NT2008-37-24

METHODOLOGY FOR MEASUREMENT OF LUBRICANT DECAY RATE FOR RAIL LUBRICANTS

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ABSTRACT

A tribologist's dream is to predict the point at which a lubricant film will fail. The precursor to this ideal situation is to predict the decay behaviour of a lubricant film prior to failure. The performance decay of lubricants is of interest to the rail industry for two reasons; first, to predict reapplication rates, and second, to predict the lubricated distance from a lubricant application point. The work discussed in this paper investigated the failure of lubricant films in a simulated rail curve environment. Three rail curve lubricants were tested under traction-limited rolling sliding conditions.

New methods for measurement of rail curve lubricant performance were developed and one method, the half life of lubricant is discussed and results presented here. Lubricant half life in this work represents the reduction of sliding performance over time at a defined shear stress level or the time taken for a lubricant to lose half of its sliding performance.

Decay of lubricant performance was measured for three different rail curve lubricants under simulated conditions. The rail/wheel simulator used in this research consists of two dissimilar wheels (disks) rotating in contact with one another, simulating a conformal gauge corner contact. The first wheel, a simulated rail, is driven by an electric motor which then drives the second wheel, a simulated railroad wheel, through the contact. Hydraulic braking on the railroad wheel is used to simulate the rolling/sliding conditions.

The research found appreciable and quantifiable differences between lubricants. Industrial application of the findings will improve positioning of lubrication systems, improve choice of lubricants and predict effective lubrication distance from the lubricant application point.

1. INTRODUCTION

Railway systems use a wide variety of lubricants to combat the effects of wear in the flange contact. These lubricants are usually oil, grease or water. Railway systems often use a combination of lubricants. Some European rail systems use grease wayside lubricators for six months of the year and rely on snow (water) for the remaining months [1]. In Australia, grease wayside lubricators are most widely used, with on-board lubricators beginning to be used as well. It is still not clear as to what parameters make a ‘good’ lubricant.

Lubricant manufacturers specify the benefits of rail curve lubrication in their advertising material. They include:

- reduction of friction and wear;
- reduction or fuel/energy consumption
- reduction of noise
- reduction of maintenance of rolling stock and rail infrastructure

Recent studies by Hannafious [2] showed benefits of rail lubrication to be reduced fuel consumption, reduced wheel wear and reduced rail wear.

Research has focused on reduction of rolling friction and energy lost to friction[3]. Unfortunately the current research and that of Kumar *et al.*[3] has yet to provide any conclusive results as to which lubricant is the best.

In Australia and USA grease is widely used as oil is considered unsuitable [4]. Assuming that grease will be the optimum lubricant, from current usage patterns, parameters that improve performance need to be targeted. It is important to consider that lubricants are a commercial product and the research in their development is therefore not available for review. Performance measurements of rail curve lubricants require further research [1, 3, 5, 6]

Rail/wheel contact is an extremely complicated interface to simulate. Drawing comparisons between field and laboratory is difficult and direct comparisons have not been made from scaled simulation results [1, 3, 7]. The focus of wear performance may not be the most direct method of determining the optimum lubricant.

The review of laboratory lubricant testing systems is limited due to the paucity of recent publications. There are four groups [1, 3, 5, 6] that have published in the area of rail lubrication, the most current work being that of Waara [1]. The work of Waara in Sweden has focussed on the correlation between laboratory and field lubrication. The field testing of rail curve lubricants, which Waara started in 1997, has investigated the influence of mineral oil based greases, environmentally adapted greases and the influence of solid lubricant additives to these greases.

Mulvihill *et al.* [6] investigated rail/wheel lubrication with a twin disk machine. Results from their experiments indicated that the relationship between lubricating grease ingredients and performance was not clearly defined. Varying amounts of extreme pressure additives and solid lubricants had an unpredictable effect on the test outcome.

Clayton *et al.* [8] identified a need for a “simple inexpensive laboratory test method” for the performance characterisation of rail curve lubricants. Following his earlier research [8], Clayton [9] reviewed the tribological issues in rail wheel contact. In this review, Clayton [9] identified a need for a laboratory test device that can measure lubricant performance under a starved lubricant film. The work presented here is a method of predicting the decay or half life of the starved lubricant film to address this deficit in rail curve lubricant research.

From the four groups of researchers that have published work on rail/wheel lubrication in the last twenty years, the current research builds upon the foundations of their research, refines the method for testing lubricant properties, and poses more accurate methods that exploit the gaps identified in the body of rail/wheel lubrication research.

2. DESCRIPTION OF EQUIPMENT (RAIL/WHEEL WEAR MACHINE)

The rail/wheel simulator developed and used for this research originated from the BHP Melbourne Research Laboratories in Australia. This machine was purpose built by the laboratories to investigate wear of rail/wheel couples [10].

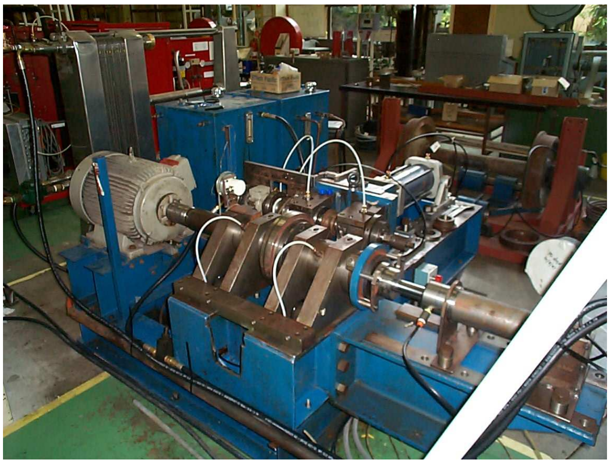

Figure 1 – Rail/wheel simulator

The equipment used to measure rail curve lubricant performance consists of two dissimilar disks, of matching contact profile, rotating in contact with one another, see Figure 1. The first disk, a simulated rail, is driven by an electric motor which then drives the second wheel, a simulated railroad wheel, through the contact. The simulated railroad wheel is hydraulically braked to simulate the traction under rolling conditions.

The variables of the simulated contact that are controlled with this equipment are contact stresses, input and output disk speeds, slip ratio between disks, disk geometries and material properties, and lubricant types including biodegradable products.

The experimental data for this paper was collected using the following method (for full experimental detail please see Wilson[11]).

- Position wheel sample holder to place rail and wheel samples in contact at specified tread load.
- Start input drive and tractive force system to gather unlubricated test results for the determination of the zero condition prior to lubricant application
- The system is then shut down and excess lubricant (~40g) is applied to the running surface of the rail sample.
- The system is then restarted without tractive force to generate a full width lubricant film. Once the set speed has been reached the tractive force is applied.
- Results are collected for a set time following development of full traction conditions.

The configuration of major components can be seen in Figure 2.

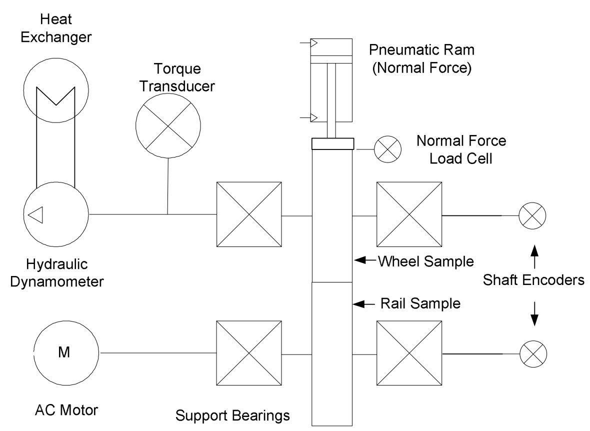

Figure 2- Schematic diagram of the rail/wheel simulator.

Rotational speed was measured with shaft encoders for the purpose of measuring rolling velocity, sliding velocity and slip ratio. Output torque was measured with a torque transducer in the hydraulic dynamometer system for calculating output power, shear force, shear stress and power absorbed by a lubricant film. The variable frequency drive on the input shaft was used to measure input torque, for calculating input power and power absorbed by a lubricant film. Temperatures of the rail and wheel samples were measured during testing using a hand held infra-red thermometer. Normal load, used in calculating the stress distribution of the contact between rail and wheel samples, was measured using a calibrated force transducer.

The results presented in this paper are for simulated conditions of a 27.5 tonne axle load travelling at 42km/hr into a 300m radius corner. The sampling of the rolling speeds through the encoders was taken at a 1Hz rate. Three commercially available rail curve lubricants were tested with three tests for each lubricant.

3. MATHEMATICAL METHOD (SLIP CALCULATION)

The half life of the lubricant is calculated from the slip ratio versus time data recorded experimentally. The two components of slip ratio and time are used to select the time period over which the half life is calculated. The time period of interest starts when the rolling conditions have reached full tractive rolling and the slip has reduced to below 5%.

The measurement of slip ratio is an approximation which takes into account a number of factors, which are discussed here. The rolling diameters of the wheel and rail were taken using contact measuring devices which have a level of precision below that required for a high precision calculation of slip/creep (> 0.01% slip). The method used in this paper does not account for the worn rings and surface texture which is necessary for measuring the 'real' diameters. Another source of error, thermal expansion, is a factor which is difficult to account for as the thermal profile and heat transfer system is highly variable. Thicknesses of the remaining lubricants influenced the value of rolling diameters and therefore the slip ratio as well.

The slip ratio, ξ, was calculated using Equation (1).

$$\xi = \left(1 - \frac{w_o r_o}{w_i r_i}\right) \qquad (1)$$

w_i = Angular velocity of input shaft – rads/sec

w_o = Angular velocity of output shaft – rads/sec

r_i = Rolling radius of input shaft - m

r_o = Rolling radius of output shaft - m

Variable	**Value**	**Variable Error**	**Value**
w_i	39.813 rad/s	Δw_i	0.001 rad/s

w_o	120.851 rad/s	Δw_o	0.001 rad/s
r_i	148.10 mm	Δr_i	0.005 mm
r_o	48.605 mm	Δr_o	0.005 mm

Table 1 - Values for experimental error calculation of slip ratio

. Subscripts refer to input and output shafts.

Using Equation (1) and the values of variables in Table 1 the experimental error in slip ratio is predicted to be 1.11E-4 or presented as slip percentage 0.011%. This value is small compared to the experimentally recorded slip ratio, and gives confidence to the prediction of slip and to the measurement of sliding speeds and distances and prediction of half life.

At the conclusion of all testing in this research a lubricant film was present and as such the subsequent wear rate was assumed to be negligible. The values of radius for this research were therefore assumed to be constant during each test.

Slip ratio is composed of the micro-slip component calculated in this section and the slip component due to lubrication. The micro-slip was calculated using the work of Johnson [12]. Over the range of shearing force the laboratory simulator is capable of producing, the maximum value of micro-slip is 0.06%.

4. HALF-LIFE PREDICTION METHOD

The performance decay of lubricants is of interest for two purposes, first to predict reapplication rates, and second, to predict the lubricated distance from a lubricant application point. The decay was calculated from the slip measurement, following the system reaching the set shear stress value (tractive force).

The time was normalised using the mean and standard deviation of the time data to improve the accuracy of regression analysis results using Equation (2).

$$\hat{x} = \frac{(x - \tilde{x})}{\sigma}$$

$\hat{x} =$ normalised values

$x =$ values (2)

$\tilde{x} =$ mean of x

$\sigma =$ standard deviation of x

Regression was carried out on the slip data using Equation (3).

$$f(x) = ae^{-bx} + c$$
$f(x) =$ Variable of interest, slip ratio in this case
$x =$ Time
$a =$ Amplitude of the exponential
$b =$ Exponential coefficient
$c =$ Minimum value of variable of interest (3)

The time for slip ratio (lubricant performance) to degrade by 50% or half life was then calculated with Equation (4) [13].

$$\lambda = \frac{\sigma}{b}\log_e(2) \quad (4)$$
$\lambda =$ half life

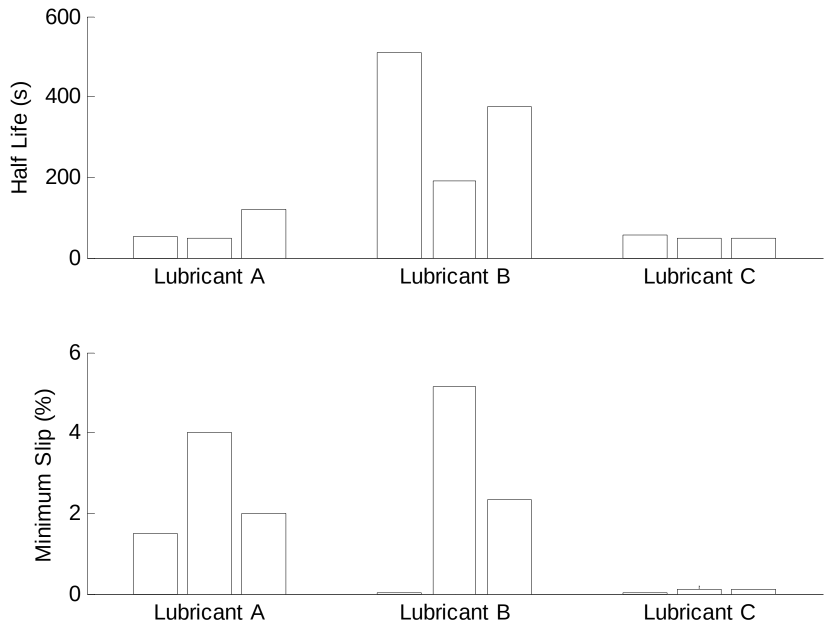

Figure 3 – (top) Half life prediction for experimental example using $f(x) = ae^{-bx} + c$. (bottom) Value of predicted minimum slip ‘c’.

Predicting the half life was highly dependent on the coefficient ‘c’ in the exponential curve fit, which has the expected value of zero (if the input variables are accurate) but the regression analysis did not support this expectation, see Figure 3 (bottom).

Inspecting the example in Figure 4, the equation with the displacement coefficient ‘c’ has a better fit. This cannot exist in practice, as the lubricant film will fully degrade and zero slip conditions will be reached. While mathematically this equation is a better fit

(R^2=0.9545 versus R^2=0.9814), the equation without ‘c’ was used to reflect the expected outcome.

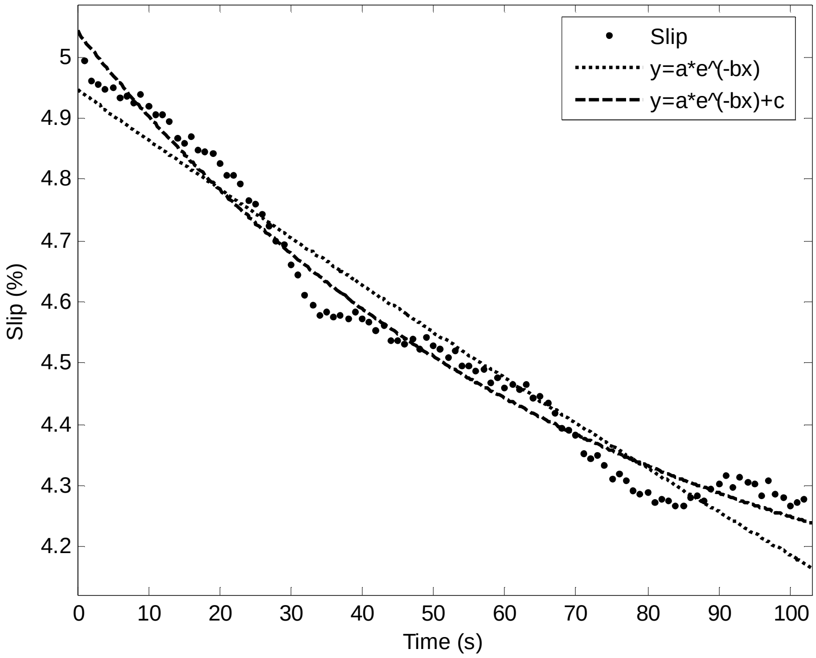

Figure 4 – Regression plots for Lubricant A Test 2 in the region < 5% slip.

	HALF LIFE (S)	
Lubricant Type	Mean	Standard Deviation
A	274.51	144.37
B	983.9	592.77
C	87.75	8.5956

Table 2 – Half life values for each lubricant in Group 1 testing using $f(x) = ae^{-bx}$.

Lubricant C had a small but predictable half life, seen by the small standard deviation, which may be the result of testing into the region of slip below 1%. The other lubricant tests ceased prior to the reduction in slip reached by Lubricant C. Lubricant B clearly had the longest half life but predictability of total film failure would be problematic with the large standard deviation. Next, in terms of performance, Lubricant A had large variability and longer life than Lubricant C.

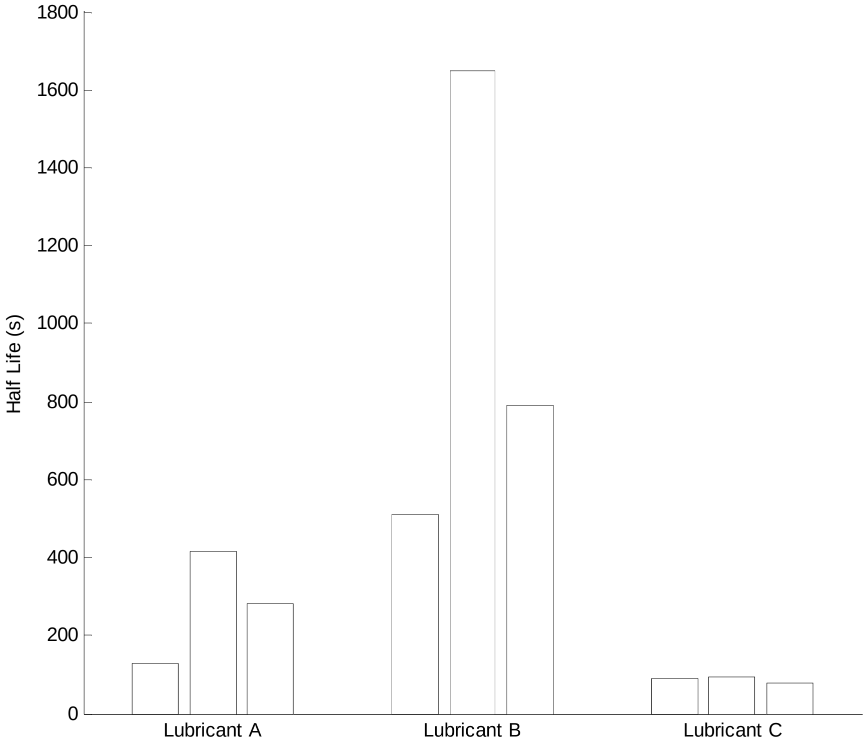

Figure 5 - Half life values for each lubricant in Group 1 testing using $f(x) = ae^{-bx}$.

5. DISCUSSION/CONCLUSION

Results of the half life performance of the rail curve lubricants may not provide an accurate measurement of the wear performance, but assuming that the presence of lubricant reduces wear, the best performance would arise from the longest half-life.

Standard lubricant tests, such as those from ASTM, provide inadequate information for rolling stock and rail infrastructure managers to make informed decisions as to which lubricant to use. The standards-based testing present results which may not be relevant to rail curve lubrication, whereas the rail/wheel simulator gives results for performance criteria that may be more relevant to rail curve lubrication due to better simulation. Performing a group of tests such as those presented in this paper can highlight advantages and deficiencies in a range of contact conditions that standards-based testing cannot achieve.

The predictions of half life are sensitive to the errors in the final value of slip. These errors are dependent upon the temperature of test pieces, which is dependent upon the number of samples taken after the set traction force is reached. In the calculations of half life the offset coefficient is representative of the thermal expansion error. Therefore to reduce the offset coefficient, sufficient samples after the frictional power has reduced below the convective power losses must be allowed for the test piece temperatures to stabilise as near to unlubricated conditions as possible.

Observing that the decay in measured slip is the result of two processes, decay of lubricant film and decay of sample temperature, the prediction of half life could be improved by modelling each of these processes. Taking each of these components as having an exponential decay gives Equation 7.

$$\xi = ae^{-bt} + ce^{-dt}$$

$\xi =$ measured slip ratio (7)

$a, b, c, d =$ regression coefficients

Using this model to perform a regression analysis of the slip gives a higher correlation coefficient than the single exponential problem. A difficulty with using this model is that there is no method to differentiate between the effects of thermal expansion and lubricant film decay. The error analysis for thermal expansion of the test pieces shows that the component of slip from thermal expansion becomes small, rapidly leaving only the lubricant film decay component.

Relating the half life predictions to the field is somewhat difficult. The simulator test failure criteria is the reaching of a set tractive force, whereas the field lubricant film failure criteria is that there is no lubricant remaining on the rail. Considering the magnitude of wear in each case, for the simulator the wear is negligible as the lubricant film still exists, for the field, wear is considerable as the protective film has been totally removed. Simulator test conditions therefore are not representative of the field situation in this aspect but do represent the desired level of lubrication from an industry view point.

The slip calculation/measurement taken by the simulator is not affected by film thickness, which allows for the estimation of film thickness, as the magnitude of the value of film thickness will always reach zero despite calibration or measurement errors. Film thickness is not required to be specified, but sliding distance is required. For a lubricant manufacturer; the higher the film thickness, the smaller the shear rate.

Following correlation with field results, the half life performance criterion will allow for improved lubricant design and better placement of lubricators and the associated benefits of improving the lubrication system. The overall impact of this methodology will be seen following future validation with field analysis. In the interim however this methodology can be used to screen lubricants for their anti-friction capacity, which is valuable for minimising operating costs of a rail network.

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