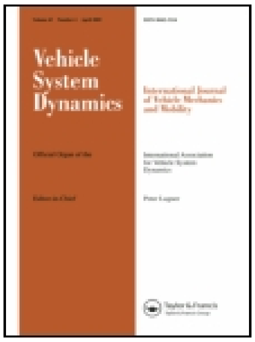

Vehicle System Dynamics

International Journal of Vehicle Mechanics and Mobility

ISSN: 0042-3114 (Print) 1744-5159 (Online) Journal homepage: https://www.tandfonline.com/loi/nvsd20

Continuous observation of wheel/rail contact forces in curved track and theoretical considerations

Akira Matsumoto , Yasuhiro Sato , Hiroyuki Ohno , Makoto Shimizu , Jun Kurihara , Masao Tomeoka , Takuya Saitou , Yohei Michitsuji , Masuhisa Tanimoto , Yoshi Sato & Masaaki Mizuno

Vehicle System Dynamics
Vol. 50, Supplement, 2012, 349–364

Taylor & Francis
Taylor & Francis Group

Continuous observation of wheel/rail contact forces in curved track and theoretical considerations

Akira Matsumoto[a]*, Yasuhiro Sato[a], Hiroyuki Ohno[a], Makoto Shimizu[b], Jun Kurihara[b], Masao Tomeoka[b], Takuya Saitou[b], Yohei Michitsuji[c], Masuhisa Tanimoto[d], Yoshi Sato[e] and Masaaki Mizuno[e]

[a]*National Traffic Safety and Environment Laboratory (NTSEL), Chofu, Tokyo 182 0012, Japan;* [b]*Tokyo Metro Co., Ltd, Taito-ku, Tokyo 110-8614, Japan;* [c]*Faculty of Engineering, Ibaraki University, Hitachi 316-8511, Japan;* [d]*Sumitomo Metal Technology Inc., Osaka 554-0024, Japan;* [e]*Sumitomo Metal Industries Ltd, Osaka 554-0024, Japan*

(*Received 22 October 2011; final version received 18 February 2012*)

By using non-contact gap sensors equipped on non-rotating parts of a bogie, a new measuring method of wheel/rail contact forces has been developed. The developed system has been verified to have sufficient durability for continuous measurement on in-service trains and sufficient practical accuracy after various stand tests and train running tests. After a long-period of continuous measurement on a commercial subway line, some important characteristics of wheel/rail contact mechanics were found by the analysis of measured data. Numerical simulations of curving with a full vehicle model using multi-body dynamics software were carried out, and according to the comparison with measured data, simulation results agree well with measured data in the steady-state values of derailment coefficients considering friction coefficient.

Keywords: wheel–rail interaction; derailment coefficient; friction coefficient; condition monitoring; vehicle dynamics simulation; measurement

1. Necessity and problems of measuring system of wheel/rail contact forces

1.1. *Necessity of continuous measuring*

Running safety of railway vehicles against derailment can be estimated mainly by using 'the ratio of lateral and vertical wheel/rail contact forces on the outside rail, i.e. high rail'. This value is called 'derailment coefficient', and is defined by the following equation:

$$\text{'Derailment coefficient'} = \frac{\text{'Lateral contact force (L)on outside rail'}}{\text{'Verticalcontact force (V)on outside rail'}}.$$

This value is denoted 'Y/Q' in Western countries, 'Q/P' in Japan, but 'L/V' is used in this paper for easier understanding.

*Corresponding author. Email: matsumoa@nifty.com

ISSN 0042-3114 print/ISSN 1744-5159 online

http://dx.doi.org/10.1080/00423114.2012.669130
http://www.tandfonline.com

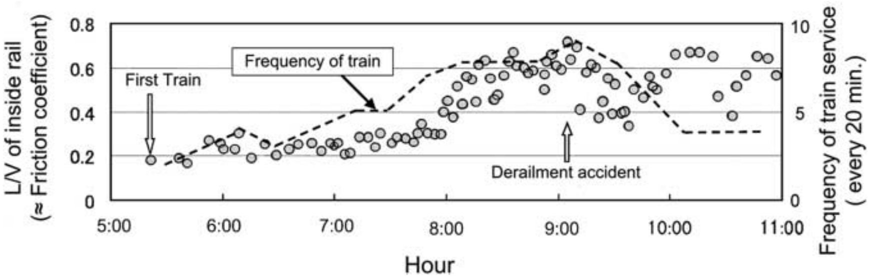

Figure 1. Change of friction coefficient of wheel/rail interface just after 'Nakameguro Accident'. Measured L/V on inside rail, which is nearly equal to friction coefficient on sharp curve. The friction coefficient increases with a corresponding increase in the frequency of train service, and reaches a maximum at 'rush hour'.

Derailment coefficient varies according to wheel/rail contact conditions, such as lubrication, time interval of train operation, rail temperature and climate. For example, Figure 1 shows the change of L/V on the inside rail, i.e. low rail, as measured on the track, just after the Tokyo subway derailment accident in 2000, which occurred on a sharp curve of 160 m radius [1]. This figure shows a remarkable change of L/V on the inside rail after the first train running of the day, L/V is also increasing according to the frequency of train service and reaches the highest value at 'rush hour', when the accident occurred. The value of L/V on the inside rail is considered to be nearly equal to the friction coefficient μ of the wheel/rail interface on a sharp curve, such as this point. As the derailment coefficient is considered to increase due to the increase of μ, consequently the derailment coefficient also changes over time. These mechanisms are discussed later in Section 2.2 (Figure 12).

1.2. *Problems in conventional measuring methods*

In spite of the circumstances mentioned above, derailment coefficients have been measured only in special measurements before the opening of newly built lines, or the operation of newly designed cars. In such special measurements, the conventional strain-gauge-equipped wheelsets are used as measurement systems, but have some problems as mentioned in (1)–(3) and cannot measure the fluctuation of derailment coefficients by the change of wheel/rail contact condition.

(1) The lifespan of the measurement system is short because of the wear of contact parts in slip-rings for data transmission.
(2) The strength of axles is insufficient for long-term use, because the axle of the wheelset is drilled through in order to make passages for signal cables from strain gauges.
(3) In the case of using telemeters, the usage is rather complicated and durability is insufficient.

Because of the above-mentioned reasons, frequent measuring or monitoring on in-service trains has been impossible, although wheel/rail contact forces change minute by minute.

Figure 2(a) shows the arrangement of strain gauges on a Japanese conventional special instrumented wheelset, so-called 'PQ wheelset', and (b) shows its outlook. Although various new measuring systems have been developed in Western countries and Japan [2–4], every system needs strain gauges and has not solved the problems mentioned above.

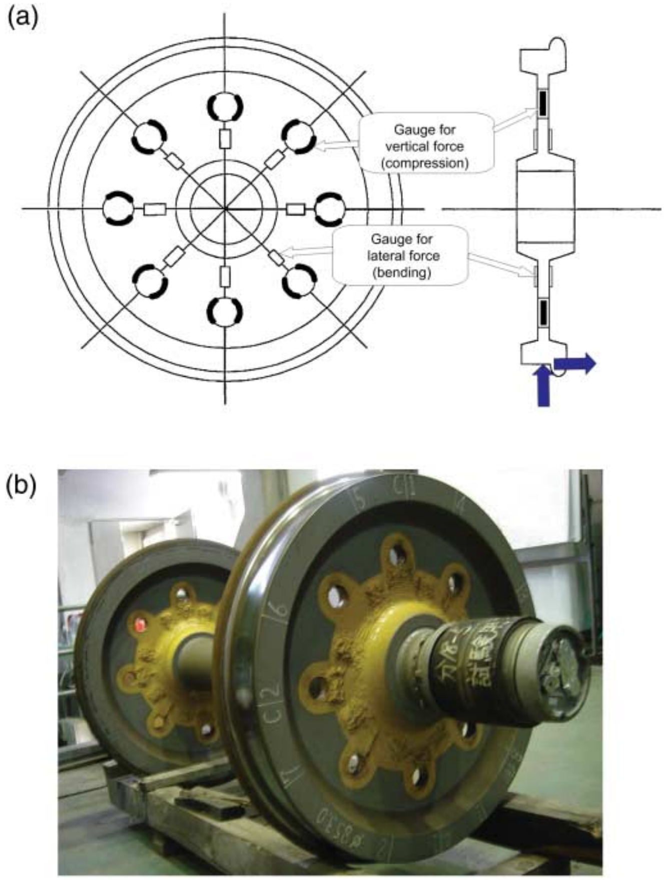

Figure 2. Conventional Japanese method for measurement of wheel/rail contact forces. (a) Arrangement of strain gauges: 32 gauges are necessary for vertical and lateral force measurement, and a slip-ring or a telemeter is also necessary for data transmission from a rotating wheelset. (b) Conventional instrumented wheelset for measurement (called 'PQ wheel' in Japan).

2. Development of new measuring system of wheel/rail contact forces

2.1. *Outline of newly developed system*

The authors had researched and developed the new measuring method, which can measure contact forces and derailment coefficients every day on all curves of commercial lines. This new method can obtain derailment coefficients statistically as a function of friction coefficients, train running speed and so on.

In the new measuring system, no measurement apparatus, such as strain gauges and slip-rings, are equipped on the rotating parts of the bogie. Vertical contact forces are measured by the deflection of the axle spring, and the lateral forces are measured directly by the bending deflection of the wheel, using non-contact gap sensors attached to the bogie frame [5].

2.2. *Measuring method of lateral contact force*

In the new method, the bending deflection of the wheel is detected by sensors attached on the parts of the bogie frame that are not rotating. As shown in Figure 3(a), the wheel deflection is

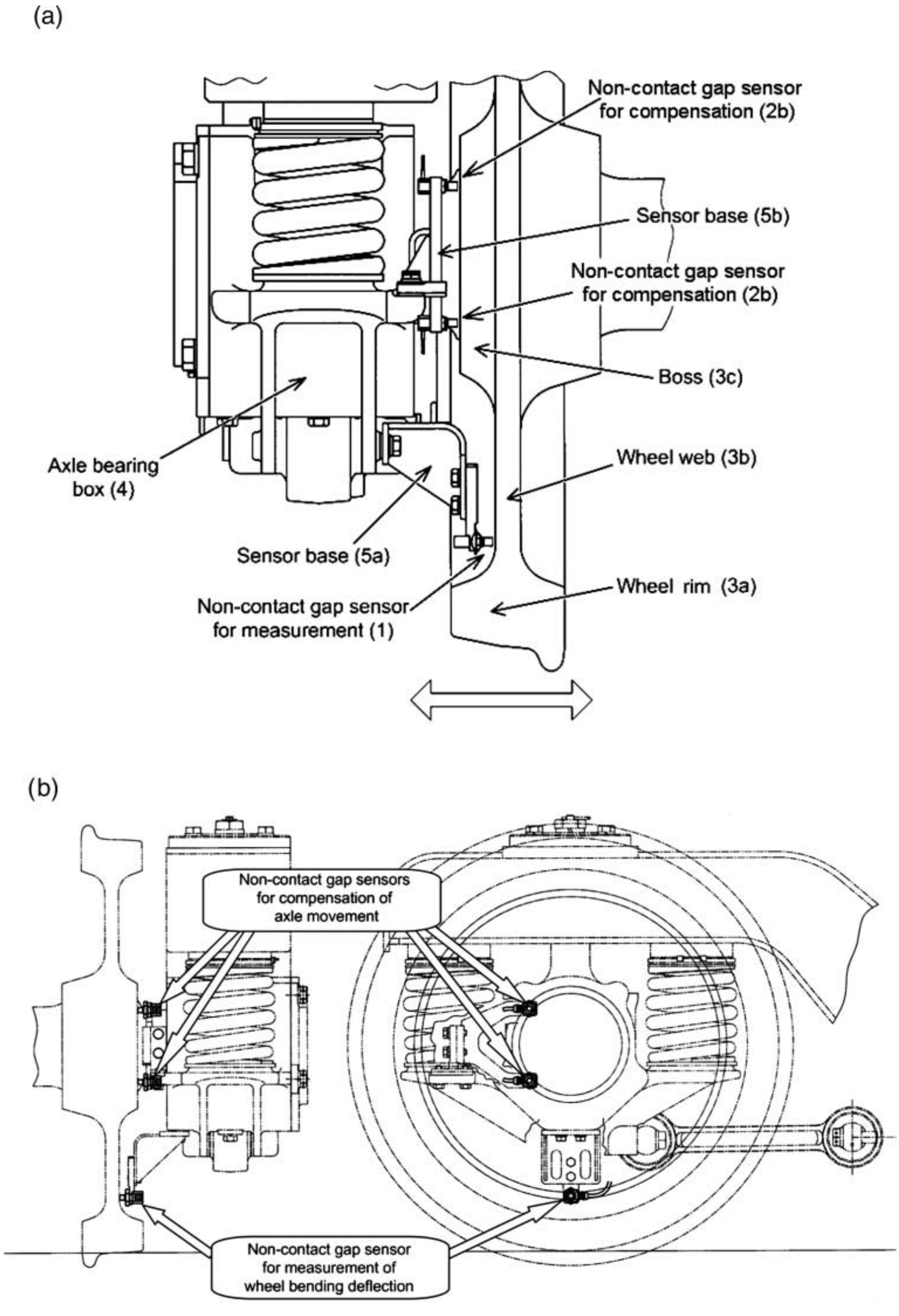

Figure 3. Arrangement of sensors for lateral contact force measurement. (a) Detailed layout of sensors for lateral force measurement. (b) Layout of non-contact gap sensors for lateral contact force measurement on newly developed measuring bogie.

(a)

(b)

Figure 4. Sensors for measurement. (a) Non-contact gap sensor for measuring bending deflection of wheel by lateral contact force. (b) Magnetostrictive displacement sensor for deflection of primary suspension by vertical contact force, i.e. 'wheel load'.

calculated from the displacement of the wheel web (3b), which is measured by the non-contact gap sensor (1) attached on the axle bearing box (4) through the sensor base (5a).

Because an accuracy of less than 0.01 mm is required, inductive displacement sensors are chosen. In order to obtain the highest accuracy, the gap sensors should be installed at wheel 'rim' position (3a), but this position is slightly out of the 'vehicle gauge.' Another position inside 'vehicle gauge', i.e. the outer edge of wheel 'web' (3b), was thus chosen.

As the values measured by gap sensors are small, the movement of the wheelset relative to the axle box cannot be neglected. For compensation of the wheelset movement, two gap sensors (2a, 2b) are attached to the bearing box. The axial movement produced by the thrust clearance of bearings and the inclination produced by the relative inclination of the wheelset against axle bearing box are compensated by the values from the two gap sensors [5].

The layout of non-contact gap sensors is shown in Figure 3(b), and a photo of the main sensor (1) is shown in Figure 4(a).

2.3. *Measuring method of vertical contact force*

As the vertical contact forces, i.e. wheel loads, are important for railway vehicle dynamics, constant monitoring is also necessary. In order to avoid using slip-rings and telemeters,

(a)

(b)

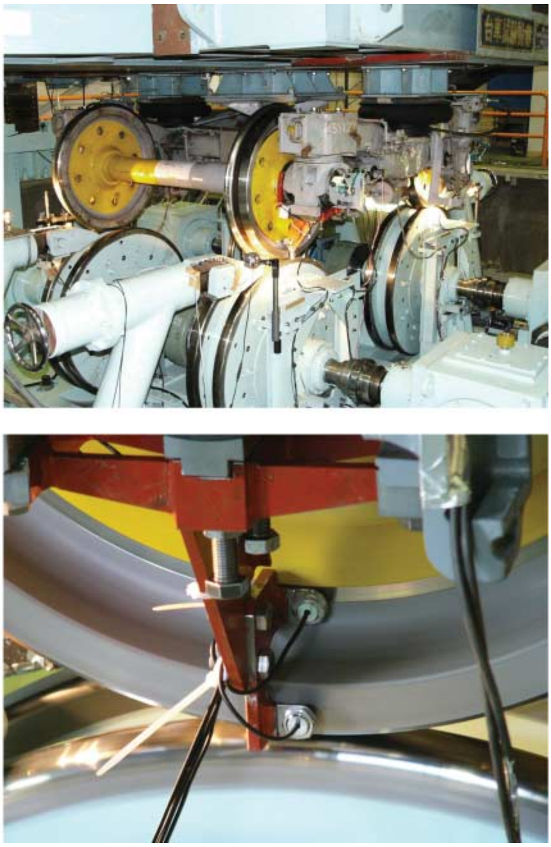

Figure 5. Curving tests on test facility in NTSEL for verification and improvement of the new measuring system. (a) Running test scene on test stand. (b) Two measuring positions: wheel rim and web.

measurements from non-rotating parts of the bogie are chosen. The vertical contact forces are measured by the deflection of primary suspension using magnetostrictive displacement sensors.

The set-up of the sensor is shown in Figure 4(b).

2.4. *Improvement for practical system – stand tests and train running tests*

Based on the concept of the new measuring system, a test bogie was designed and manufactured. To verify the measuring method and to design the practical system for use on in-service trains, running tests were carried out in a test facility in the National Traffic Safety and Environment Laboratory (NTSEL). Here, curving conditions, such as curving radius, difference of running path of outside/inside rails and deficiency of superelevation of track, could be evaluated [6,7]. In these tests, the conventional instrumented wheelset was used to compare the new method with the conventional method. Figure 5(a) shows the test set-up in NTSEL and Figure 5(b) shows the test concerning the measuring position of wheel bending for lateral force.

Train running tests on a commercial line were also carried out. Based on these test results, the following improvements were carried out for the practical system.

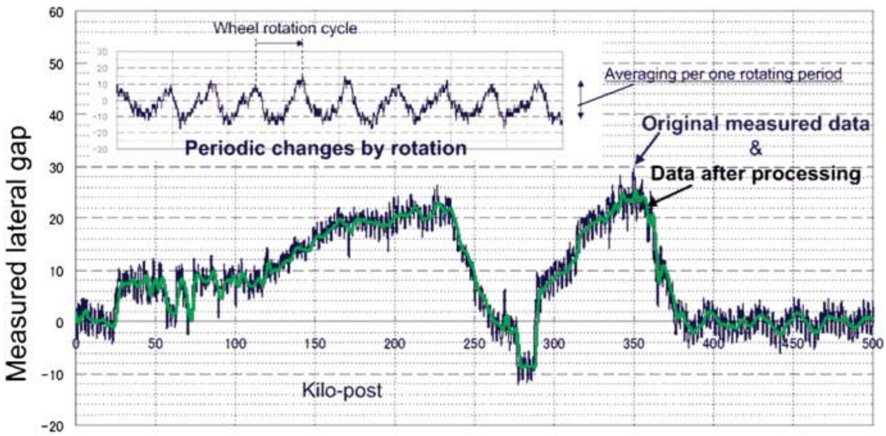

Figure 6. Reduction in fluctuation of wheel lateral movement by data processing.

2.4.1. *Improvement of bogie structure*

In order to decrease the movement of wheelset, axial bearings with a small thrust clearance were used, and these bearings were fixed to axle box by keys.

2.4.2. *Fine machining for prevention of wheel lateral fluctuation*

In order to reduce the wheel lateral fluctuation during rotation, fine machining of the side surfaces of wheels had to be carried out. After the fine machining, the lateral fluctuations of the wheels were reduced to less than 0.02 mm.

2.4.3. *Reduction of fluctuation of wheel lateral movement by data processing*

In spite of the countermeasures on bogie structure and fine machining mentioned above, some fluctuations remained as shown in Figure 6. Most of these residual fluctuations are caused by the rotation of the wheel, and are rather similar in each revolution. Thus, these fluctuations can be removed by data processing, such as 'averaging per one rotating period'. It is the simplest method, and this method is adopted in this system for the present. If higher frequency is required in measurements, compensation methods considering rotation phase can achieve such a demand.

2.5. *Newly developed measuring bogie for practical use*

Besides the improvements mentioned above, the following improvements have also been implemented:

(1) Compensation against the difference of wheel load acting position and sensor position.
(2) Compensation against load transfer by bogie pitching for braking reaction.

These improvements removed various factors which reduced accuracy and durability. Figure 7 shows the newly designed bogie.

Figure 7. Total view of the newly developed measuring bogie for a service train.

2.6. *Verification of newly developed system – comparison of newly developed method and conventional method*

Before practical use on in-service trains, the new measuring system was tested on a train on the Marunouchi line, with a length of about 30 km, to compare it with the conventional method and to verify its durability. The signals of lateral force sensors are sampled every 8 ms and processed by the moving average method after temperature compensation. The signals of wheel load sensors are also sampled every 8 ms. The zeroing of lateral force is adjusted by the lateral force value of the trailing axle running on a particular straight track.

Figure 8 shows the results of running test data on the commercial line. In the figure, 'vertical forces' and 'lateral forces' measured by the conventional method and the newly developed method are shown. The values from the new measuring system agree with the conventional method data, with a ± 2 kN difference in the lateral force, and with a ± 0.1 kN difference

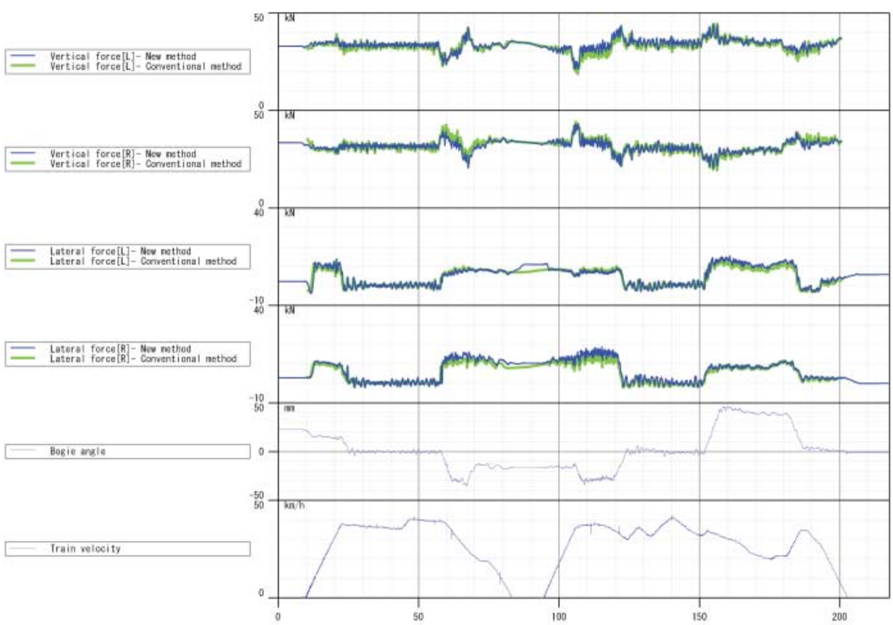

Figure 8. Comparison of measured data between new method and conventional method – train running test on a commercial line.

for derailment coefficients. Consequently, it was verified that this new method has sufficient practical measurement accuracy.

3. Continuous observation on in-service trains

3.1. *Examples of measurement on curved tracks*

Trial measurements using the new measuring system have been carried out continuously by using an in-service train-set all along a commercial subway line of Marunouchi [8,9]. Among the several measured data, two samples on sharp curves by the new system are shown in Figure 9(a) and (b). These figures show positional changes of the derailment coefficient ('lateral contact force L'/'vertical contact force V') of the front-outside wheel of the leading bogie from the beginning of the transition curve to the end of the transition curve of two 160 m-radius curves. Each line in a figure indicates the data of each passing train. Texts on each figure indicate the date and time of measuring train passage (25 trains passed at different times of the day).

Figure 9 indicates that derailment coefficient values change as a function of 'time', etc. but the shapes of wavy lines produced by each train are similar every time, and the maximum point of derailment coefficients appears at the same kilometre post (at the exit-side of transient curve) every time. It is thus possible to extract the maximum points and effectively set up 'anti-derailment guards' in the curves. If the wavy lines deviated from their regular patterns, it would be possible to detect track condition failure.

The difference in the level of each wavy line is mainly due to the difference in the friction coefficient between wheel/rail in each pass. This is caused by the changes in train service frequency, rail temperature and so on. The difference in L/V along the curved track is considered to be due to the difference in friction coefficient, track irregularity, vehicle movement and so on. The two sharp curves used for the analysis have the same specification, such as 160 m radius, superelevation and gauge widening. The data are thus similar, but different in detail. The difference is due to the difference of friction conditions of wheel/rail interface, the difference of vehicle movement produced by different boundary conditions of each curve, etc.

3.2. *Increase in derailment coefficient and consideration on influential factors*

Figure 10(a) shows the relationship between derailment coefficients and curving radius along the whole line. The coefficients increase with a corresponding increase in curvature as well as the friction of the inside wheel/rail contact surface, which is influenced by wayside lubricant application.

Figure 10(b) shows the relationship only on well-maintained curves. The derailment coefficients are lower compared with those of whole curves, and in more than 200 m curves derailment coefficients stay below the safety limit calculated by Nadal's equation. Tracks in curves should thus be maintained to keep the derailment coefficient under the safety limit. In curves of 200 m-radius or less, 'anti-derailment guards' should be used for safety. Thus, the track maintenance based on the continuous monitoring by the new measuring system can keep tracks under the safety limit all ways.

Figure 11 shows the relationship between derailment coefficients, i.e. L/V of the outside wheel and L/V of the inside wheel of the leading axle. The value of 'inside L/V' is considered nearly equal to the friction coefficient μ on sharp curves, because L/V is saturated to the Coulomb friction coefficient due to large creepage. As shown in the figure, derailment coefficients increase proportionally with the increase in the ratio of 'inside L/V,' i.e. μ. The

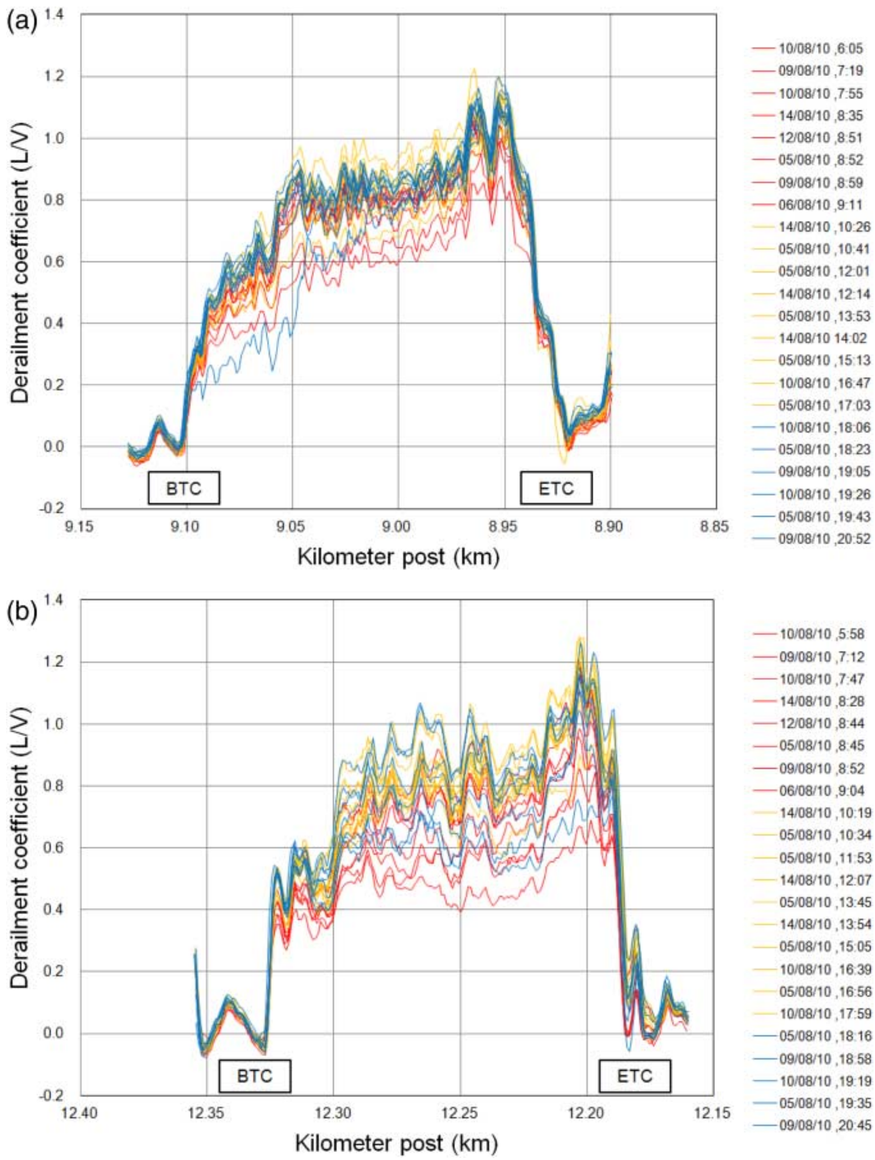

Figure 9. Measured data: (a) in sharp curve A ($R = 160$ m, superelevation $= 125$ mm, gauge widening $= 13$ mm) and (b) in sharp curve B ($R = 160$ m, superelevation $= 125$ mm, gauge widening $= 13$ mm). Variation of derailment coefficients (outside L/V) with positional changes and multiple train passes.

increase in derailment coefficients is due to the increase in anti-steering moment produced by high friction inside the wheel/rail interface where the rolling radius difference is insufficient between inside/outside rails.

This mechanism is illustrated in Figure 12.

Figure 12 shows the typical forces for a curving bogie in a sharp curve. The increase in friction coefficient μ causes an increase in the lateral creep force in the leading wheelset, which is produced by a large attack angle. At the same time, the longitudinal creep force in the trailing wheelset increases due to an insufficient rolling radius difference. The increase in these forces results in an increase in the anti-steering moment of the bogie. Consequently,

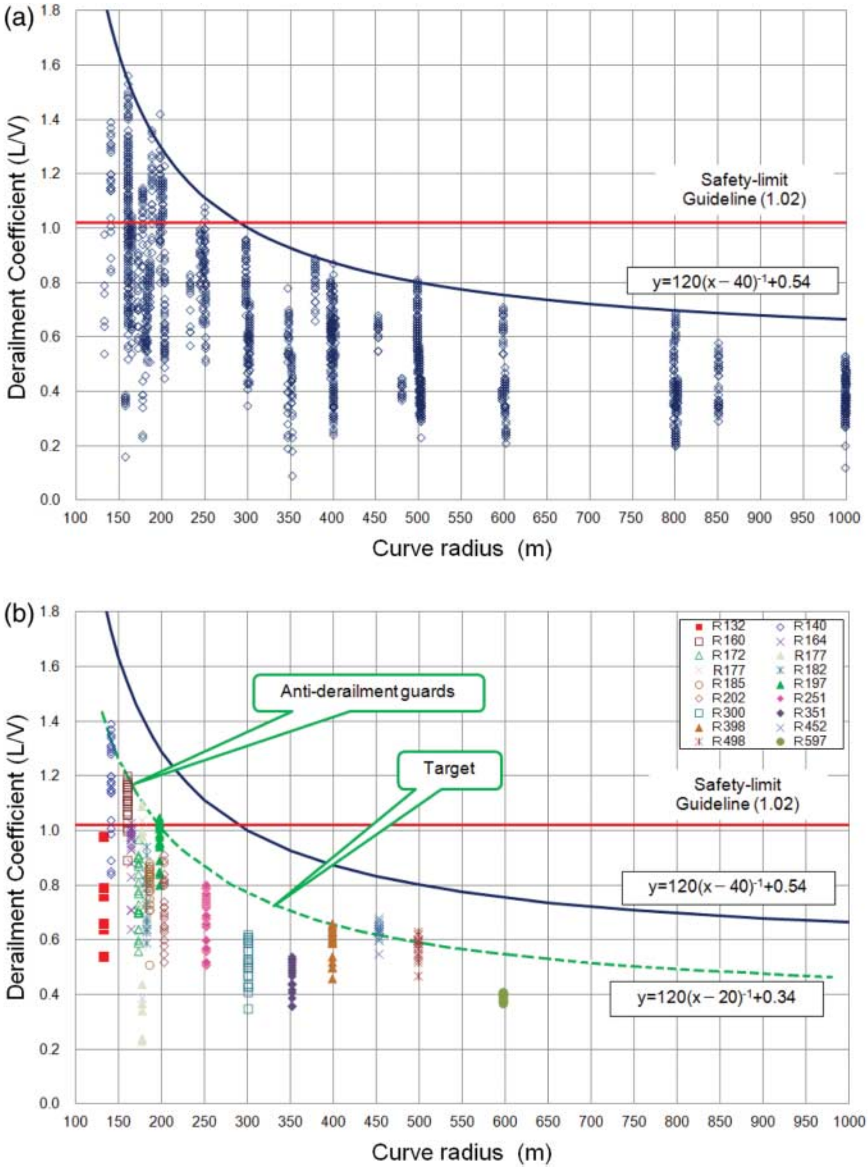

Figure 10. Relationship between derailment coefficients and curving radius along the whole line. (a) On all curves. (b) On well-maintained curves.

the flange force of the leading wheelset increases. By this mechanism, the increase in friction coefficient μ directly influences the increase in derailment coefficient. Conversely, derailment coefficients can be controlled by the control of friction coefficient μ. The newly developed measuring system can also measure friction coefficient μ from L/V of inside rail, so the system can control derailment coefficients. Off course the control of μ is also useful for the countermeasure of rail corrugation, etc.

Figure 13 shows measured derailment coefficients on three successive sharp curves following a rail lubricator. The average values of inside L/V, i.e. μ, increase with distance from the rail lubricator, and the derailment coefficients are the highest in the last curve.

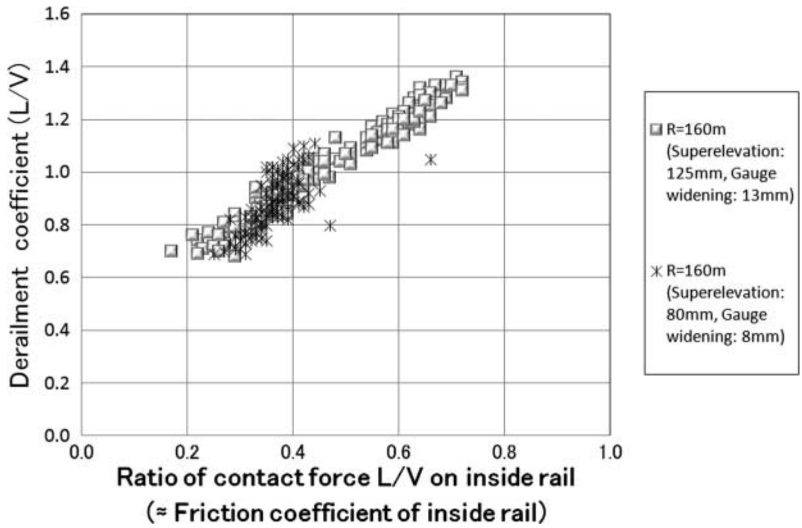

Figure 11. Relationship between derailment coefficients of outside rail and friction coefficient of inside rail on sharp curves ($R = 160$ m).

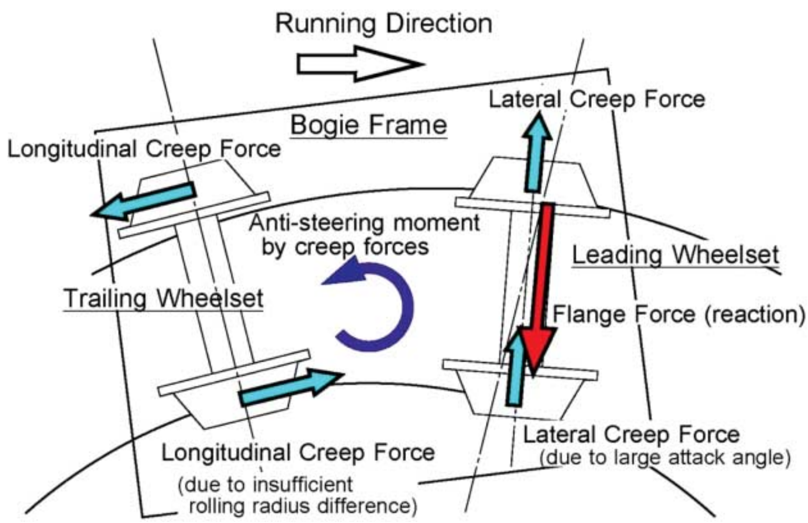

Figure 12. Typical forces acting in curving bogie on sharp curve. According to previous measurements, in general, longitudinal creep forces in the leading axle are small compared with lateral creep forces, and lateral creep forces in trailing axle are small compared with longitudinal creep forces, therefore these forces are omitted in the figure.

Figure 14 shows the relationship between derailment coefficients and wheel loads, i.e. vertical contact forces of the outside wheel of the leading wheelset. The coefficients decrease moderately corresponding to the increase in wheel loads. The decrease in derailment coefficients by higher wheel loads is due to the decrease in anti-steering moment because of lower vertical contact forces of inside wheels in such a case.

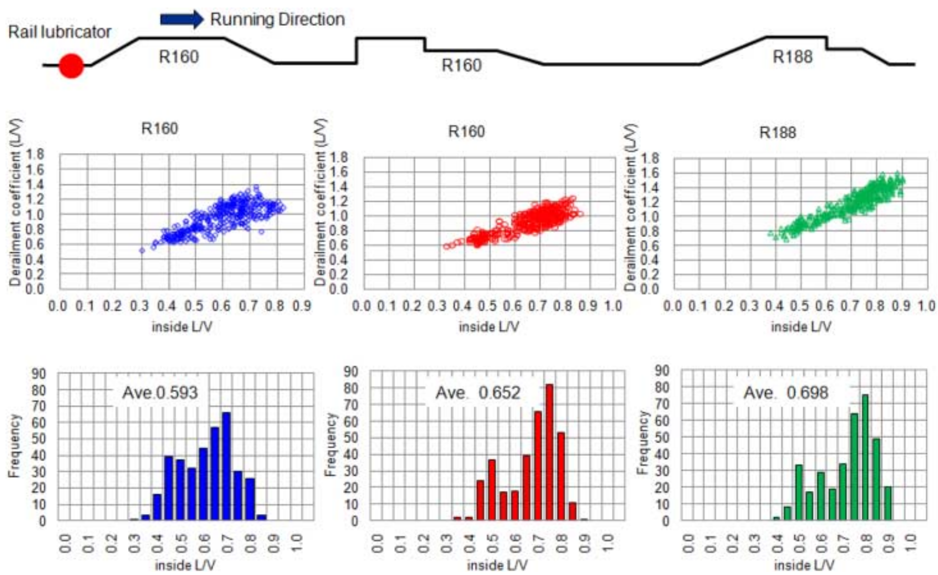

Figure 13. Derailment coefficients on three successive curves after passing through rail lubricator.

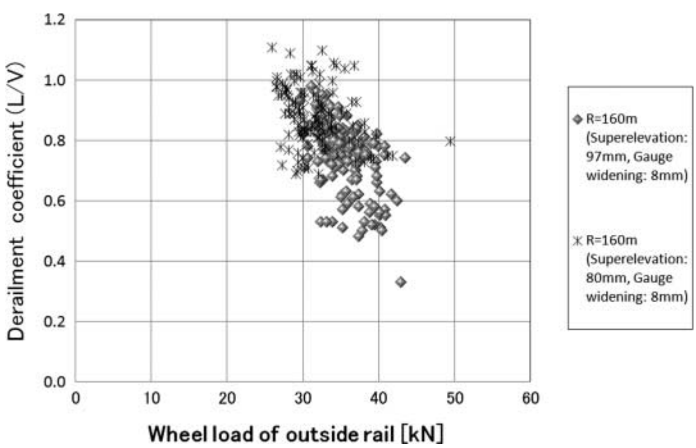

Figure 14. Relationship between derailment coefficients and wheel loads of outside rail on sharp curves ($R = 160$ m).

4. Comparison between simulation and actually operated line

4.1. *Multi-body dynamics simulation platform*

In order to fully comprehend the effect of lubrication which greatly affects the value of L/V, numerical simulation is also conducted. Multi-body dynamics (MBD) software based on automatic generation of equations of motion (A'GEM) is used for numerical analysis [10]. In A'GEM, vehicle data, the geometric data of the wheel and rail profile, and running conditions are considered as initial parameters. The FASTSIM algorithm is used for wheel–rail contact force calculation. In the numerical simulation, the value of μ for each wheel is changed in order to experimentally compare the obtained data. The other primary parameters of the vehicle are given in Table 1.

Table 1. Vehicle primary parameters.

Vehicle body, bogie frame and wheelset mass	20,820, 840, 914 (kg)
Radius of inertia (body, bogie frame and wheelset)	3.08, 0.637, 0.686 (m)
Suspension stiffness of axle box (longitudinal, lateral and vertical)	7355, 4903, 1761 (kN/m)
Air spring stiffness (lateral and vertical)	89.2, 152.0 (kN/m)
Damping coefficient of secondary suspension (lateral)	98.0 (kN/m·s)
Length between bogie centres	12.0 (m)
Length between wheelsets	1.90 (m)
Radius of wheel	0.43 (m)
Wheel back gauge	1.348 (m)
Rail gauge	1.435 (m)
Tread	Non-linear profiled tread

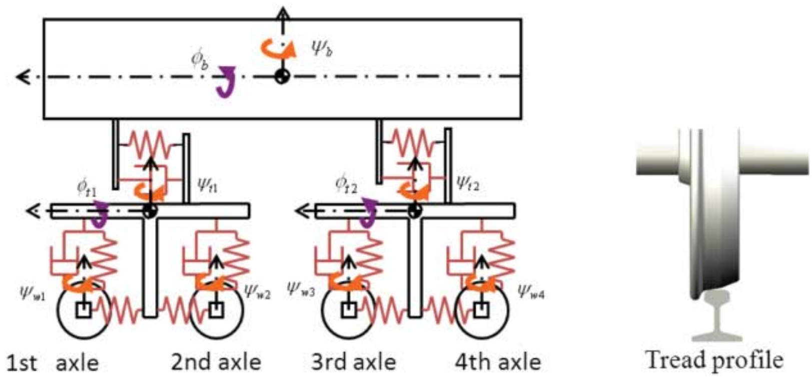

Figure 15. 17 DOF vehicle model for numerical simulation.

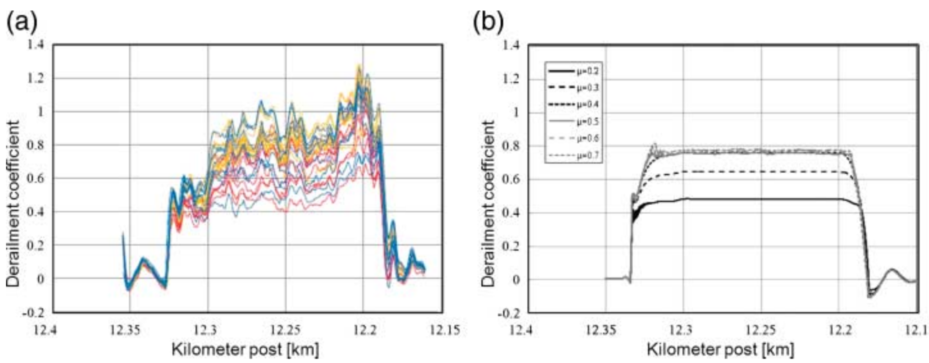

Figure 16. Comparison of derailment coefficients between simulation and real track. (a) measured data. (b) simulation results.

4.2. *Comparison between simulation result and actual data in service operation*

MBD simulation for a typical full vehicle model with 17 degrees of freedom (DOF), as shown in Figure 15, is performed. Figure 16 shows the compared results of the derailment coefficient for the front-outside wheel of the leading bogie. As shown in the figure, the value gets larger as the value of μ increases. The shape of curve in numerical simulation is relatively moderate.

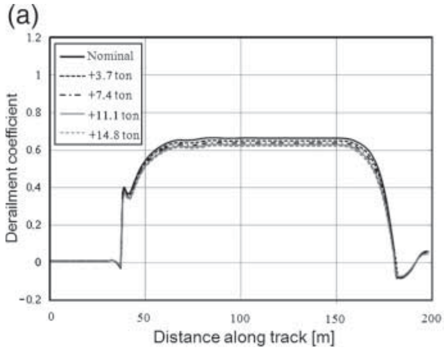

Figure 17. Effect of vehicle load on the value of derailment coefficient – simulation results. (a) Effect of vehicle body weight increase. (b) Effect of wheel load unbalance (reduction ratio between outside and inside wheel load).

On the other hand, the actual data are fluctuating. The difference in the steady-state value of the derailment coefficient is well understood to be produced by changing the value of μ as shown in numerical simulation results.

There are numerous factors that influence the time series behaviour of the derailment coefficient during curving. The most dominating factor may be the value of friction coefficient as shown in Figure 16. Furthermore, the track irregularity, such as gauge and cross-level irregularity, may greatly affect the fluctuation of the value along a single curve.

In Figure 17, numerical results regarding the change of vehicle body weight and the wheel load unbalance ratio are shown. As shown in Figure 17(a), the value of vehicle body mass does not significantly affect the results. On the other hand, the ratio of wheel load unbalance greatly affects the value of derailment coefficient as shown in Figure 17(b).

In order to comprehend these phenomena in detail, further investigation is necessary. These phenomena can be solved by numerical simulations, considering accurate track irregularities and a more sophisticated vehicle model.

5. Conclusions

By using non-contact gap sensors fitted on non-rotating parts of a bogie, a new measuring method of wheel/rail contact forces has been successfully developed. After train running tests on a commercial line, the authors verified that the system has sufficient durability for continuous measurement on in-service trains and sufficient accuracy compared with conventional methods.

After continuous measurement on a commercial line for a long period, a number of data were obtained. According to the analysis of this data, derailment coefficients increase corresponding to the increase in friction on the inside wheel/rail contact surface, which is influenced by the timing of wayside lubricant application. On the other hand, the fluctuations of the coefficients by positional change along each curve are rather similar for each passing train, which is considered to be produced by track irregularity, vehicle vibration specified by the feature of each curve.

Fundamental numerical simulation of curving by MBD software was carried out, and the steady-state values of derailment coefficients considering μ agree with the measured data. The fluctuation of the coefficients influenced by various factors, such as track irregularity and wheel load reduction ratio, will be analysed in further studies.

References

[1] Accident investigation committee, *Investigation report on the train collision accident in Nakameguro Sta. of Hibiya Line of Teito rapid transportation authority*, Ministry of Transport, 2000, p. 47 (in Japanese).

[2] H. Ishida, M. Matsuo, K. Tezuka, and K. Ueki, *Method of measuring wheel and rail contact forces and derailment quotients continuously: Development of a measuring device*, Trans. JSME 63c(614) (1997), pp. 3417–3423 (in Japanese).

[3] P. Gullers, L. Anderson, and R. Lunden, *High-frequency vertical wheel–rail contact forces – field measurements and influence of track irregularities*, Wear 265(9–10) (2008), pp. 1472–1478.

[4] M. Wrang, *Expanding the potential of load-measuring wheelsets*, Rail. Gazette Int. 162(9) (2006), p. 61.

[5] A. Matsumoto, Y. Sato, H. Ohno, M. Tomeoka, K. Matsumoto, J. Kurihara, T. Ogino, M. Tanimoto, Y. Kishimoto, Y. Sato, T. Nakai. *A new measuring method of wheel–rail contact forces and related considerations*, Wear 265(9–10) (2008), pp. 1518–1525.

[6] A. Matsumoto, Y. Sato, H. Ohno, T. Mizuma, M. Tanimoto, Y. Oka, and Y. Suda, *Development of a Bogie Rolling Test Facility for the Experiment of Curving Performance and its Application*, Proceedings of the 6th International Conference on Railway Bogies and Running Gears, Budapest, 2004, pp. 191–200.

[7] A. Matsumoto, Y. Sato, H. Ohno, T. Mizuma, Y. Suda, M. Tanimoto, and Y. Oka. *Study on curving performance of railway bogies by using full-scale stand test*, Veh. Syst. Dyn. 44(supp. 1) (2006), pp. 862–873.

[8] M. Shimizu, M. Tomeoka, K. Matsumoto, A. Matsumoto, Y. Sato, H. Ohno, M. Tanimoto, and Y. Sato, *A New Monitoring System for Derailment Coefficients Without a Special Wheelset – 2nd report*, Proceedings of Bogie'10, Budapest, 2010, pp. 231–238.

[9] M. Tanomoto, E. Miyauchi, M. Tomeoka, K. Matsumoto, M. Shimizu, T. Saitou, and Y. Sato, *Monitoring System of Derailment Coefficient with Sensor-Equipped Rail in Service Line*, Proceedings of Bogie'10, Budapest, 2010, pp. 383–389.

[10] R.J. Anderson, *A'GEM Rail Vehicle Dynamics Software Package Users Manual*, Queen's Univ., 1996.