  Therefore cumulative LEI can differentiate between individual grease and can generate appropriate placement location plan for a vast track network.

LEI can be calculated for each individual grease and equipment configuration. Cumulative LEI for lubricator placement for each combination can be validated as it will be different based on the field test performance of that combination.

# Chapter 6

## DEVELOPMENT OF ECONOMIC MODEL FOR LUBRICATION DECISION

### 6.1 Introduction

Rail transport operates in a very highly competitive environment, where there is a continuous expectation of cost savings, asset life improvement and environmental friendliness through efficient operations and optimal energy consumption. Compared to other industries and even other transport sectors, railroads contribute significantly to lower greenhouse emissions. According to AAR (2012), rail freight emits 75% less greenhouse gas emissions per ton-mile of freight movement compared to road freight transport. Transporting more freight by rail can also save U$101billion (AUD132.31 billion) each year by eliminating highway congestion which causes 4.8 billion hours of time delays and 1.9 billion gallons (7.41 billion litres) of fuel consumption. Figure 6.1 and Table 6.1 show the greenhouse gas emissions from different industries.

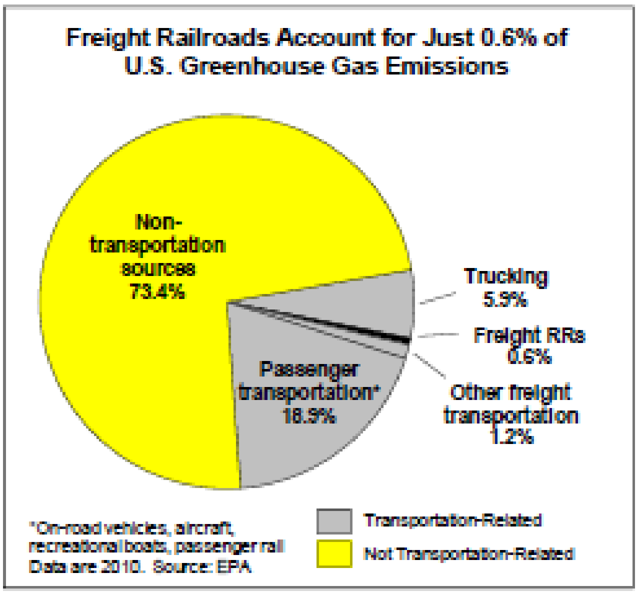



Figure 6.1: Greenhouse gas emissions distribution by various industries including freight rail roads in the USA (AAR 2012)

Table 6-1: US greenhouse gas emissions by economic sector and from transportation sector (AAR 2012)

**U.S. Greenhouse Gas Emissions By Economic Sector: 2010**

| Economic Sector | Tg CO2 Eq. | % of Total |
|---|---|---|
| Electric. generation | 2,306.5 | 33.8% |
| Residential | 365.2 | 5.4% |
| Industry | 1,394.2 | 20.4% |
| Agriculture | 494.8 | 7.3% |
| Transportation | 1,834.0 | 26.9% |
| Commercial | 381.7 | 5.6% |
| U.S. Territories | 45.5 | 0.7% |
| Total | 6,821.8 | 100.0% |

Data are in teragrams of CO2 equivalents.

**U.S. Greenhouse Gas Emissions from Transportation: 2010**

| Economic Sector | Tg CO2 Eq. | % of Transp. Total |
|---|---|---|
| Trucking | 402.2 | 22.1% |
| **Freight Railroads** | **40.0** | 2.2% |
| Waterborne Freight | 26.5 | 1.5% |
| Pipelines | 38.8 | 2.1% |
| Aircraft | 131.2 | 7.2% |
| Recrational Boats | 16.8 | 0.9% |
| Passenger Railroads | 6.2 | 0.3% |
| Cars, Light Trucks, Motorcycles | 1,138.1 | 62.7% |
| Buses | 16.5 | 0.9% |
| | 1,816.3 | 100.0% |

Source: EPA, *Inventory of U.S Greenhouse Gas Emissions and Sinks: 1990-2010*, Tables ES-7, A-113, and A-114. Totals for "transportation" in the two tables do not match because the table on the left includes emissions from sources considered to be transportation but not considered to be passenger or freight (e.g., lubricants).

Dramatic improvements of this performance was contributed to by new freight car designs, new locomotives, highly advanced computer software and systems in rail operations, new technologies such as rail lubrication to reduce friction at the wheel/rail interface, power distribution, low torque bearings and automated defect detections.

Wheel/rail lubrication has a crucial capacity to improve both rail and wheel service life. It is mainly used to reduce wear, operating costs, energy consumption and noise. Widely used lubrication methods are wayside, hi-rail and on-board lubrication. Wayside lubrication is commonly used in Australian heavy haul, freight and passenger railway networks. Financial investment in wayside lubrication could reduce wheel/rail wear and significantly reduce rail and wheel maintenance and replacement costs as well as energy consumption. The Association of American Railroads estimated that wear and friction occurring at the wheel/rail interface due to ineffective lubrication costs in excess of US$ 2 billion (AUD2.62 billion) each year (Sid et al. 2002). Profillidis (2000) reported that rail transport consumes one third of the energy for the road transport of the same load. According to de Koker (2010), the effectiveness of lubrication can be improved by well managed lubrication equipment maintenance and could save AUD8.2 million/annum in a railway system similar to Transnet, South Africa. The economic models proposed here for the Australian heavy haul railways are illustrated using numerical examples and would be useful to those industries for asset life enhancement, reliability and safety improvement along with reduction of risks and costs. This work has been also been reported in detail by the author in the CRC for Rail Innovation Project Report (CRC Australia 2014).

### 6.2 Life Cycle Cost

Life cycle cost analysis is a must do action before any major asset acquisition. According to New South Wales Treasury (2004), the determination of costs is an integral part of the asset management process and is a common element of many of the asset manager's tools, particularly for economic appraisal, financial appraisal, value management, risk management and demand management.

Life cycle cost is an estimation of the total ownership cost of an asset throughout of its life. The practical asset life may be considered from the concept generation phase to disposal or from purchasing to the disposal of that asset. The Total Asset Management- Life Cycle Costing Guideline (NSW Treasury 2004) reports that the Life Cycle Cost (LCC) of an asset is defined as "the total cost throughout its life including planning, design, acquisition and support costs and other costs directly attributable to owning or using the asset". It adds together all the costs of alternatives over their lifetime and enables an evaluation on a common basis for the period of interest. Hastings (2000 & 2009) reported that the aim of life cycle costing is to minimise costs over the life cycle of the equipment. The Life Cycle Costing- Better Practice Guide of the Australian National Audit Office (2001) indicates that the process of life cycle costing fundamentally involves assessing costs arising from an asset over its life cycle and evaluating alternatives that have an impact on this cost of ownership. It estimates the life cycle cost of an asset using the following formula:

LCC = capital cost + life-time operating costs + life-time maintenance costs + disposal cost – residual value

The values of each item in the above formula may be difficult to determine before application of the asset, and there will be uncertainty regarding future costs such as the prediction of usage over time, nature and level of operating cost, necessary maintenance cost, impact of inflation, predicted useful life of the asset and significance of future expenditure compared to current day expenditure.

Following the asset acquisition and a reasonable period of usage, the estimated life cycle cost should be compared with the real life cycle cost to understand the variation between estimation and actual cost. A proper asset reporting process should be maintained throughout the life span in service which may include initial purchasing and set up costs, accumulated depreciation costs, accumulated maintenance and servicing costs, life cycle cost modelling, and detailed asset activity. This report may be used to monitor actual costs to make comparisons with predicted costs and determine the appropriate economic disposal point.

### 6.2.1 LCC Analysis of Wayside Lubrication Practice

Cost effective wayside lubrication must be established through appropriate decision making during selection of equipment and the preferred method of application should be based on field and laboratory test findings on different equipment set-ups and grease types. Though there is a variation of the overall LCC, all the various types of wayside lubrication technologies have similar types of costs over their life spans. Due to the technological difference, cost components have huge variation between older hydraulic/mechanical equipment and modern high performance electric equipment. The comparative study of major LCCs of wayside lubrication and their evaluation is discussed below.

### 6.2.2 Capital Cost

Capital cost is the initial investment cost of equipment purchasing and set-up on site. The old hydraulic or mechanical lubricator may be considered as less expensive compared to new electric lubricators. However, in reality, when the decision is made for acquisition of a bulk number of the electric lubricator units, this will be more cost effective compared to hydraulic or mechanical units. The number of hydraulic or mechanical units required for a network is much greater than the number of electric units needed.

Though the single electric units cost more than three times the capital expenditure compared to mechanical and hydraulic units, operating cost of the electric type is extremely low. Overall LCC shows that the electric lubricator is highly attractive compared to older technology even though the individual initial capital cost is higher. Initial capital cost includes:

- Design & Development Cost, $C_{dd}$, if it is paid by the railway operator/owner.
- Purchasing Cost ($C_p$).
- Installation Cost ($C_i$).
- Accessories Cost ($C_a$).

Remote condition monitoring and the selection of a range of electric sensors may incur some additional capital costs in the latest electric lubricators, but such features can generate potential savings in future operating and maintenance costs. Solar power and remote condition monitoring enhance the capability of electric units and establish them as unparalleled in comparison to older technology.

### 6.2.3 Operating Cost

Operating cost is the most significant cost in the life cycle of a wayside lubricator and plays an enormous role in asset selection. It is the total cost which is incurred during the operating life. Operating costs are incurred by any equipment unless it does not need any power/fuel or personnel to operate, no space is required for installation, no people are needed for maintenance, no material is consumed or equipment movement needed to continue operation, and it does not wear or fail during its life time.

In heavy haul railway lubrication, the operating cost is the biggest cost in a lubricator’s life span. Different equipment technologies contribute at a different extent to operating costs and benefits. Therefore, equipment selection has to be perfect to achieve the most cost effective operation. The downstream impact of bulk numbers of equipment purchasing should be determined through field trials before finalising the decision to acquire.

Major operating costs in the performance of wayside lubrication include:

- Lubricant Cost ($C_l$).
- Servicing Labour Cost ($C_{lab}$).
- Servicing Vehicle Operating Cost ($C_v$).
- Track downtime cost (if applicable).

Each of the above costs can be broken down into multiple levels of costs which sum to the above cost items.

### 6.2.4 Maintenance Cost

Maintenance cost is the second largest cost component in a lubricator’s life cycle. Due to the operating conditions, environmental and climatic conditions and mechanical forces from moving traffic, there are diverse maintenance activities involved with lubricator units. The main maintenance actions are required because of damage to or blockage of applicator bars and grease delivery hoses, grease tank failure, plunger or actuator failure, grease pump failure, or electrical breakdowns such as control panel, digital display, power supply, software, and various sensors. A portion of minor maintenance is done in-situ which can be managed by servicing people and affords quick return to online status. The major events of maintenance such as major failure/breakdown and routine change-out/overhaul are conducted in a permanent maintenance facility. Maintenance actions are required more frequently on mechanical and hydraulic lubricators compared to electric units. Main maintenance cost items may include, but are not limited to:

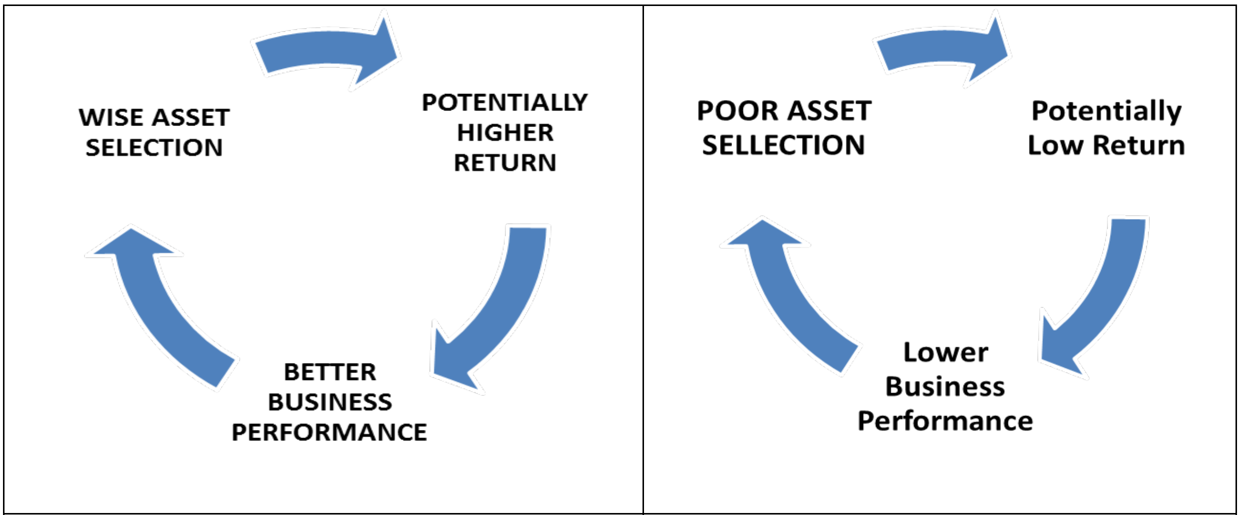
- Cost of spare parts ($C_{sp}$).
- Maintenance Labour Cost.

### 6.2.5 Replacement or Disposal Cost

At the end of the service life, replacement of lubricators may include some costs which may cover unit removal cost, transport to the depot, cleaning for proper disposal and payment of fees to the vendors or environmental management people. Lubricator disposal should follow appropriate standards and procedures to eliminate environmental impacts such as ground contamination, aquatic contamination, and vegetation contamination. Due to the long service life, the disposal cost is comparatively low compared to other costs in the life cycle.

## 6.3 LCC Modelling

Effective life cycle cost analysis on preferred alternatives is required to generate an effective modelling of the real life future cost of investment. According to the International Road Federation (IRF 2014), the value of life cycle cost analysis is that it generates a tool to evaluate resource trade-offs and prioritisation for long timescales compared to the way the capital budgeting decisions are generally done. It measures the streams of cost and benefits over the unit's specified lifetime. The lowest cost option may not generate the best outcome in terms of business performance. Appropriate decisions in the early stages may generate enormous savings and potential benefits in the operating stage. The impact of poor asset selection even with low capital cost may continue through poor business performance as a vicious circle as shown in Figure 6.2, and this cycle may continue for the whole asset life. By way of contrast, real life experience shows appropriate asset selection with comparatively high capital cost may result in very low operating and maintenance costs which generate multiple times the benefit compared to poor equipment. Such a successful outcome maintains high business performance throughout the asset service life.




(a) (b)

Figure 6.2: Continuous consequence cycle of wise and poor asset selection: (a) Wise asset selection and continuity of better performance (Uddin et al. 2014b), (b) Poor asset selection and vicious circle of lower performance (Uddin et al. 2014b)

When alternative equipment suppliers are competing to gain their business, a comprehensive and thoroughly understood analysis needs to be put in place before the acquisition decision is made. In heavy haul railway lubrication practice, it is clear that less expensive but ineffective equipment and grease both contribute to high operating costs and very poor protection of wheel/rail assets. A large proportion of high maintenance cost for rail and wheel assets comes from ineffective lubrication. Therefore, lifelong costs and benefits should be effectively identified before the acquisition of planned assets as this may produce sustainable business in the competitive environment.

### 6.4 Cost Modelling for Wayside Lubrication

The following items are required for developing a cost model (Uddin & Chattopadhyay 2009; Uddin et al. 2011a) for wayside lubrication:

- Purchase and set-up cost of lubricator ($C_{ps}$).
- Electricity consumption cost (if applicable) ($C_e$).
- Servicing and maintenance cost ($C_{sm}$).
- Repair cost ($C_r$).
- Vehicle cost ($C_v$).

- Travelling cost/fuel cost ($C_t$).
- Labour cost ($C_{lab}$).
- Emergency maintenance cost ($C_{em}$).
- Lubricant cost ($C_l$).
- Cost of spare parts ($C_{sp}$).
- Grinding cycle cost ($C_{grind}$).
- Replacement cost ($C_{rep}$).
- Lubricant cost for wastage of lubricant ($C_{lw}$):
    - Track downtime cost ($C_{downtime}$) (if applicable).
    - Wheel/rail life loss due to breakdown of unit ($C_{breakdown}$).
    - Risk cost due to derailments/incidents caused by poor/excessive lubrication ($C_{risk}$).

## 6.5 Cost Calculation for Wayside Lubricators

Servicing and maintenance costs per service interval can be given by:

$$C_{sm,i} = C_{v,i} + C_{t,i} + C_{lab,i} + C_{em,i} + C_{sp,i} + C_{lw,i} + C_{grindi} \tag{6.1}$$

where, $C_{v,i}$ = Vehicle cost per service interval

$C_{t,i}$ = Travelling cost per service interval

$C_{lab,i}$ = Labour cost per service interval

$C_{em,i}$ = Emergency maintenance cost per service interval

$C_{sp,i}$ = Cost of spare parts per service interval

$C_{lw,i}$ = Lubricant cost for wastage per service interval

$C_{grindi}$ = Grinding cycle cost

$$\text{Servicing and maintenance}\frac{\text{cost}}{\text{year}}, C_{sm,yr} = \sum_{i=1}^{n} C_{sm,i} \tag{6.2}$$

If each servicing and maintenance cost/ service interval is considered to be the same, then:

$$\text{Servicing and maintenance}\frac{\text{cost}}{\text{year}} = C_{sm,i} * n \tag{6.3}$$

$$\text{Emergency maintenance cost /service interval, } C_{em} = \sum C_{em,i} \tag{6.4}$$

where, $C_{em,i} =$ Cost of each emergency maintenance $= C_{v,i} + C_{t,i} + C_{r,i}$

$C_{v,i}$ = Vehicle cost per emergency maintenance

$C_{t,i}$ = Travelling cost per emergency maintenance

$C_{r,i}$ = Repair cost per emergency maintenance

Emergency maintenance $\frac{\text{cost}}{\text{year}}, C_{em,yr} = \sum_{N=1}^{n}(\sum C_{em,i})$ (6.5)

where $N$ = No. of service intervals per year

Grinding Cycle Cost/year, $C_{grind,yr} = \sum_{i=1}^{n} C_{grind}$ (6.6)

Wheel/rail life loss due to breakdown of units per year is given by:

$$C_{breakdown,yr} = \sum_{i=1}^{n} C_{breakdown} \tag{6.7}$$

Excessive lubrication and poor lubrication may both contribute to vehicle rollover and derailments. A major derailment may cause line closure, rolling stock damage, and below rail infrastructure damage with total costs of millions of dollars. Himark Consulting Group Pty Ltd (2005) reported that the Black Mountain derailment in Queensland, Australia in July 2001 closed the line for nearly 2 weeks, destroyed 73 wagons and 2 locomotives valued at $20 million and caused extensive damage to below rail infrastructure. Further work needs to be done to determine the risk of derailment cost in lubrication practice.

The annual cost of a lubricator at time t can be given by:

$$C_{annum,t} = C_{sm,yr,t} + C_{l,yr,t} + C_{e,yr,t} + I_{annum,t} + C_{breakdown,yr} \tag{6.8}$$

where $C_{annum,t}$ = Annual cost of lubricator at time, t

$C_{sm,yr,t}$ = Annual servicing and maintenance cost at time, t

$C_{l,yr,t}$ = Annual lubricant cost at time, t

$C_{e,yr,t}$ = Annual electricity cost at time, t (if applicable)

$C_{breakdown,yr}$ = Annual wheel/rail life cost due to breakdown of lubricator unit

$I_{annum,t}$ = Annual capital servicing charge for purchasing and set-up cost of the lubricator

$t$ = index number of year of evaluation

$r$ = discount rate per year

$N$ = the number of years within the planning horizon

$y$ = expected life of lubricator in years

$$I_{annum,t} = PMT[r, y, C_{ps}] \quad (6.9)$$

If the annual cost $C_{annum,t}$ is constant, the present value of the cost of a lubricator over the economic life evaluation can be described by:

$$C = PVA(r, N, C_{annum,t}) \quad (6.10)$$

For solar powered lubricators, the purchase price will include the cost of solar panels. The annual running costs are then less than the cost of a standard electric lubricator by the amount of the cost of purchased electricity.

### 6.8 Cost-Benefit Analysis of Applicators and Various Lubricants

In any given time period t, the net value of a lubrication system in a particular curve is given by:

$$NV_T = C_{benefit,t} - C_{cost,t} \quad (6.11)$$

where

$C_{benefit,t}$ = Savings due to lubrication effectiveness in the period, t

($C_{cost,t}$) = Total cost in the period, t

This model can be replicated for different lubricator placements, lubricant types, lubricator types, application rates, lubricator service intervals and maintenance strategies.

The proposed model could be considered for heavy haul, freight and passenger traffic. Evaluation of the lubrication strategy needs to be systematic. The following proposed model shows a systematic approach of modelling of lubrication strategy from the dedicated track selection to decision evaluation.

## 6.6 Cost Data Analysis for Lubrication Methods

The economics of best practice in friction management are analysed for the Australian heavy haul industry. Based on field study data, a cost-benefit analysis has been conducted to define the optimal combination of equipment and greases.

### 6.6.1 Economic Data Analysis

For long term operating decisions, it is crucial to analyse economic data for different technology options. Decisions should be taken based on various costs and potential benefits available from the particular technology. The economic evaluation should be based on simple calculations for the considered scenario of line MGT (Million Gross Tonnes).

#### *6.6.1.1 Grease Cost Data Analysis: Electric Lubricators*

Electric lubricators are compatible with both long bar and short bar applications. There is a significant cost difference for both applications of electric lubricators as a significantly higher number of lubricators are needed for the short bar application compared to the long bar application. Also, the short bar lubrication method protects a much smaller length of rail compared to long bars. Therefore, there are varying costs and benefits of these applications.

##### ***6.6.1.1.1 Comparison: Grease Consumption Cost per Year for Electric Lubricator with Long and Short Applicator Bars***

Equipment: Electric Lubricators

Type of Applicator Bars:

2 Long Bars on Each Rail at tangent track location and

2 Short Bars on High Rail at the transition spiral of the curve

Train Frequency: 480 trains per month having 504 axles per train

MGT per year: 76.93

The grease Consumption Cost per Year is given in Table 6.2

Table 6-2: Greaser consumption rate and cost for long bars and short bars

| Type of Bar | Grease Type | Trains per Month | Axles per Train | Grease per 12 Axles (gm) | Grease Used per Month (gm) | Monthly Grease Cost ($AUD) | Annual Grease Cost per Lubricator ($AUD) | Grease Used per 1000 Axles (gm) |
|---|---|---|---|---|---|---|---|---|
| Long Bars | Grease C | 480 | 504 | 6.08 | 122572.8 | 858.0096 | 10296.12 | 506.6667 |
| | | | | Grease per 18 Axles (gm) | Grease Used per Month (gm) | Monthly Grease Cost ($AUD) | Annual Grease Cost per Lubricator ($AUD) | Grease Used per 1000 Axles (gm) |
| Short Bars | Grease C | 480 | 504 | 5.08 | 68275.2 | 477.9264 | 5735.117 | 282.2222 |

The following conclusions can be drawn from the data given in Table 6.2:

- Grease cost of one lubricator with long bars per year equals $10296.12.
- Grease cost of one lubricator with short bars per year equals $5735.12.
- Short bar units cover only one direction of curves (left hand or right hand).

- To replace one long bar unit needs a minimum of two short bar units.
- The number of short bar units required to reproduce the coverage of one long bar unit will cost more.

#### *6.6.1.1.2 Comparison: Grease Consumption Costs for Axle Passes*

The following data has been collected for the electric lubricator based on simple economic calculations which show the costs involved.

The salient features of the field tests with axle passes and grease consumptions are listed below:

2 Long Bars on Each Rail at tangent track location and

2 Short Bars on High Rail at the transition spiral of the curve

Train Frequency: 480 trains per month having 504 axles per train

MGT per year: 76.93

The grease Consumption Cost per metre per 1000 axles is given in Table 6.3

The following conclusions can be drawn from the data given in Table 6.3:

- Grease cost of one lubricator with long bars per year equals $10296.12.
- Grease cost of two lubricators with short bars per year equals $11470.23.
- Total rail length lubricated by long bar units equals 9200m and grease cost per metre per 1000 axles equals $0.000386.
- Total rail length lubricated by two short bar units equals 4856m and grease cost per metre per 1000 axles equals $0.000407.
- Realistically you need 4 short bar units to lubricate the same rail length as is lubricated by one long bar unit.
- First year equipment and grease cost for one long bar unit equals $30296.12.
- First year equipment and grease cost for two short bar units equals $51470.23.
- Therefore, long bar units are highly economic compared to short bar units when considering the level of protection provided and the rail length served by the units.

Table 6-3: Grease consumption cost

| Bar Type | Grease Type | Carry Distance (m) | Total Rail Length Lubricated (m) | | No of Lubricators | Total Annual Grease Cost ($AUD) | Total Lubricator Cost ($AUD) | Grease & Lubricator Cost ($AUD) | Grease Cost/metre/1000 Axles ($AUD) |
|---|---|---|---|---|---|---|---|---|---|
| Long Bars | Grease C | 4600 | 9200 | | 1 | 10296.12 | 20000 | 30296.12 | 0.000386 |
| | | Carry Distance (m), right hand curve | Total Rail Length Lubricated (m) | Carry Distance (m), left hand curve | No of Lubricators | Total Annual Grease Cost ($AUD) | Total Lubricator Cost ($AUD) | Grease & Lubricator Cost ($AUD) | Grease Cost/metre/1000 Axles ($AUD) |
| Short Bars | Grease C | 2870 | 4856 | 1986 | 2 | 11470.23 | 40000 | 51470.23 | 0.000407 |

#### *6.6.1.1.3 Grease Consumption Cost based on the Service Life of Lubricators*

In order to calculate the grease consumption for both types of Supplier X and Supplier Y equipment, the following data is required:

2 Long Bars on Each Rail at tangent track location and

2 Short Bars on High Rail at the transition spiral of the curve

Train Frequency: 480 trains per month having 504 axles per train

MGT per year: 76.93

The grease Consumption Cost per MGT and for the total lubricator service life (estimated to be 30 years) are given in Table 6.4 which shows that:

- Grease consumption in one lubricator with long bars per MGT equals 19.12kg and it cost $133.83 per MGT.
- Grease consumption in two lubricators with short bars per MGT equals 21.299kg and it cost $149.09 per MGT.
- Over the expected service life of 30 years, the total grease cost of one lubricator with long bars would be $308883.50 for 2307.9 MGT.
- Grease cost per MGT for lubricating 9200m of rail using short bar units equals $282.47.
- For lubricating the same rail length, the short bar system costs 2.11 times to the cost of the long bar system (please note that short bar system means 1 unit for left hand curves and 1 unit for right hand curves).

The above analysis demonstrates that the long bar units are highly economical in comparison to the short units.

Table 6-4: Grease consumption cost based on service life of electric lubricators

| Bar Type | Grease Type | Trains per Month | No of Axles per Train | Operating Axle Load (tonnes) | Load Carried per 1000 Axles (tonnes) | No of Axles per MGT | Grease Consumption per MGT (kg) | Grease Cost per MGT ($AUD) | Considered Service Life (years) | Cost for Life ($AUD) | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Long Bars | Grease C | 480 | 504 | 26.5 | 26500 | 37735.85 | 19.1195 | 133.8365 | 30 | 308883.5 | | |
| | | | | Operating Axle Load (tonnes) | Load Carried per 1000 Axles (tonnes) | No Of Axles per MGT | Grease Consumption per MGT (kg) | Grease Cost per MGT ($AUD) | Grease Cost per MGT ($AUD) for Lubricating a Length of 9200m | Short Bar Unit/Long Bar Unit per MGT Cost Ratio | Considered Service Life (years) | Cost for 1 Unit for Service Life |
| Short Bars | Grease C | 480 | 504 | 26.5 | 26500 | 37735.85 | 21.29979 | 149.0985 | 282.4766 | 2.11061 | 30 | 172053.5 |

#### *6.6.1.2 Grease Cost Data Analysis: Hydraulic Lubricators*

As reported earlier, this study is mainly focused on electric lubricators; however, a brief cost analysis for hydraulic lubricators is also carried out. Hydraulic or mechanical lubricators are very low capacity lubricator units. Therefore, there are practical limitations to the use of these units for high MGT lines. As these units do not have precise control over their dispensing rate, the grease consumption is further discussed in the following section.

##### *6.6.1.2.1 Grease Consumption Cost per Year for Hydraulic Lubricator with Short Applicator Bars*

In case of a hydraulic lubricator, the cost of grease consumption per year is based on the salient features as listed below:

2 Short Bars on High Rail at the transition spiral of the curve

Train Frequency: 480 trains per month having 504 axles per train

MGT per year: 76.93

Grease consumption cost per year for hydraulic lubricators are given in Table 6.5.

Table 6-5: Grease consumption cost per year for hydraulic lubricators

| Bar Type | Grease Type | Trains per Month | Axles per Train | Grease per Axle (gm) | Grease Used per Month (gm) | Monthly Grease Cost ($AUD) | Annual Grease Cost ($AUD) |
|---|---|---|---|---|---|---|---|
| Short Bars | Grease A | 480 | 504 | 1 | 241920 | 1693.44 | 20321.28 |

The outcome of this data collection can be summarised from Table 6.5 as follows:

- Grease cost of one hydraulic lubricator with short applicator bars per year equals $20321.20.
- Short bar units cover only one direction of curves (either left hand or right hand).
- To replace one electric long bar unit requires at least two hydraulic short bar units.
- Therefore, the required number of hydraulic short bar units may cost multiple times of $20321.28.

- It is economically unacceptable to run hydraulic units in preference to electric long bar units.
- Hydraulic lubricators are not suitable at all for high MGT lines due to exorbitantly high costs without any noticeable benefit.

#### *6.6.1.2.2 Hydraulic Lubricator Grease Consumption based on Axle Passes*

The features of the field testing for hydraulic lubricators are as below:

2 Short Bars on High Rail at the transition spiral of the curve

Train Frequency: 480 trains per month having 504 axles per train

MGT per year: 76.93

Grease Consumption Cost per metre per 1000 axles has been given in Table 6.6.

Table 6-6: Hydraulic lubricator grease consumption cost based on axle passes

| Bar Type | Grease Type | Carry Distance (m) | Total Rail Length Lubricated (m) | No of Lubricators | Total Lubricator Cost ($AUD) | Annual Grease & Lubricator Cost ($AUD) | Grease Cost per metre per 1000 Axles ($AUD) |
|---|---|---|---|---|---|---|---|
| Short Bars | Grease A | 100 | 100 | 2 | 7000 | 27321.28 | 0.07 |

Table 6.6 shows that:

- Total rail length lubricated by hydraulic short bar unit was 100m and grease cost per metre per 1000 axles equals $0.07.
- Realistically, too many hydraulic short bar units may be required to lubricate the same rail length served by one electric long bar unit.
- Grease cost for one hydraulic short bar unit for one year equals $27321.28
- Therefore, long bar units are extremely economic compared to hydraulic short bar units when considering the level of protection provided and the rail length served by the unit. Hydraulic units should not be recommended for a high MGT line with very high traffic density.

#### *6.6.1.2.3 Grease Consumption Cost per MGT based on Service Life of Hydraulic Lubricators*

2 Short Bars on High Rail at the transition of the curve

Train Frequency: 480 trains per month having 504 axles per train

MGT per year: 76.93

Grease Consumption Cost per MGT and for service life has been given in able 6.7.

The data in Table 6.7 reveals the following information:

- Grease consumption in one hydraulic lubricator with short bars per MGT was 75.47kg at a cost of $528.30.
- Cost comparison shows the hydraulic short bar unit is 363 times more costly than an electric long bar unit.
- Hydraulic units should therefore not be recommended over electric units either on high MGT or low MGT lines as they are not cost effective from the grease consumption point of view.

Table 6-7: Grease consumption based on service life of hydraulic unit

| Bar Type | Grease | Trains per Month | Axles per Train | Operating Axle Load (tonnes) | Load Carried per 1000 Axles (tonnes) | No of Axles per MGT | Grease Consumption per MGT (kg) | Grease Cost per MGT ($AUD) | Grease Cost per MGT ($AUD) for Length Lubricated of 9200m | Hydraulic Unit/Electric Long Bar Unit | Hydraulic Unit/Electric Short Bar Unit |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Short Bars | Grease A | 480 | 504 | 26.5 | 26500 | 37735.85 | 75.4717 | 528.3019 | 48603.77 | 363.1579 | |

### 6.6.2 Grease Loading Analysis

Grease loading (frequency of filling the grease tank) depends on the load and the axle passes, i.e., the traffic. This interval of loading must be considered during the decision making process of acquiring lubricator units. The tank size should be selected to match the traffic density of the specific lines being lubricated. Hydraulic units in high MGT lines may cause excessive workload for the maintenance personnel due to highly frequent loading cycles, whereas high capacity units can serve in high MGT lines with a reasonably maintainable frequency of loading. The frequencies of grease loading for different lubricator units are given in Table 6.8.

Table 6-8: Grease loading frequency for various lubricator configurations

| Grease Loading Analysis | | | | | | |
|---|---|---|---|---|---|---|
| Lubricator Type | Tank Capacity (kg) | Useful Tank Capacity 75% (kg) | Grease Consumption per MGT | Grease Loading Interval (MGT) | Grease Loading Interval (day) | Annual Traffic Volume (MGT) |
| Electric Long Bar | 380 | 285 | 19.11 | 14.91366 | 70.69461 | 77 |
| Electric Short Bar | 380 | 285 | 10.64 | 26.78571 | 126.9712 | 77 |
| Hydraulic Short Bar | 37.5 | 28.125 | 37.73 | 0.745428 | 3.533523 | 77 |

The data acquired in Table 6.8 is graphically presented in Figure 6.3. The analysis concludes that:

- Grease loading interval is too small for hydraulic units which are totally uneconomic to maintain due to limitation of operating resources.
- Electric long bar unit has a long interval in between grease loading which is very economic and manageable compare to other two.
- The hydraulic units are not applicable and maintainable in high MGT lines.

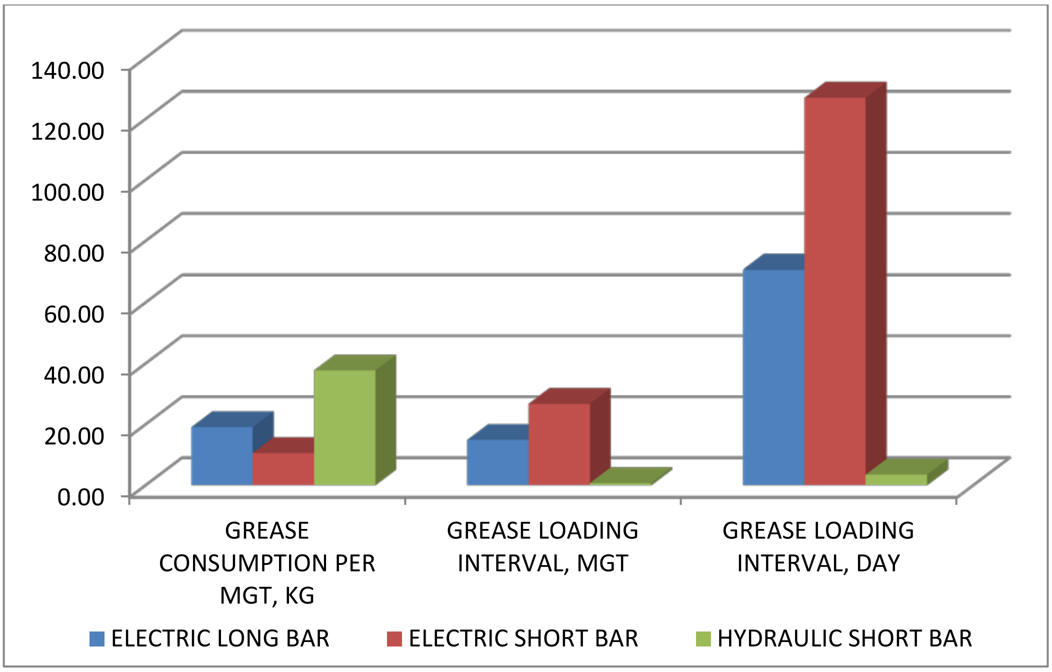



Figure 6.3: Grease consumption and grease loading intervals comparison

### 6.6.3 Operating & Maintenance Cost

The above analysis demonstrated that electric lubricators with long bar applicators provide significantly reduced operating and maintenance costs. This is mainly due to their placement locations and enhanced capability; the field trials have proven the superiority of this lubricator configuration over both electric short bar and hydraulic short bar units. Comparison of the major cost items incurred for operating and maintenance need attention. Electric long bar units with effective grease can significantly reduce the number of lubricators needed. Operating and maintenance cost comparisons for various lubricator configurations are given in Table 6.9.

Table 6-9: Operating and maintenance cost comparison for electric long bar unit, electric short bar unit and hydraulic unit

| Type of Unit | Lube Type | Cost ($AUD) | Cost Item Description |
|---|---|---|---|
| Hardware | | | |
| Hydraulic Unit | Poor Lube | 0 | Already Installed |
| Electric Long Bar Unit | Effective GF Lube | 20000 | Brand New Unit |
| Electric Short Bar Unit | Effective GF Lube | 20000 | Brand New Unit |

| | | | |
|---|---|---|---|
| Upgrade Truck | | | |
| Hydraulic Unit | Poor Lube | 0 | Gang Truck, No Change |
| Electric Long Bar Unit | Effective GF Lube | 15000 | Upgrade Lube Truck, Bulk Fill Pumps and Maintenance Facilities - Cost of Equipment |
| Electric Short Bar Unit | Effective GF Lube | 15000 | Upgrade Lube Truck, Bulk Fill Pumps and Maintenance Facilities - Cost of Equipment |
| Install New GF Units | | | |
| Hydraulic Unit | Poor Lube | Nil | Already Installed |
| Electric Long Bar Unit | Effective GF Lube | Incurred Cost | Lubricator Labour Installation Cost Per Unit |
| Electric Short Bar Unit | Effective GF Lube | Incurred Cost | Lubricator Labour Installation Cost Per Unit |
| Parts for GF Units | | | |
| Hydraulic Unit | Poor Lube | Very High | Parts for Units |
| Electric Long Bar Unit | Effective GF Lube | Significantly Low | Parts for Units |
| Electric Short Bar Unit | Effective GF Lube | Significantly Low | Parts for Units |
| Maintain Upgraded Truck | | | |
| Hydraulic Unit | Poor Lube | Incurred Cost | Maintain Truck Parts |
| Electric Long Bar Unit | Effective GF Lube | Incurred Cost | Maintain Truck Parts Upgraded for Bulk Filling @% of Cost of GF Pumping Equipment / Year |
| Electric Short Bar Unit | Effective GF Lube | Incurred Cost | Maintain Truck Parts Upgraded for Bulk Filling @% of Cost of GF Pumping Equipment / Year |
| Truck Usage & Travel | | | |
| Hydraulic Unit | Poor Lube | Very High Usage | Truck 100% of Year |
| Electric Long Bar Unit | Effective GF Lube | Low Usage | Truck 60% of Year |
| Electric Short Bar Unit | Effective GF Lube | Low Usage | Truck 30% of Year |
| Labour per Year to Maintain Units | | | |
| Hydraulic Unit | Poor Lube | Extremely High Usage of Labour | Labour Cost |

| | | | |
|---|---|---|---|
| Electric Long Bar Unit | Effective GF Lube | Very Low Usage of Labour | Labour Cost |
| Electric Short Bar Unit | Effective GF Lube | Low Usage of Labour | Labour Cost |
| Removal & Reinstallation at Grinding Cycles | | | |
| Hydraulic Unit | Poor Lube | Incurred Cost at Each Grinding Cycle | Removal and Reinstallation Of Unit |
| Electric Long Bar Unit | Effective GF Lube | Nil | Removal and Reinstallation Of Unit |
| Electric Short Bar Unit | Effective GF Lube | Incurred Cost at Each Grinding Cycle | Removal and Reinstallation Of Unit |

Based on operating and maintenance costs, the following conclusions can be derived from Table 6.9:

- Electric long bar units with an effective grease significantly reduce the number of lubricators needed over the track network which can be revealed from a detailed design plan of the network for effective lubrication.
- Due to low numbers of electric lubricators and technological advances, grease loading and maintenance need low efforts and resources.
- Electric lubricators can generate significant resource flexibility and eliminate resource constraints.
- Due to their location in the tangent track, applicator bars cause no interference with rail grinding operations, therefore not incurring any cost for removal and reinstallation. The common problem of missing reinstallation of lubricators after grinding is totally eliminated.
- Except for the applicator bars, there are no parts in the unit which can come into contact with a train, hence there are no issues with train movements damaging equipment.

### 6.6.4 Cost Savings with Remote Condition Monitoring and Preventive Maintenance

Lubricators demand regular servicing and preventive maintenance which can be expensive. In this case study, remote condition monitoring was used. It enhanced the capability, operability and maintainability of the electric lubricators with either long bar units or short bar units. It can dramatically reduce the manual inspection frequency of the units. Through remote performance monitoring, the units can generate:

- Any incident report on the unit.
- Any vandalism report.
- Any power interruption report.
- Grease tank level monitoring report.
- Grease pump motor amps report.
- Equipment status report such as online or offline.
- Surrounding weather/climate report such as ambient temperature, humidity.
- Rainfall report through rain sensor.
- Any unauthorised door opening report.
- Total number of train and wheel counts through the site.
- Alarm to fill up grease.
- Alarm on any malfunction of pump or motor or any other parts.

These automations eliminate guesswork or frequent visits to the units. The lubrication program can achieve world class performance with significantly low operating and maintenance costs. Remote condition monitoring provides the option to the rail operator of an effective process to monitor the reliability and usage of units, consumption of grease, time to fill, and MGT of traffic through the site, etc. As the equipment is powered by solar power with back up battery, there is a negligible risk of power failure except electrical cable collapse or damage.

## 6.7 Conclusions

The economic data analysis provides an in depth understanding and decision making power to decide appropriate cost effective practices and adds value to the best practice rail curve lubrication which effective and efficient. The following conclusions and recommendations can be made:

- Electric lubricator with long bars from Supplier X and the best grease ‘Grease C’ configuration shows the most economical solution in this research. Thus lubricant quality plays a vital role in the best practice lubrication.

- Appropriate applicator bar and grease configuration can develop best opportunity and economic benefits for the rail operators.
- Lower grease consumption and higher carry distance would be the best savings for rail operators. But lower grease consumption and lower carry distance do not fulfil the economic needs of the operators. Wheel/rail protection with longest carry distance are the goal of wayside lubrication.
- Serviceability, operability, maintainability, reliability and economic benefits are highly achievable with electric long bar units.
- Long bar units can be implemented both in high MGT or low MGT lines with appropriate grease loading frequency and maintenance but hydraulic short bar units are only applicable in very low MGT lines.
- Technologically improved and highly capable equipment and highly effective grease should be used by the rail operators to maximise financial benefits and maximise assets and resources utilisation.

# Chapter 7

# OBSERVATION OF GREASE TRANSPORT MECHANISM

## 7.1 Introduction

The grease transport mechanism is a complicated phenomenon as many controllable and uncontrollable parameters are involved in this process. As wayside lubrication is applied by static trackside equipment, grease has to be transported from the lubricator sites in the direction of train operation, either uni-directional or bi-directional, and has to lubricate the nearby curves. Effectiveness of wayside lubrication depends solely on successful transport of grease throughout curves over the maximum achievable distances while still maintaining the friction level at the acceptable value. In hi-rail and on-board lubrication systems, grease is “transported” by being applied while running the hi-rail vehicle or the locomotive along the track. Grease transport mechanism is a repetitive give-and-take exchange of grease between rails and wheels. While travelling through the wayside lubrication site, wheels pick up the grease beads and continuously distribute lubricant to the wheel/rail interface along the track. Though there are number of studies available on industrial lubrication mechanisms, in depth research on the grease transport mechanism and its significance to rail systems is hardly available. Some valuable thoughts and practical issues about the grease transport mechanism and influencing parameters have been introduced in Thelen and Lovette (1996). This chapter presents the model development of grease transport mechanism.

The model of the grease transport mechanism proposed in this thesis is based on a volumetric loss process as the grease bead proceeds from the lubricator site in the direction of traffic. The impact of wayside equipment, applicators bars, grease bead, grease composition and properties on the grease transport mechanism in wayside lubrication is presented in this chapter and also discussed in the final report of CRC Australia (2014).

## 7.2 Effects of Parameters on Grease Transport

### 7.2.1 Wayside Equipment and Applicator Bars

Wayside lubrication equipment and applicator bars make a great contribution to the grease transfer and transport mechanism. With perfect location and placement, this equipment can ensure a high degree of effective grease distribution within the interactive moving wheel and static rail system. Evolution of wayside lubrication equipment has contributed significantly to the proper application of grease. The latest electronic based electric lubricators are highly precise compared to the older technology mechanical and hydraulic equipment.

#### *7.2.1.1 Mechanical and Hydraulic Lubricators*

Both mechanical and hydraulic lubricators are installed in the transition spiral of the circular curves with short applicator bars and should deliver grease to each wheel. There is no control of wheel frequency or interval between applications of grease because it works based on the each wheel striking an actuator plunger. The mechanical plunger completely depends on the wheel strike force for the expected grease delivery volume. Damage due to excessive wheel strikes causes progressive changes of height and shape of the plunger. When the conical plunger tip becomes flattened or mushroom shaped, it affects or stops grease delivery. Though the grease applicator bar has a specific number of grease ports, due to excessive blockage and very weak actuation from flattened plungers, it is frequently found that only one or two ports are able to push a small amount of grease out. Therefore, the circumference of passing wheels remains without grease except for one or two points gained at the active ports. This causes a catastrophic failure of lubricant distribution due to the lack of sufficient grease. Field investigations revealed that it was frequently the case that either excessive and/or poorly directed pumping of grease was resulting in wastage and top of rail contamination, or an undersupply of grease was resulting in poor or nil lubrication.

Even if the plunger works, hydraulic fluid leakage in hydraulic lubricators may totally weaken the grease pump and stop grease delivery which causes the rails and wheels to remain unlubricated. All of the above situations affect the grease transport mechanism and may result in wheel/rail lubrication failure. Figures 7.1 and 7.2 show the conditions often found in mechanical and hydraulic lubricator sites. Common conditions on lubricator sites show grease has not been transported along the rails and which are shinny due to high friction, whereas the accumulation of large amounts of grease occurs on the track around the application site. Such situations arise not only from equipment faults, but also the poor quality of some greases and from deficiencies in lubricator system maintenance practices. On high MGT lines, these units need extremely frequent filling and servicing which may not be manageable due to resource restrictions, spare parts or lubricant availability problems and site access issues.

(a) (b)

Figure 7.1: Severe conditions on mechanical and hydraulic lubricator sites: (a) Severe blockage or very weak actuation force lead to delivery of only one small grease bead, (b) Severe wastage of grease (Uddin et al. 2010a)

(a) (b)

Figure 7.2: (a) Close look of an applicator bar on a hydraulic lubricator site with shiny rail and grease beads fallen away from rail head, (b) Rail damage from severe RCF after a hydraulic lubricator

The mechanical and hydraulic lubricators have no condition monitoring options except the site visit to find any failure or damage. Failure or damage only can discover after occurrence and it is completely remains unknown when and how long before the lubricators failed and lubrication stopped. Figure 7.3 shows a severe damage of rail gauge side by wheel contact without any lubrication. It was amazed to see that the gauge face material of high rail in the curve went to slices and fallen off from the rail head.

(a) (b)

Figure 7.3: Severe rail gauge side damage due to no lubrication from hydraulic lubricator failure: (a) Severe gauge face wear on a sharp curve, (b) Close look at the rail gauge face on a sharp curve (Chattopadhyay et al. 2010)

The above scenarios show the severe effects that the limitations of the lubrication units could have on rails and wheels if they are not capable of ensuring grease engagement between the wheel/rail contact surfaces. These hydraulic unit applicator bars also have limitations regarding the ability to install the applicator bars at the appropriate height on the rail gauge face to apply the grease into the wheel/rail contact area. Therefore, most of the grease may not be picked up by wheels and falls onto the ballast.

Investigations on Australian heavy haul railway networks show that grease dispensed from hydraulic or mechanical units often did not carry far along the track. In these cases, a major portion of the grease is wasted through leaks from the applicator bars and assemblies (Uddin et al. 2010). The grease carry ends within a short distance of the lubricator and the rest of the track remains unprotected.

#### *7.2.1.2 Electric Lubricators*

To overcome the common mechanical problems and excessive interruptions to the operation of mechanical and hydraulic equipment, modern technology electric lubricators (Figure 7.4) are being introduced and have proved their performance in this research.

Figure 7.4: Electric lubricator and its main components onsite

Electric lubricators with high performance grease have shown an improved degree of effectiveness in grease distribution onto wheels at the optimum rate that can result in lubricant transport more effectively over the curves. However, the results of the field trials showed significant variation within various combinations of electric lubricators, applicator bars and various greases.

These units strongly contribute to effective grease transport by delivering the controlled amount of grease at recommended intervals and rates with high levels of reliability. The failure rate in electrical equipment is negligible compared to mechanical or hydraulic equipment. The electric motor driven pump has no change in the grease feed rate unless the operator change the feed rate. Same feed rate is maintained throughout the whole interval of servicing and grease loading. Unless pump failure or any blockage in the delivery hose, electric units deliver grease with significant pressure to ensure grease bead height and size along the applicator bars. Even electric units may not generate expected level of grease transport along the track if the grease quality is poor. The performance of various electric units with various combinations of applicator bars and grease has been discussed in data analysis chapter (Chapter 4). Electric lubricators with remote condition monitoring technology may contribute exceptionally to maintain certainty of continuous lubrication. Highly effective sensors are implemented in the electric lubricators and support the remote performance monitoring system for various purposes. The following sensors were used:

- Grease level sensor.
- Optimal pressure sensor.
- Temperature sensor.
- Battery voltage sensor.
- Motor current sensor.
- Traffic direction sensor.
- Optimal motor shaft encoder.
- Rain sensor.
- Door open/close sensor.

Remote condition monitoring ensures the availability of useful real time data, particularly on the grease level in the tank and the current feed rate. This enables a proactive maintenance strategy to be developed to maintain lubricators before any failure.

Any discontinuity of grease feed can be instantly detected from real time motor ampere and power consumption data and can attend to the breakdown straightaway.

To maintain the achievement of the grease transport mechanism, the units’ feed rate can be changed for any reason such as traffic frequency changes, seasonal effects or resource constraints.

Hundreds of lubricator units in various networks can be monitored from an established control room so the many data reports can be effectively handled. Remote condition monitoring completely eliminates the unknown failure of lubrication.

Alarms and warnings can be produced on critically low grease levels, door open/close, critically low battery voltage and changes to the digital control box.

#### *7.2.1.3 Grease & Applicator Bars*

The grease and applicator bars play an important role in the grease transport mechanism. These bars are not effective if they are not placed appropriately on the head of the rail at appropriate locations along the track, or are not capable of precise grease application, no matter what type of metering equipment is employed. Success of wayside lubrication is highly dependent on the amount of grease picked up by wheels from the delivered grease bead at the application site. Grease beads have to spread grease over the wheel surface effectively so that each wheel can collect enough grease for delivery further along the track. Bar installation instructions for Long bar/ XL bars specified the bar height of 1.055 inch (26.80 mm) to 1.25 inch (31.75 mm) from the top of the thin blade on the applicator bar to the top of the rail head. The insufficient grease carry for some tests was due to the bars being too low so that the wheels were not picking up grease. For the experimental procedure, bar heights were kept constant for the entire trial. Sufficient grease transport can occur at lower bar heights if the grease beads maintain their height and, in this case, bars are less likely to suffer wheel impact damage. From the trial it has been observed that 1.055 inch (26.80 mm) to 1.25 inch (31.75 mm) appears to be too low for the wheels to pick up the grease.

Figure 7.5: Weak grease bead fallen onto grease guide bar which reduces grease pick up due to not contacting wheels

Limitations of bar height for effective grease pick up can be managed by a strong grease bead. Field trials showed that grease pick up from a strong grease bead (Figure 7.6) was much higher compared to weak and dull grease beads (Figure 7.5). Figure 7.5 and Figure 7.6 show the difference in quality and strength of grease beads projected out from applicator ports.

Figure 7.6: Strong grease bead with full column height (CRC Australia 2014)

Field trials showed that, for the same type of grease, transport has achieved for longer distances from two long applicator bars on tangent track sites compared to two short applicator bars on the high rail in the curve transition sites. Due to the placement location, applicator bar length and grease bead size, long applicator bars show superior grease transport compared to short applicator bars. Short bars sites (Figure 7.7b) have more wastage and splash compared to long bars sites (Figure 7.7a). Long bars applied grease to a greater length of the wheel circumference and distributed grease on both rails. Grease can be transported by wheels to be applied on the high rails of both left hand and right hand curves from one single pair of long bar units.

(a) (b)

Figure 7.7: Long bar and short bar application sites: (a) Long bar site with two long bars on each rail in tangent track, (b) Short bar site with two short bars on high rail in the curve transition (CRC Australia 2014)

Figure 7.8 shows the direction of grease travel on a short bar site and how a large amount of grease is wasted because it is delivered too far away from the rail gauge face contact interface.

(a) (b)

Figure 7.8: Direction of grease movement from grease bead on a short bar site (on high rail in the curve transition): (a) Grease moving out from the gauge face during wheel strike on the grease bead, (b) Grease movement across the rail head from gauge face towards the field side of the rail

Effective wayside lubrication needs to maintain an effective grease transport mechanism which will ensure continuous grease distribution over the expected length of the rail gauge face and at the wheel flange contact area. Long bars with grease guide bars helps significantly in grease transfer to the wheels. Without fall protection, low quality grease with weak beads dropped off the gauge face application area and an inadequate amount of grease is available to be picked up.

Figure 7.9: Freshly distributed grease beads fallen off from the gauge face and therefore an inadequate amount of grease will be picked up in the wheel/rail contact area

Even if there are grease guide bars to support grease transfer, it will not be successful if the grease beads do not stick to the gauge face with strong column.

Therefore, the transportation of grease has a catastrophic failure and lubrication was totally ineffective. In the field trials, the location and installation height of the applicator bars was found to be critical to the success of the wheels' ability to pick up grease from the bars and carry it down the track (Thelen & Lovette 1996; Uddin et al. 2010).

### 7.2.2 Grease

Grease has been applied in wayside rail lubrication for decades. Grease application in rolling contact bearings has a defined environment which is protected from foreign contaminants by effective sealing. In wayside lubrication grease in an uncontrolled environment with an unsealed boundary which results in an undefined number of variables. The function of grease is to remain in contact with and lubricate moving surfaces without leaking out under the force of gravity, centrifugal action or being squeezed out under pressure. Its major practical requirement is that it retains its properties under shear forces at all temperatures it experience during use. There is a total lack of sealing or controlled boundaries along the rail length to contain the grease within a certain area. It is therefore a great challenge to keep the grease only on the expected area of rail gauge face contact to maintain effective lubrication.

There are three basic components that contribute to the multi-phase structure of lubricating grease; a base fluid, a thickener and very frequently, in modern grease, a group of additives. The function of the thickener is to provide a physical matrix to hold the base fluid in a solid structure until operating conditions, such as load, shear and temperature, initiate viscoelastic flow in the grease. To achieve this matrix, a careful balance of solubility between the base fluid and the thickener is required. The primary type of thickener used in current grease is metallic soap. These soaps include lithium, aluminium, clay, polyuria, sodium and calcium. Lately, complex thickener-type greases are gaining popularity. They are being selected because of their high dropping points and excellent load-carrying abilities.

Grease is required to possess few salient properties such as ease to pump called, pumpability. Resistance to deformation called consistency, resistance to liquefying called dropping point, chemical stability under high temperature called oxidation stability, tendency to remain semifluid stiffening t low temperature called pour point.

Additives play a vital role in effective lubrication in mechanical components. Other than separating the two mating surface a lubricant has to cater other duties also such as cooling, adhering to the surface, protecting surface under high pressure, reducing wear, friction, resistance to change in viscosity, due to temperature change, sealing the bearing contact from environment, ease in pumping at a required temperature, corrosion resistance etc. these extra duties are performed by the chemicals present in the additive package. In greases controlled bleed rate is required for so that oil is supplied in the contact when it is needed and that function is performed by the required additives.

There are empirical relations for calculating grease in bearings, which is not possible in rail curve lubrication. The amount of grease required for lubricating a contact is in fact too small and it depends upon the lubrication mechanism that contact is subjected to, such as boundary, mixed, elastohydrodynamic or hydrodynamic regime. Lubrication regime is primarily governed by the load, speed and viscosity of the base oil. However the actual quantity of lubricant needed to lubricate a contact is much more than the contact needs and is governed by the application method and cooling requirements of the lubricant itself.

Grease are identified by their consistency which is measured by penetration number or popularly known as NLGI. Penetration number of grease is associated with the grease chemical composition including additives, viscosity of base oil and type of thickener used. However type of lubricant application also decides quantity of lubricant used. To ensure that the required quantity of grease enters the rail-wheel contact, enough grease must be picked up by the wheel and lesser amount should be wasted in transferring from the source tot eh contact surfaces. Wastage of grease takes place due to leakage and transfer from one medium to another in an open system and it needs to be minimised.

Amongst various parameters which affect grease usage, the most critical ones are vehicle speed, load, nature of the contact interface of mating parts (rail and wheel), environmental conditions such as humidity/moisture, vegetation and dust contaminants, wheel/rail interface, frost and extreme temperatures, etc. The wheel/rail gauge face interface generates high pressure and temperature during curving which significantly affects grease properties. In the mechanism of the grease lubrication the base oil of grease provides the lubrication, therefore complex grease structure is required to hold the base oil and release it when required in a controlled manner by a phenomenon called viscoelastic flow. Too much or too little release called bleeding of base oil from grease is not and that depends upon the consistency of the grease which is represented by penetration number or NLGI.

The other requirement is that the grease picked up by the wheels once, lubricates longer rail distance or covers longer carry distance. It has been observed that some greases have generated expected level of friction with a very good layer of grease on the gauge corner but ends up within lower carry distance as compared to other grease with same NLGI. It has also been observed grease has dropped off from the rail gauge corner within the first curve after lubricator and it has not been transported to further curves. Grease must have the necessary properties to prevent it from running off the gauge face within a short distance or after a small number of load cycles called tackiness. The grease layer must be sustained at the gauge face within the expected load range and weather conditions. Rail curve grease should have variety of properties for high retention, effective lubrication and longer life such as high dropping point , low pour point, necessary additives, negligible bleeding rate, high lubricity, excellent shear stability, good adhesion to the rail at the required friction, water washout resistance and chemical stability. Poor retention of the grease could also be due to variety of reasons where operating parameters, grease structure, additives and environmental conditions all may contribute.

A structured method of grease selection for rail curve lubrication is not a common practice. Many heavy haul rail operators have been using very low performance grease for decades due to a substantial lack of understanding of grease properties and their requirements to achieve maximum benefits. Performance reviews or economic investigations are rare in commercial applications, which have extremely high significance of maximising economic returns in this competitive business environment. Currently, proper specifications or guidelines for the selection of the appropriate rail curve grease in heavy haul applications are hard to find.

Performance analysis of various tested greases has been discussed in the data analysis chapter (Chapter 4) of this thesis. The field trials have shown that different greases show different performance levels based on carry distance, grease sustainability on the gauge face and splash out from the application area.

#### *7.2.2.1 Grease Chemistry*

There are three main components of grease i.e. base oil, thickener and additives. Grease chemistry and its combined functionality are highly complicated phenomena. Types of soaps and their chemical process are optimised in such a way that desired properties are achieved. Additive chemistry is again a highly confidential area and no company preferred to share their composition. Grease chemistry should be as effective as possible for the purpose of its uses.. The field trials in this research revealed that different greases give different level of performance in rail curve lubrication under similar operating conditions. Though NLGI number of each grease is the same their additive packages are different and their chemistry is highly confidential. Therefore the only information about their properties is available is from the data sheet supplied by the manufacturer. All test greases are mineral oil based none of the grease used in the field tests was synthetic oil based. Recently, synthetic oils have been used in grease as an alternative to mineral oils where grease is expected to be operated in extreme conditions. However such greases need to be cost effective in rail curve lubrications.

Thickeners in grease form an interlocking matrix of particles (Figure 7.10) which trap the base oils and allow the establishment of a continuous grease network. The thickener and the base oil in a grease both take part in the lubrication process, and the performance is the result of their combined interaction. The most commonly used soap type greases are calcium, lithium, aluminium, and sodium. Complex greases are developed by using multi thickeners and in some cases variety of base oils.

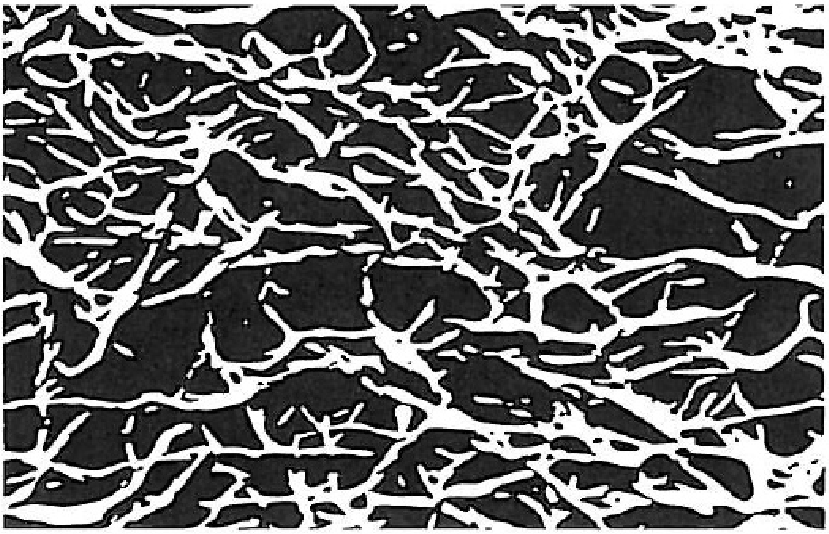


Figure 7.10: Thickeners form an interlocking matrix of particles which construct a fibrous structure in soap based grease (Stachowiak & Batchelor 2005; Uddin et al. 2012; (CRC Australia 2014))

#### *7.2.2.2 Effects of Additives in Rail Curve Grease*

Special packages of additives can improve and sustain high performance of grease; these include anti-oxidants, rust preventers, tackiness, and anti-wear and extreme pressure (EP) additives. Appropriate additives selection must suit the climatic conditions of the rail network and the operating load conditions. In rail curve lubrication, as grease should stay on the rail and wheel contact area for extended periods of time it must sustain its original properties and condition without breakdown. Flange-to-rail rubbing contact involves extremely high concentrated energy dissipation which leads to high surface temperatures and causes rapid evaporation and oxidation of grease. Anti-oxidants are therefore desirable in rail curve grease. Rail curve applications are exposed to harsh climatic conditions where rain, high humidity, temperature variations and other drastic conditions affect the performance of grease.

Tackiness additives are utilised to impart a stringy texture and to increase the cohesion and adhesion of the grease to the surface. Field investigations found that some grease stays on the contact zone and many others disappear quickly. There is a strong tendency in many types of grease to travel quickly downward from the gauge corner.

Anti-wear and extreme pressure (EP) additives improve the load carrying capacity of rail curve grease.  EP additives react with the surface to form protective films which prevent metal to metal contact and the consequent scoring or welding of the surface. The solid additives most commonly used as anti-seize and anti-scuffing compounds are graphite and molybdenum disulphide.

Consistency is used to measure the shear strength of grease. The consistency must be sufficient enough to remain as grease in the sliding or rolling contact. Too hard a grease is difficult to pump and carry through the delivery hose, may cause ‘channelling’ where the rolling or sliding elements cut a path through the grease and cause lubricant starvation (Stachowiak & Batchelor 2005). It has been observed that the grease pumps get clogged due to too thick a grease lump and the grease cannot pump through the delivery system. Again, if the grease is too soft it is ejected away from the rolling wheel surface and away from the contact area. It is necessary to conduct field and laboratory testing to determine the desirable grease consistency for the desired application in heavy haul rail curve lubrication.

The grease consistency and temperature relationship (Figure 7.11) shows changes in the consistency of typical grease with temperature. Consistency or penetration number of a grease vary with the temperature and the temperature at which it loses its semifluid properties cannot regain the same structure even after lowering down the temperature. At this temperature, the grease structure breaks down and it becomes liquid and this temperature is known as dropping point.

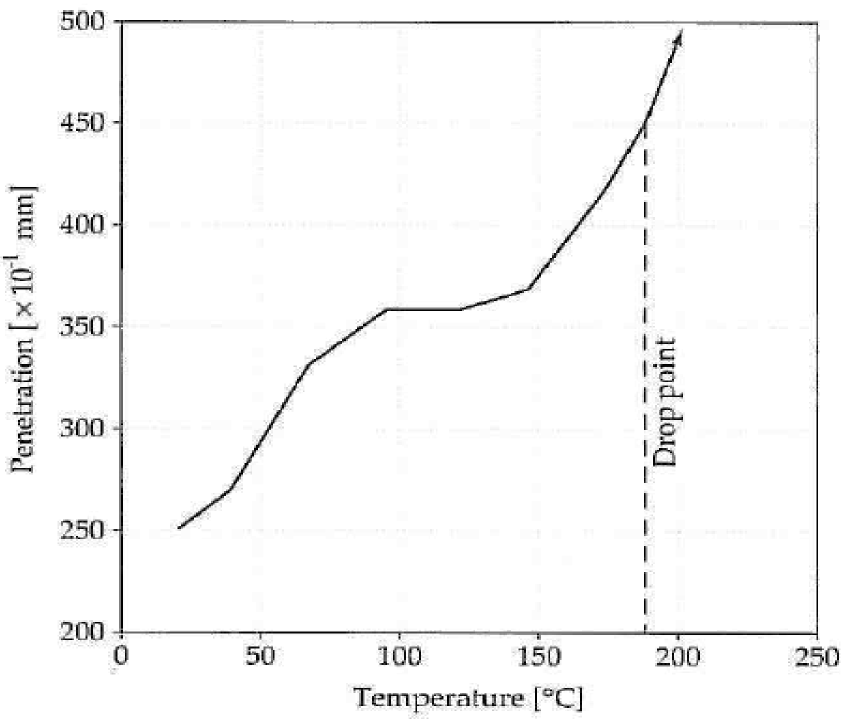



Figure 7.11: Grease consistency variation with temperature in terms of penetration for sodium soap grease (Stachowiak & Batchelor 2005; Uddin et al. 2012; (CRC Australia 2014))

Real life data shows that wheel temperatures may rise beyond the dropping point of the grease and may cause loss of consistency and grease effectiveness. Figure 5.12 shows the temperature data of heavy haul train wheels operating on the QR network as detected by wayside hot box sensors. Though there was intention to collect wheel temperature data in rail wheel interface it was not possible to determine wheel temperature while train was running through the test sites due to lack of resources and restriction. Accordingly, the wheel/rail interface temperature needs to be minimised. Various greases may differ significantly in the level of damage they incur due to mechanical load. The field trials show that grease had been burnt and left only a graphite residue on the gauge face which was not providing expected level of friction at all and lumps on the foot of the rail. It was also out of scope of this research to develop a standard pumpability test. This research was not aimed at the improvement of grease and lubricator equipment.

Dropping point plays a significant role in the grease transport mechanism. If the grease reaches its dropping point under extreme load on the wheel/rail gauge face, it will liquefy and flow from the rail gauge face and wheel flange. As a result, there will be severe loss of grease and an insufficient amount will be available to be transported further along the rail. Investigation shows that many rail curve greases have dropping points lower than a significant percentage of operating temperatures. Different grease has different dropping point and it would vary significantly from grease to grease. The wheel temperature detected in hot box is goes from 50 degree Centigrade above the ambient up to couple of hundred degree Centigrade. Comparison between Table 5.10 and Figure 5.12 could provide a basic understanding of wheel temperature distribution against the dropping point of each of the test grease. Weight losses of greases due to evaporation can be quite drastic for wheel/rail contact. Volatile compounds and products of thermal degradation contribute to weight loss, thickening of lubricant; higher shear resistance and higher temperature during the surface interaction. A visual observation indicated that in field investigation oil separation and hardening of grease significantly influenced the grease distribution and carry distance.

## 7.3 Grease Transport Mechanism

Although grease plays a significant role in heavy haul lubrication, research on lubrication transport mechanisms is rare. According to Thelen and Lovette (1996), the lubrication transport mechanism in a rail curve application can be optimised quantitatively under laboratory conditions, but many trade-offs must be made in the field between an ideal set up and what is practically achievable. Human factors have as much impact on the transport of lubricant to the rail as any technical issue. Even though the grease is applied directly to the gauge corner, its performance is still heavily influenced by the frequency of trains, frequency of grease application and the amount of grease applied.

Higher axle loads have a strong influence on rail and wheel wear and fatigue. This makes wheel/rail lubrication a crucial requirement for the cost-effective operation of today’s rail transport networks. Though premium rails are commonly used in curves of less than 349m radius, turnouts and switches in North American railroads still experience unacceptable rates of wear without lubrication.

In wayside lubrication, grease is intermittently applied to the wheel/rail contact area so that the present grease is always replenished. At predefined intervals, a controlled amount of fresh grease is applied to the rail gauge corner and each wheel picks up an amount of grease from the these grease beads on the applicator bars at the lubricator site. Successful grease transfer from the beads to wheel flanges ensures that continuous grease exchange takes place between wheels and rails along the rail gauge face and at the throat of the wheels (the curve in the profile where the wheel flange and tread come together). Along this continuous journey, the challenge is to maintain the existence of satisfactory grease coverage with which predefined levels of friction must be achieved. Figure 7.12 shows the overall grease transport mechanism in the wheel/rail interface which shows grease exchange from rail to wheels and back to rail, plus the wastage of grease along the track from both wheels and rails. Minimising wastage of grease is likely to contribute significant improvements in the grease transport mechanism and therefore provide more effective lubrication.

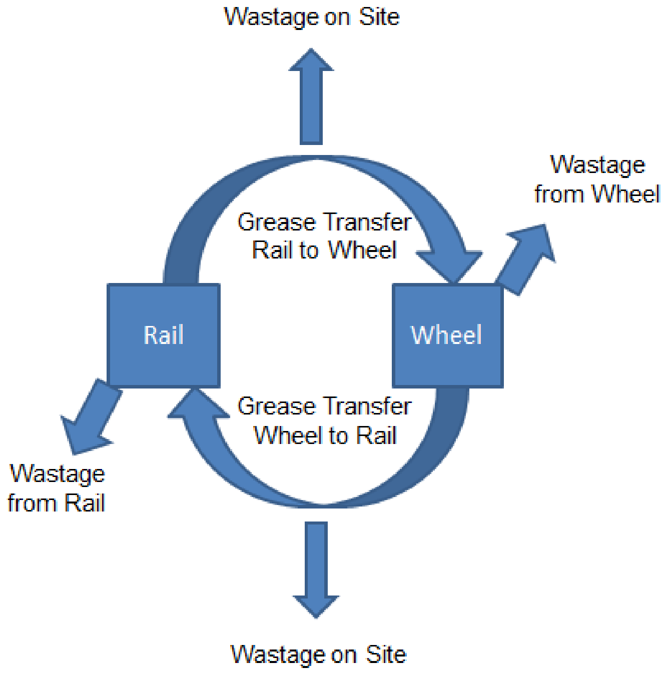



Figure 7.12: Overall Grease Transport Mechanism in the wheel/rail interface shows grease exchange in between rail and wheels and wastage of grease in this mechanism

Figure 7.13 demonstrates a sequence of collected images of rail and wheels showing the real grease transport mechanism at the wheel/rail interface. As the train travels through the wayside lubricator site, wheels strike the grease beads and squeeze them in all the freely available directions of travel. This results in displacement of grease both in the direction of wheel travel, and also downwards along the gauge face. Displacement towards the top of the rail head is limited by wheel/rail contact on the rail head. A significant portion of the applied grease may be picked up by the wheels and the remainder may be wasted. Through the continuous rolling and sliding contact between wheel and rail as the wheel advances, grease beads become thinner and increase in length. After a small number of wheel revolutions, the bead on the wheel flange forms a continuous layer of grease and covers the whole wheel circumference. This will produce a corresponding continuous grease layer on the gauge face of the rail.  Eventually the grease layer becomes so thin that it can no longer provide the required lubrication function, and the coefficient of friction will increase.


Figure 7.13: Grease transport mechanism progressing from grease beads on applicator bars towards the rail gauge face, and eventually to a continuous film on both rails and wheels (CRC Australia 2014)

Descriptions of the images in the Figure 7.13, viewed from left to right, and top to bottom are as follows -

- Grease is delivered to the distribution bars on the rail gauge face.
- The wheels pass over the distribution bar and pick up some of the grease.
- The wheels travel forward, carrying the grease on the wheel flange/throat.
- The wheel flange approaches the rail after a few rotations.
- After several rotations, the grease has been spread over the circumference of the wheel.
- Several curves away from the applicator site, the grease has been thoroughly spread over the circumference of the wheel. The film is the result of many interactions between the grease on the wheel and the grease on the rail.
- The grease has been spread smoothly over the wheel circumference after a considerable travel distance.
- Grease is still evident on the wheel surface after 1.5km.

- Smooth grease cover on the gauge face of the rail after 2.5km from the application point.

## 7.4 Conclusions

The grease transport mechanism in wayside lubrication is a complex phenomenon. Successful grease transport along the rail gauge face depends on various controllable and uncontrollable parameters. Various equipment configurations, their limitations/performance, appropriate location, flexibility of installation, the quality and various properties of greases, and column strength of the grease bead contribute significantly to the grease transport mechanism. Two long applicator bars on each rail achieved the longest grease transport distance with the best grease used in the field trials. Wayside lubrication in the transition of the curves with the worst performing grease gave the most wastage of grease on site and along the track and achieved the least grease transport distance.

# Chapter 8

# SUMMARY, CONCLUSIONS AND RECOMMENDATIONS

## 8.1 Summary

In heavy haul railway operations, rail and wheel contact surfaces continuously wear out and service life reduction occurs due to pressure from increased volume of traffic density and higher and higher axle loads with no proper strategic practice of lubricating rails and wheels. Reduction of friction at the wheel/rail interface on the rail gauge face can be brought down to an acceptable level only by effective lubrication. The method of achieving effective gauge face lubrication was the focus in this research. This study conducted real life field trials on rail curve lubrication to determine effective and economic practices of wayside lubrication based on appropriate placement location and positioning of wayside lubricators, implementation of long applicator bars or short applicator bars with various greases with the necessary properties and additives for heavy haul load application. Field trials established the integrity and authenticity of the data collections and evidence of the track activities and lubrication condition during live trials. The findings of this research are summarised below.

### 8.1.1 Appropriate Selection of Lubricator Technology

Thorough investigations and data analysis showed that:

- Technologically advanced electric lubricators perform significantly better compared to the older mechanical and hydraulic lubricators.
- Mechanical and hydraulic lubricators are not suitable for high MGT lines due to their physical, functional and life cycle cost constraints.
- Electric lubricators with large grease tanks are highly economic to run for high MGT lines due to very low operating and maintenance costs.
- Long term strategy for heavy haul should be to remove mechanical and hydraulic lubricators from service and adopt highly effective electric lubricators.

- Mechanical and hydraulic lubricators cost considerably more for operation and maintenance compared to electric lubricators.
- Electric lubricators are highly reliable compared to mechanical and electric lubricators. Electric lubricators can push the rate and pressure of grease to the delivery points by electronic control settings, electric motor operated pump, solar or electric power supply and have numerous electronic sensors in place to manage lubricator performance.
- Remote performance monitoring technology boosts the reliability, operability and maintainability of electric lubricators so that it makes the lubricators extremely attractive for wide application in a heavy haul network which covers hundreds of kilometres of rail lines.

### 8.1.2 Selection of Applicator Bars

There is a tremendously positive effect of appropriate applicator bar selection in achieving successful wayside rail lubrication. It is a great challenge to transfer grease from the rail gauge face to wheel flanges at the appropriate location and transport it throughout the maximum possible carry distance to maintain the expected level of friction. It depends on the type of bars and the appropriate location of placement. Applicator bars are mounted on the gauge side of the rails to the lower part of the rail head to deliver grease to the wheels. Suitable positioning must be confirmed by visual inspection, MiniProf rail profile measurement, and confirmation of the desired wheel/rail contact pattern based on correct wheel gauge and track gauge (including appropriate width adjustment on curves) as determined by a track gauge bar. To avoid hunting and the risk of damaging the applicator bars, track gauge width must have minimum clearance from the distance of wheel flange to wheel flange.

Applicator bars play a significant role in the grease transport mechanism, and are not effective if they are not located properly or not suitable for precise grease application, no matter what type of metering equipment is in place. Currently available long applicator bars (1400 mm in length) are placed in the tangent track before curves and short applicator bars (600 mm in length) are placed in the transition spiral of curves.

Field trials and investigation of existing practices in this study showed the following effects of both types of applicator bars:

- As long bars are installed on both rails in tangent track, both left hand curves and right hand curves are lubricated by one unit. Therefore the number of units required to lubricate the track is halved compared to short bars.
- Long applicator bars in tangent track are technically and economically attractive compared to short applicator bars in the transition spiral of curves. Long bars deliver grease to the complete circumference of a rolling and sliding wheel and significantly contribute to effective lubrication over a much longer distance compared to short applicator bars.
- Considering rail length protection, application of lubricant using long bars consumes comparatively less grease compared to short bars.
- Long bars do not need to be removed during curve grinding cycles; this reduces the work requirement for the lubricator maintainer.
- Short bars rely on their placement in the transition spiral of mild left and right hand curves. Two short bars are used in each transition and their placement is just at the point where the wheel throat starts to contact the gauge corner. Mild curves are used to properly distribute the grease around the throat area of the wheels to ensure more effective lubrication in sharper curves along the track.
- Contributions of different bars in the grease transport mechanism play an important role in improving the achieved carry distance. In this study, field trials on an Australian heavy haul network found that 1.6 times longer carry distance was achieved with long bars compared to short bars. Figure 5.6 in Chapter 5 shows the performance (in carry distance) of electric lubricators with long bars in tangent track and short bars in transition spirals of curves with the same grease, grease C.
- Table 5.4 shows ‘ACOF GF-Hi (2 LB-ER, Supplier X, Grease C)’ has the accepted level of friction up to Curve 14 and demonstrated the longest carry distance of 4.623 km, whereas the ‘ACOF GF-Hi (2 SB-HR, Supplier X, Grease A)’ has the accepted level of friction only up to Curve 2 and demonstrated the shortest carry distance of 0.34 km.

### 8.1.3 Settings of Grease Application

The application rate and interval between applications of grease is controlled by the electronic control box. The optimal setting of grease application rate can be achieved by use of the ‘splash test’. Starting with the manufacturer’s recommended settings, progressive adjustments were made based on the condition of splash and rail head contamination with grease until an optimal setting was reached. Once the optimal setting was achieved, lubricant application was observed for more than 50000 wheels for each test segment and it was found that very little grease had been scattered away from the rail gauge side. Table 8.1 shows the recommended optimal pump setting for each type of applicator bar and grease.

Table 8-1: Optimal grease application rate/ pump for each type of bar and grease, achieved by splash test

| Type of Bar | Name of Test Grease | Pump Activation | | Wheel Count | Application Rate/Pump (gm) |
|---|---|---|---|---|---|
| | | Seconds | Wheels | | |
| Short | A | 0.2 | 18 | 25195 | 5.08 |
| Long | A | 0.25 | 12 | 124500 | 6.80 |
| Long | C | 0.25 | 12 | 359497 | 6.80 |
| Long | D | 0.25 | 12 | | 6.80 |
| Long | B | 0.25 | 12 | | 6.80 |
| Long | E | 0.25 | 12 | 518109 | 6.80 |
| Short | C | 0.20 | 18 | | 5.08 |

### 8.1.4 Location of Short Bars in Curve Transitions and Long Bars in the Tangent Track

When wheels pass through the short applicator bars in the transition spiral of a curve, the grease transport mechanism is severely affected by wheel motion and alignment with the rail gauge face. The planetary dual motion of wheels causes severe sliding or flanging between rail and wheel. The rotary motion of wheels in a curve takes place at an angle (angle of attack) to the rail’s longitudinal axis. In contrast, the rotary motion of wheels at the tangent lubricator sites is parallel to the rail axis and does not cause severe sliding and flanging as occurs at the short bar sites.

Due to the bogie steering capability in curving, instead of travelling parallel to the rail, wheels travel obliquely at an angle to the rail. The angle between the tangent of the rail and the direction of wheel rotation is equivalent to the angle of attack.

This angle of attack of wheel rotation causes the grease to be scattered away from the longitudinal axis of the rail towards the field side and gauge side of the track; this effect is known as “fling off”. Another issue is that, when a wheel is travelling fast with an angle of attack, it squeezes the grease bead out and wastes the grease around the short bars.

Provided there is no hunting, proper site selection and positioning can significantly reduce the amount of grease wastage and improve the amount of grease being transferred and transported from long bars site.

If a significant portion of grease is lost on site, very little remains to be distributed along the track for friction control. Grease “fling off” may vary based on the grease bead size and quality. Short bars have a bigger bead size than long bars which causes more wastage of grease compared to long bar units.

### 8.1.5 Effect of Grease

This research found that grease composition has significant effects on effective lubrication. Different greases with the same applicator bars achieved very different carry distances. The properties of grease have to be optimised to achieve the longer carry distance. The selection of grease for the long term operation is a decision that should be made only after field trial results have been analysed.

## 8.2 Conclusions

With optimal combinations of grease and state-of-the-art lubricator equipment, it is possible to make a significant saving in the cost of track maintenance. Field trials have been carried out to develop techniques for measuring lubrication effectiveness and the effectiveness of different types of lubricants and applicator bars. Different applicator bars have been monitored closely and it was found that the applicator bars has significant effects on grease performance. Wheel/rail contact pattern plays a crucial role in grease distribution along the curves. This research showed that long bars achieved better performance compared to short bars while the same grease was applied. The Australian heavy haul railways may benefit significantly from utilising long bars with high performance grease, saving maintenance, repair and breakdown costs for both wheel and rail assets.

Wayside lubrication is a proven technology in heavy haul railways. Savings are substantial if implemented in line with best practice guidelines. With optimal combinations of grease and state-of-the-art lubricator equipment, it is possible to make a significant saving in the cost of track maintenance. Field trials have been carried out to develop techniques for measuring lubrication effectiveness and the effectiveness of different types of lubricants and applicator bars. Extensive field testing has covered several grease performance with several combination of long and short applicator bars. The applicator bars from different suppliers also performed significantly differently. Applicator bars (long bars) in the tangent track and applicator bars (short bars) in the transition spirals of curves were monitored closely and it was found that the applicator bars has significant effects on consumption of grease, transport of grease and carry distance. It was found that different greases with the same applicator bars achieved significantly different results in carry distance in the same track and traffic conditions. It has been noticed that the wheel/rail contact pattern has a significant impact on grease sustainability and transport. Grease in the curves with conformal contact showed very good grease coverage compared to curves having single point or two-point contact. Quick burn out of grease due to excessive load in single point or two-point contact occurred, whereas a very good coverage of grease was observed in the curves with conformal contact. This trial also showed that long bars in the tangent track achieved superior performance compared to short bars in curves with the same grease application. The Australian heavy haul railways may benefit significantly from utilising long bars with high performance grease, saving maintenance, repair and breakdown costs for both wheel and rail assets.

The technical and economic data analysis provided an in depth understanding and decision making power to choose appropriate cost effective practices. Overall the following conclusions can be made:

- Electric lubricators with long bar applicators installed in tangent track are the most effective set-up for wayside lubrication.
- Electric lubricators with long bar applicators utilising the best extreme pressure grease can create the most effective lubrication with longest carry distance.
- In addition to achieving an effective friction level, the longest possible carry distances are also desired from the wayside lubrication system.
- In the field tests, the electric units with long bars from Supplier X using the best grease configuration (Grease C) achieved the longest carry distance.
- Many other greases, including the current practice, did not perform well with any equipment set-up.
- Combinations of the appropriate applicator bar and grease configuration can provide the best opportunity and economic benefits for rail operators.

- Lower grease consumption and higher carry distance would provide savings for rail operators. But lower grease consumption and higher carry distance alone do not fulfil the desire of economic benefits for operators. Optimum wheel/rail wear protection with the longest possible carry distance is the ultimate goal of wayside lubrication.
- Serviceability, operability, maintainability, reliability and economic benefits are highly achievable with electric lubricators and long bar applicators.
- Long bar applicators can be implemented both in high MGT or low MGT lines with appropriate grease loading frequency and maintenance, but hydraulic short bar units are only applicable in very low MGT lines.
- Technological enhancement of the equipment and the resulting higher capability should be adopted by the rail operators because they can generate enormous savings and flexibility in resource allocation and motivation.

## 8.3 Future Research

Wheel/rail lubrication is a very critical issue. A variety of factors contribute in improving the mechanism of rail lubrication.  However, many factors still need to be researched in depth. The following studies could be the scope for future research.

- Extended research need to be conducted to evaluate the long term effectiveness of different methods of wayside rail curve lubrication and the impact on wheel/rail wear, noise and energy consumption. Wheel/rail wear trend need to monitor for extended period of time with appropriate wayside lubrication program in place.
- Various grease chemistry and their behaviour in boundary lubrication need to be studied to extend the knowledge and understanding of grease chemistry in rail curve lubrication.
- Study need to be conducted on formulation and specification of rail curve grease as per the necessity of rail lubrication.
- Economics of different methods of wayside lubrication also need to be studied over the long periods to establish long term goals for any rail operators. Though this research showed economic cost and benefits of various methods of wayside lubrication, real life outcomes need to be concluded from through application throughout the rail network.

- Wayside lubrication asset management and maintenance need to be studied to develop effective and appropriate asset strategy and necessary tactics to maintain highest reliability, maintainability, availability and effectiveness.
- Further study is needed to develop a model of grease transport mechanism in order to clearly understand the grease transport behaviour found in the practical measurement.

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