INTRODUCING A NOVEL METHOD FOR MEASURING ROLLING RESISTANCE AND DYNAMIC VERTICAL LOAD ON A SEMI-TRAILER SUSPENSION

W. Levesque
PhD student and lecturer in mechanical engineering at Université Laval. Currently working on a research project about the effects of pavement characteristics on fuel consumption of heavy vehicles at the traffic level. The research project means to provide practical insight to policy-makers regarding road network management and greenhouse gas minimization.

A. Bégin-Drolet
Professor of mechanical engineering at Université Laval in Canada and CEO of Instrumentation Icetek a spinoff company launched in November 2020 that commercialize an ice sensor he and his team have developed at Université Laval. His research focuses towards sustainable mechanical design as well as the development and management of intelligent objects and instruments.

J. Lépine
Professor at Université Laval in Canada specialized in sustainable freight transport from an engineering, operational and logistic approach. He is former research associate at the Centre for Sustainable Road Freight and the University of Cambridge UK Obtained his PhD in mechanical engineering from Victoria University, Australia.

Abstract

This paper presents the development of a new instrumentation system to measure rolling resistance and dynamic vertical load on a semi-trailer suspension. The aim of this novel measurement approach is to offer greater realism than laboratory experiments on steel drums, such as those standardized by ISO and SAE, and higher accuracy than indirect methods such as fuel consumption measurements. To compute the rolling resistance coefficient and the dynamic vertical load, the instrumentation system measures all forces acting between the suspension and the semi-trailer frame to solve the corresponding free body diagram. In addition, direct measurement of energy dissipated by suspension dampers provides a more complete understanding of rolling resistance, linking it directly to road roughness (wavelengths from 0.5 to 50 m). The instrumentation system employs various sensors, including a custom designed bidirectional load cell developed at Université Laval, capable of withstanding full semi-trailer braking force without mechanical failure and preventing mechanical crosstalk between longitudinal and vertical force measurements. Laboratory validation has confirmed these two features of the load cell. The instrumentation system was designed to be used on actual roads to correlate measurements with pavement characteristics, further improving the understanding of rolling resistance and dynamic vertical load.

Keywords: Rolling Resistance, Dynamic vertical load, Measurements, Road Roughness, Pavement Characteristics

1 Introduction

Rolling resistance is a topic of interest for tire manufacturers as it is one of the main causes of energy consumption in heavy vehicles (1,2). Rolling resistance is typically measured in laboratory conditions by rolling a tire on a steel drum, an approach that has been standardized by ISO and SAE (3). However, this method has some limitations such as abnormal tire deformation caused by the curvature of the drum (4,5). Despite the availability of a formula to account for drum curvature (6), there is still debate in the literature on this topic (7,8). Furthermore, real-world operating conditions, such as temperature and inflation pressure, can significantly affect rolling resistance (8) and these effects are not captured by the temperature limits set in ISO and SAE standards (9). Additionally, laboratory measurements of rolling resistance do not account for the impact of pavement characteristics (*e.g.* pavement macrotexture, road roughness, structure-induced rolling resistance) (10). Therefore, it is necessary to develop a methodology for measuring rolling resistance on actual roads to overcome these limitations.

There are indirect on-road methods to estimate the energy dissipation from rolling resistance of heavy vehicles, which are all subject to driving resistance forces not related to rolling resistance (*e.g.* aerodynamic drag, uphill travel). For instance, coast-down tests can be performed to estimate the rolling resistance coefficient of a heavy vehicle (11). However, this approach has been found to be inconclusive when correlated with pavement characteristic metrics (12). An alternative method is to measure fuel consumption (*e.g.* with the Controller Area Network bus), which has been demonstrated effective when comparing different tire models (13,14). Nevertheless, when used to quantify the impact of pavement characteristics on energy consumption, this method has yielded inconsistent results across different studies (10,15,16). Thus, there is a need for a more direct and precise measurement approach to effectively capture the impact of pavement characteristics on rolling resistance of heavy vehicles. This insight can serve as a valuable tool for policy-makers to optimize road network management (17). For instance, some authors suggest that an increase in the elastic modulus of pavement surfaces can have a significant impact on greenhouse gas emissions of the transportation sector (18). Moreover, this novel measurement approach could highlight potential discrepancies between laboratory tests and direct on-road rolling resistance measurements (4).

2 Methodology

To measure rolling resistance and dynamic vertical load of a heavy vehicle with a direct approach under real driving conditions, a semi-trailer (flat bed, 3 axles, dual tires) operated by a truck driving school (*Centre de Formation en Transports de Charlesbourg*) has been selected for the experiment. The semi-trailer suspension (Hendrickson) on the central axle was chosen for installing the instrumentation system as it experiences fewer lateral forces in case of turning. The instrumentation system was designed to bring a minimal level of mechanical modifications to the suspension in order to keep the same dynamic behaviour under the effect of road roughness (wavelengths from 0.5 to 50 m). In addition, its design enables simple removal of experimental equipment for reinstallation of original components.

The instrumentation system was designed to perform rolling resistance and dynamic vertical load measurements under typical highway conditions. To ensure maximum reliability in measurements, it is optimal to use the instrumentation system at constant speed (*e.g.* 105 km/h) on flat sections of highways. Measurements can be performed on different pavement types of various conditions (*e.g.* road roughness, age) to evaluate the variability in rolling resistance caused by pavement characteristics. Such measurements at different times of the year can be used to evaluate the impact of meteorological conditions on the interaction between pavement characteristics and rolling resistance (19).

2.1 Design of the Instrumentation System

The main idea of the instrumentation system is to measure the forces at each physical link between the suspension and the frame of the semi-trailer. By solving the free body diagram of the suspension system, it is possible to isolate the forces related to rolling resistance and the load supported by the tire which results in the computation of the rolling resistance coefficient. All the required physical quantities to measure and their respective location on the semi-trailer suspension are summarized in Figure 1.

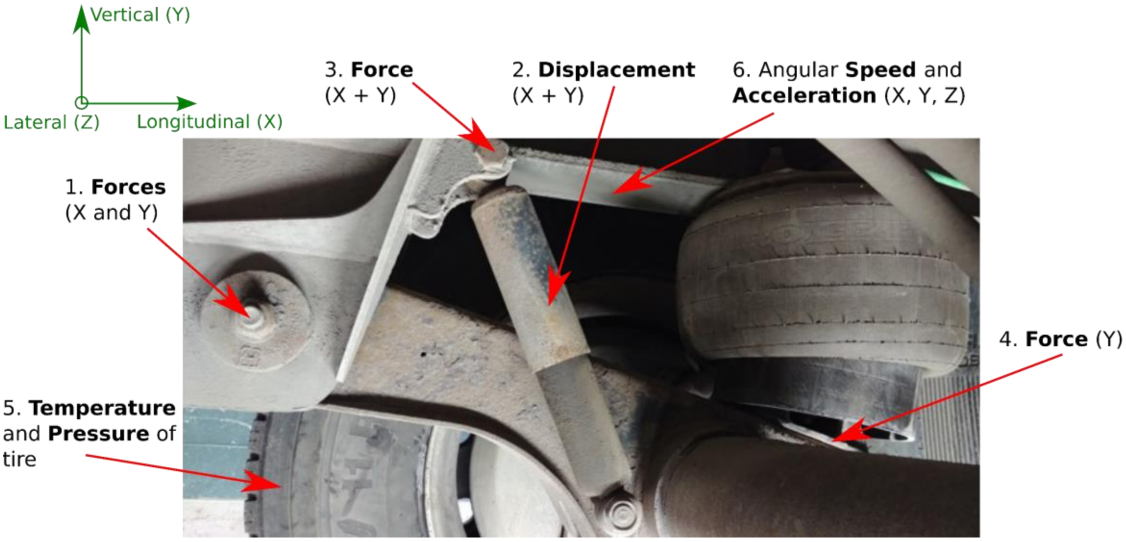

Figure 1: Locations on the semi-trailer suspension of the physical quantities measured by the instrumentation system.

Instrumented components were designed or selected to replace certain existing components within the suspension system. The various instruments are summarized in Table 1.

Table 1 : List of the different instruments installed on the semi-trailer suspension.

Num.	Measurement type	Location	Instrument
1	Force	Flexible Bushing	Custom bidirectional load cell
2	Displacement	Damper	Linear potentiometer
3	Force	Fixed pivot of the damper	Load pin
4	Force	Suspension air bellow	Load cell
5	Temperature and Pressure	Tire (fixed on rim)	Tire Temperature and Pressure Monitoring System (TTPMS)
6	Speed and acceleration	Semi-trailer I beam	Accelerometer and gyroscope (*InvenSense*, MPU-6050)

The longitudinal and vertical forces that occur on each side of the frame bracket (see position 1 in Figure 1) can be measured by a load cell. The design requirements for such load cell are as follows:

- Installed without any permanent modifications on the Hendrickson suspension.
- Offers a longitudinal measurement range of ±2 kN and mechanically resists to the semi-trailer braking forces (up to +20 kN per load cell).
- Has a low mechanical crosstalk between longitudinal and vertical measurement signals which are more than an order of magnitude different.

The magnitude of the rolling resistance force is expected to be less than 0.2 kN on average per load cell. However, the force in the longitudinal direction of the load cell is expected to reach up to 2 kN when the damper is solicited due to road roughness.

Since no load cell on the market was appropriate for this application, a custom bidirectional load cell was designed and built at Université Laval. The load cell was designed to undergo longitudinal displacement proportional to the applied longitudinal force. When the threshold value is exceeded (*e.g.* 2 kN), a mechanical end stop has the effect of transferring the exceeding mechanical force from the weak regions of the load cell to the surrounding parts. This is a crucial aspect that ensures the load cell mechanical integrity during braking which enables the safe use of the instrumentation system on actual roads.

The displacement of the damper (see position 2 in Figure 1) has been planned to be measured using a linear potentiometer. It is estimated that the energy dissipation caused by road roughness originates almost exclusively from suspension damping (20). Moreover, the linear displacement of the damper measured by the linear potentiometer was designed to be used as an input in a kinematic/dynamic model of the suspension, which enables the estimation of relevant parameters such as the instantaneous orientation of the damping force.

A load pin has been selected for the measurement of the damping force (see position 3 in Figure 1). Direct measurement of damping force addresses the issue of non-constant damping coefficient values, which could vary with operating temperature and the direction of damper displacement (extension *vs* compression). By combining damping force and damper movement measurements, the direct calculation of energy dissipation in the suspension can be achieved, which is thought to be closely linked to road roughness (21). In addition, damping force measurement enables the consideration of its instantaneous effect on other load cells in the instrumentation system.

The vertical load supported by the air bellow was designed to be measured with a load cell installed beneath it (see position 4 in Figure 1). This measurement facilitates the calculation of the total load supported by the suspension and establishes a direct correlation with dynamic vertical load. In addition, this load cell measurement offers redundancy in measuring vertical force, which enhances data accuracy and error detection during data post-processing. Indeed, by considering the various dimensions of the suspension, the vertical force measured at the air bellow can be used to corroborate the vertical force measured at the flexible bushing.

To account for tire operating conditions, a Tire Temperature and Pressure Monitoring System (TTPMS) was selected to be installed inside the tires of the suspension on each side of the semi-trailer (see position 5 in Figure 1). Measuring the internal temperature of the tire is considered more pertinent than the external temperature for rolling resistance measurement, as it provides a better reference for the thermal conditions of the tire (9). Moreover, the internal temperature of a tire has a good correlation with its operating inflation pressure, both being interdependent following the perfect gas law (9). In the instrumentation system, the internal temperature of the tire at 16 different points of the cross section of the tire and the operating inflation pressure were identified to be measured at a frequency of 0.1 Hz. The use of the TTPMS allows to consider the effect of tire operating conditions on rolling resistance measurements. Furthermore, this approach facilitates the execution of the warm-up phase at the beginning of the measurements.

Accurate road grade estimation is crucial for rolling resistance coefficient measurements because a non-zero value of road grade implies that gravitational force has a non-zero component in the longitudinal direction of the vehicle (*i.e.* suspension mass). To minimize the disturbances caused by road grade and ensure high spatial resolution of the measured rolling resistance coefficient, a 3-axis accelerometer and 3-axis gyroscope (MPU-6050) was selected to be installed on the semi-trailer I-beam (see number 6 in Figure 1). The role of the MPU-6050 is to allow the calculation of the pitch angle of the semi-trailer I-beam, which is assumed to be equal to the road grade. By knowing the instantaneous road grade, it would be possible to consider its interaction on the various sensors. Since motion tracking devices such as the MPU-6050 are subjected to "random walk" and "bias instability", the latter being problematic during long measurements, an Allan variance test (22) was performed on the MPU-6050 to determine the character of the underlying noise processes which are presented in Table 2.

Table 2 : Results of the Allan Variance Test on the MPU-6050.

Axis	Gyroscopes		Accelerometers	
	Bias Instability [°/h]	Angle Random Walk $[°/\sqrt{\mathrm{h}}]$	Bias Instability $[(\mathrm{m/s})/\mathrm{h}]$	Velocity Random Walk $[(\mathrm{m/s})/\sqrt{\mathrm{h}}]$
X	29.2 (at 12 sec)	1.09	2.60 (at 140 sec)	0.294
Y	9.42 (at 470 sec)	0.876	1.19 (at 450 sec)	0.264
Z	9.65 (at 220 sec)	0.804	5.95 (at 97 sec)	0.426

The values of Table 2 are useful to determine the appropriate averaging window for road grade estimation during data post-processing. In addition, the values derived from the Allan variance test can be implemented in more advanced filtering methods (*e.g.* Kalman filter) to improve the accuracy of the road grade estimation (23), which is expected to help ensure high spatial resolution of the measured rolling resistance coefficient.

The sensors 1, 2, 3, and 5 in Table 1 were selected to be installed on each side of the suspension to account for any potential variability between the left and right side of the instrumented suspension (*e.g.* wheel misalignment, and road rutting). All the forces and displacement sensors (numbers 1 to 4 in Table 1) were connected to two custom-printed circuit boards to perform data acquisition. The data acquisition system was designed to sample data at a rate of 16 kHz, and the collected data is subsequently downsampled to 1 kHz through averaging before being saved on an SD card. Additionally, a GPS was included in the system for geolocation of the measured data, which is essential to express the results according to the characteristics of the pavement.

2.2 Data Analysis

The measurements of the various sensors installed on the suspension were integrated in a kinematic/dynamic model in order to compute the instantaneous rolling resistance coefficient. This model allows considering the oscillations of the suspension under the effect of road roughness or a non-zero road grade or vehicle acceleration.

The kinetic/dynamic model of the suspension (Figure 2) is based on these hypotheses:

1. The flexible Bushing within the frame bracket is considered rigid and represents the pivot point of the suspension motion.
2. The center of mass of the suspension system is coincident with the axis of the axle, which follows an oscillatory circular motion around the flexible Bushing.
3. The roll motion of the suspension is negligible, which justified a two-dimensional modelling.
4. The damper's orientation is aligned with the vertical plane.
5. The air bellow only transmits force in the vertical direction of the vehicle reference frame.
   a. This assumption is based on the uniform inflation pressure within the air bellow and its attachment to the semi-trailer I-Beam.
   b. Additionally, the flexible material of the air bellow is assumed to transmit no shear forces due to its lower longitudinal stiffness compared to the flexible Bushing.

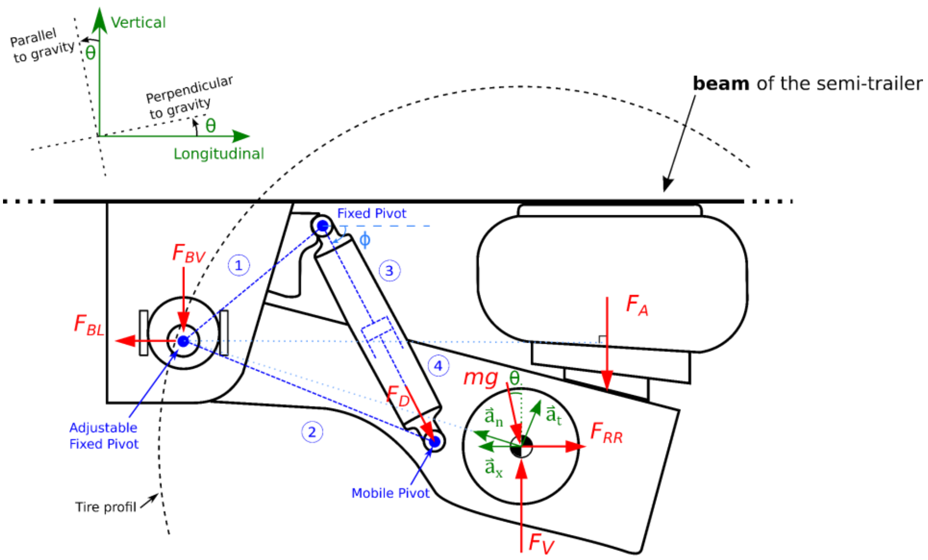

Figure 2: Free body diagram of the upward-moving semi-trailer suspension: forces, acceleration vectors, and four-bar mechanism during uphill travel.

Figure 2 depicts a generalized model of a suspension system accelerating upwards on a semi-trailer that is accelerating uphill (see reference frames at the top left corner). In the proposed instrumentation system, the forces F_{BV} and F_{BL} are measured by two custom bidirectional load cells (one on each side of the frame bracket). These forces are depicted in the vehicle reference frame since they are fixed on the frame bracket and rotate with it. The force F_D is the damping force that occurs when there is a motion of the suspension relative to the I-beam of the semi-trailer. The force mg corresponds to the weight of the suspension itself (*e.g.* axle, wheels) and acts in the direction of gravity. The force F_A is the load supported by the air bellow and is always oriented in the vertical direction of the vehicle reference frame. The forces F_{RR} and F_V are the rolling resistance force and the vertical load supported by the tire respectively. The coefficient of rolling resistance is given by:

$$C_{RR} = \frac{F_{RR}}{F_V} \tag{1}$$

The acceleration of the center of mass of the suspension system is the vectorial sum of $\vec{a}_n$, $\vec{a}_t$, and $\vec{a}_x$ which are all depicted in green in Figure 2. The vectorial sum of $\vec{a}_n$, $\vec{a}_t$ is caused by the relative motion of the suspension under the effect of road roughness. The component $\vec{a}_x$ is caused by the vehicle speed variations. This can be summarized with the following equation:

$$\vec{a} = \vec{a}_n + \vec{a}_t + \vec{a}_x = \vec{a}_L + \vec{a}_V \tag{2}$$

Where $\vec{a}_L$ and $\vec{a}_V$ are the two orthogonal acceleration vectors in the two-dimensional vehicle reference frame. By solving the free body diagram of Figure 2, the following equations were established.

$$F_{RR} = m\|\vec{a}_L\| - F_D\cos(\phi) + F_{BL} - mgsin(\theta) \tag{3}$$
$$F_V = m\|\vec{a}_V\| + F_D\sin(\phi) + F_{BV} + mgcos(\theta) + F_A \tag{4}$$

As can be seen in equations 3 and 4, it is necessary to calculate the acceleration of the semi-trailer ($\vec{a}_x$), the road grade (θ) and the kinematic states of the suspension (ϕ, $\vec{a}_n, \vec{a}_t$) in order to calculate the instantaneous rolling resistance coefficient. The acceleration vector $\vec{a}_x$ and the road grade θ are expected to be measured with the MPU-6050 installed on the semi-trailer I-beam (see number 6 in Table 1). The rest of the parameters aforementioned can be inferred from the measurements of the linear potentiometer installed on the damper (see number 2 in table 1).

As depicted with the blue triangle in Figure 2, the frame bracket (bar 1), the suspension frame (bar 2) and the damper (bars 3 and 4) can be modelled as a four-bar mechanism which allows calculating the kinematic states of the suspension (ϕ, $\vec{a}_n, \vec{a}_t$) based on the measurements of the linear potentiometer installed on the damper. Since every length of the four bars is either constant or measured, the loop closure equations can be used to compute analytically the orientation of the damper in the vehicle reference frame (ϕ). Additionally, by knowing the position of the center of mass relative to the flexible Bushing, the kinematic model allows computing the acceleration vectors $\vec{a}_n, \vec{a}_t$ caused by road roughness.

3 Results and Discussion

Before installing the instrumentation system on the semi-trailer, the custom bidirectional load cell was tested in laboratory conditions. Two crucial technical aspects were assessed regarding this load cell: 1) its mechanical integrity in case of braking; 2) its ability to cancel out mechanical crosstalk between longitudinal and vertical measurement signals.

To perform these laboratory experiments, an electric traction machine (MTS Insight 100) with a capacity of 100 kN has been used. This machine was commanded in position and a reference load cell of a capacity of 10 kN was used to measure the force applied. Even if this latter value is relatively higher than the required measurement range of longitudinal force (*i.e.* +/-2 kN), the electric traction machine has been judged practical and accurate enough for proof of concept. Moreover, a reference load cell with a higher capacity (*i.e.* 100 kN) was used to simulate the braking of the semi-trailer suspension with the traction machine. All the measurement signals are normalised in mV/V, which corresponds to the intensity of the Wheatstone bridge measurement signal for every volt of its supply voltage.

3.1 Braking Test

An experimental setup has been designed and fabricated to integrate the custom bidirectional load cell on the electric traction machine in a configuration where only longitudinal force is applied. By doing so, longitudinal forces from 0 to 20 kN were developed in order to observe the evolution of the longitudinal measurement signal, the latter being a direct way to assess the mechanical integrity of the load cell. The results are depicted in Figure 3.

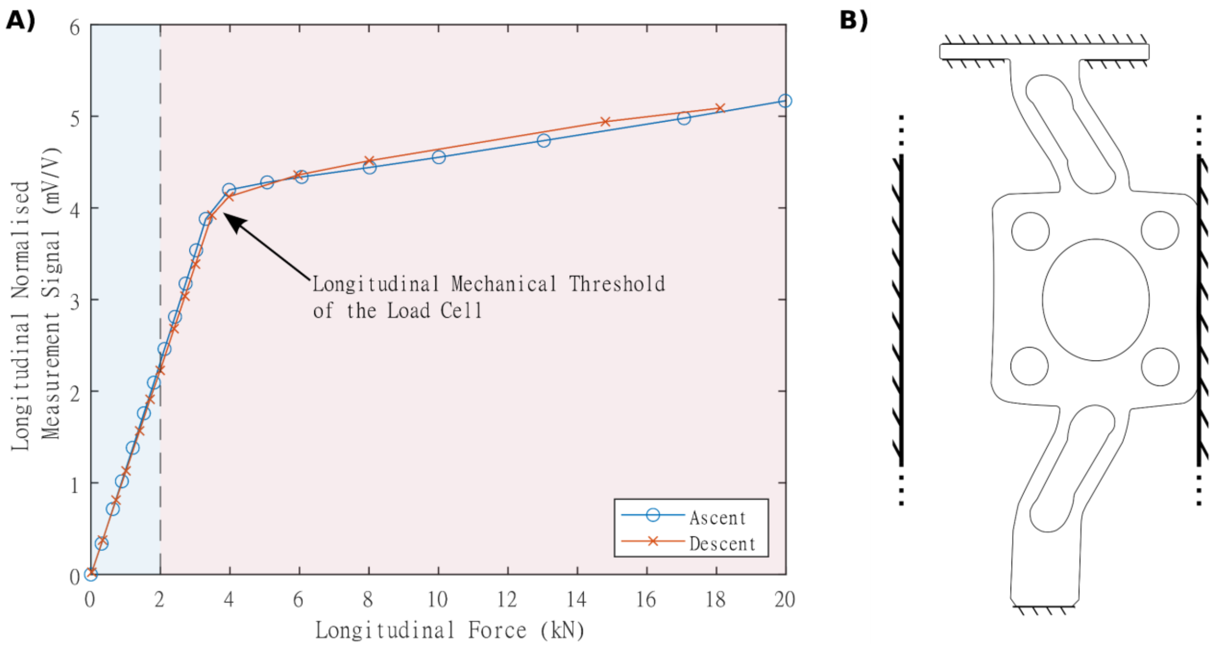

Figure 3: A) Braking test of the custom bidirectional load cell using the electric traction machine; measurement zone in blue from 0 to 2 kN and braking zone in red from 2 to 20 kN; B) Schematic representation (not to scale) of the load cell deformation under the effect of a longitudinal force when the mechanical end stop is reached.

The blue zone in Figure 3A corresponds to the expected required measurement range in the longitudinal direction due to the combined effect of rolling resistance and road roughness (*e.g.* damping force) for each load cell installed on the semi-trailer suspension. The red zone in Figure 3A corresponds to the braking zone that does not require to be measured, but must not cause any failure of the load cell. Figure 3B depicts the exaggerated displacement of the bidirectional load cell when it undergoes a longitudinal force high enough to create contact with the mechanical end stop.

It can be observed that the longitudinal end-gap starts to act at a longitudinal force of 3.3 kN, which represents a normalized measurement signal of 3.9 mV/V. After this mechanical threshold, a longitudinal force increasing at 20 kN only increases the normalized measurement signal to a total value of 5.2 mV/V. Therefore, when the mechanical threshold of the load cell is exceeded, the increase in stress in the weaker regions of the load cell caused by the longitudinal force is reduced by a factor of 15. For this reason, it was concluded that the custom bidirectional load can be safely installed on the semi-trailer suspension to perform measurements on actual roads.

3.2 Mechanical Crosstalk Test

The mechanical crosstalk of the custom bidirectional load cell is the interaction between measurement signals (*i.e.* longitudinal and vertical) related to the strain distribution within the load cell material. Indeed, it is possible that the vertical force has a non-negligible effect on a Wheatstone bridge designed to measure a longitudinal force, and vice versa.

To investigate this aspect, a second experimental setup has been designed and fabricated to install the custom bidirectional load cell with a known angle relative to the force created by the traction machine. By measuring this angle with a digital inclinometer, it was possible to estimate the longitudinal and vertical forces that were applied simultaneously on the custom bidirectional load cell and evaluate the potential mechanical crosstalk in the measurement signals.

This aspect is very important because it is expected that road roughness can induce a significant vertical force variation on the custom bidirectional load cells. Indeed, the coefficient of variation (CV) of the dynamic vertical load on a heavy vehicle axle can be estimated with the following empirical formula (26).

$$CV = V^a R^b \tag{5}$$

Where,

CV: Coefficient of variation (standard deviation divided by mean) [%]
V: Speed of the vehicle [km/h]
R: International Roughness Index [m/km]
a, b : Empirical constants for Air-Spring Suspensions [0.346 and 0.798 respectively]

For a semi-trailer with air suspensions travelling at 105 km/h on a pavement with an International Roughness Index of 3 m/km, the corresponding coefficient of variation at the axle location is 12.0 %. Considering a fully loaded suspension and its various dimensions, the vertical force variation at twice the standard deviation for each custom bidirectional load cell is estimated at ± 1.2 kN.

The custom bidirectional load cell was tested in laboratory at more extreme conditions where the applied force was varied from 0 to 8.5 kN. The angle between longitudinal and vertical forces was varied from 5 to 20 degrees.

The experiment demonstrated that employing a two-dimensional regression model on both measurement signals (*i.e*. longitudinal and vertical) proved to be optimal for establishing a precise statistical correlation between the measurement signals and applied forces. This approach remains suitable for load cell calibration, given the condition of maintaining linearity within the regression.

In the bidimensional linear regression model, the effect of both measurement signals (*i.e.* longitudinal, vertical) are added and two sums are obtained to estimate both forces independently. The bidimensional linear regression model is presented below:

$$F'_{BL} = R_1 S_L + R_2 S_V \tag{6}$$
$$F'_{BV} = R_3 S_L + R_4 S_V \tag{7}$$

Where,

F'_{BL} is the applied longitudinal force on a load cell [N]
F'_{BV} is the applied vertical force on a load cell [N]
S_L is the longitudinal measurement signal [mV/V]
S_V is the vertical measurement signal [mV/V]
R_1, R_2, R_3, R_4 are regression coefficients

The forces F_{BL} and F_{BV}, as depicted in the free body diagram of Figure 2, result from the combination of two forces measured by two custom bidirectional load cells: F'_{BL} and F'_{BV}, respectively. It is anticipated that these latter values should be approximately equal when measured by two custom bidirectional load cells installed on the same frame bracket."

The objective of the calibration procedure was to obtain the regression coefficients of equations 6 and 7. The bidimensional linear regression models of one of the custom bidirectional load cells for both longitudinal and vertical forces are depicted in Figure 4.

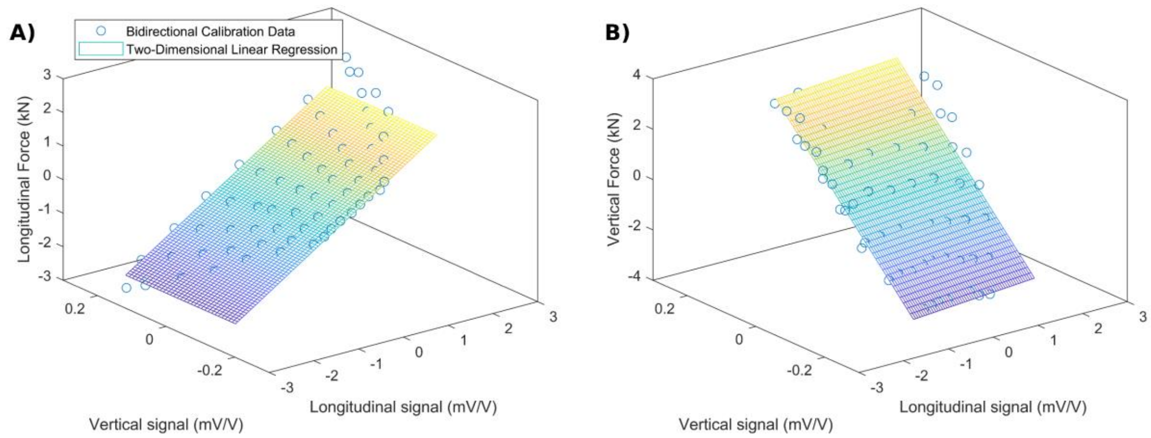

Figure 4: A) Bidimensionnal calibration for longitudinal force estimation based on longitudinal and vertical measurement signals. B) Bidimensionnal calibration for vertical force estimation based on longitudinal and vertical measurement signals.

The small but non-negligible spreading of the data in both images of Figure 4 is believed to be partly due to the resolution of the digital inclinometer (*i.e.* 0.1°). Even a slight estimation error in the angle of the applied force from the traction machine can exert a discernible impact on the signals. This concern is heightened by the observed angle drift of up to 0.2° during the experiment. For this reason, it is believed that improvements in the experimental setup could improve the accuracy of the calibration procedure.

In addition, there was a non-negligible difference in regression coefficients obtained with the calibration procedure between each custom bidirectional load cells. This is believed to be due to the strain gauge transverse sensitivity (24) and the potential misalignment of the strain gauges during bonding to the surface of the load cell (25). This highlight the importance of calibrating each custom bidirectional load cells individually.

From the data depicted in Figure 4, it was concluded that the custom bidirectional load cell has the potential to measure longitudinal and vertical forces on the semi-trailer suspension without any significant mechanical crosstalk.

4 Conclusion

This paper presented a novel instrumentation system to measure the rolling resistance and the dynamic vertical load on a semi-trailer suspension. Various sensors were presented to measure the forces transmitted between the suspension and the semi-trailer frame which allows solving the free body diagram of the suspension and compute the rolling resistance coefficient. This includes the usage of a custom bidirectional load cell developed at the Université Laval. The custom bidirectional load cell was tested in laboratory conditions and it was proven to be capable of withstanding full braking force without any mechanical failure. This means that the instrumentation system presented in this paper can be safely used on actual roads. It was also demonstrated that the custom bidirectional load cell can adequately compensate for mechanical crosstalk. The instrumentation system was designed to be used on real roads to correlate measurements with pavement characteristics, which is expected to give insight about rolling resistance and dynamic vertical load under real driving conditions. Indeed, it is hypothesized that this novel approach will offer greater realism than standardized laboratory experiments on steel drums, such as those standardized by ISO and SAE, and higher accuracy than indirect methods such as fuel consumption measurements.

5 References

1. Sandberg T. Heavy Truck Modeling for Fuel Consumption Simulations and Measurements [Internet]. (Doctoral dissertation, Linköping University); 2001 [cited 2020 Nov 28]. Available from: http://www.fs.isy.liu.se/Publications/Lic/01_LIC_924_TS.pdf
2. Odhams AMC, Roebuck RL, Lee YJ, Hunt SW, Cebon D. Factors influencing the energy consumption of road freight transport. Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science. 2010;224(9):1995–2010.
3. Popov AA, Cole DJ, Winkler CB, Cebon D. Laboratory measurement of rolling resistance in truck tyres under dynamic vertical load. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering. 2003;217(12):1071–9.
4. Vieira T, Sandberg U, Erlingsson S. Rolling Resistance Evaluation of Winter Tires on In-Service Road Surfaces. Tire Science and Technology [Internet]. 2020 Mar 27 [cited 2020 Sep 16];48(2). Available from: /doi/abs/10.2346/tire.20.190225
5. Ydrefors L. The relationship between rolling resistance and tyre operating conditions, with a focus on tyre temperature. (Doctoral dissertation, KTH Royal Institute of Technology); 2022.
6. Clark SK. Rolling Resistance Forces in Pneumatic Tires. 1976.
7. Freudenmann T, Unrau H-J, El-Haji M. Experimental Determination of the Effect of the Surface Curvature on Rolling Resistance Measurements. Tire Science and Techology. 2009;37(4):254–78.
8. Ydrefors L, Hjort M, Kharrazi S, Jerrelind J, Stensson Trigell A. Rolling resistance and its relation to operating conditions – a literature review. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering. 2021;235(12):2931–48.
9. Ejsmont J, Ronowski G, Owczarzak W, Sommer S. Temperature Influence on Tire Rolling Resistance Measurements Quality. International Journal of Automotive Technology. 2022;23(1):109–23.
10. Levesque W, Bégin-Drolet A, Lépine J. Effects of Pavement Characteristics on Rolling Resistance of Heavy Vehicles : A Literature Review. Transportation Research Record. 2023;1–21.
11. Madhusudhanan AK, Ainalis D, Na X, Garcia IV, Sutcliffe M, Cebon D. Effects of semi-trailer modifications on HGV fuel consumption. Transportation Research Part D: Transport and Environment [Internet]. 2021;92:102717. Available from: https://doi.org/10.1016/j.trd.2021.102717
12. Karlsson R, Hammarström U, Sörensen H, Eriksson O. Road surface influence on rolling resistance Coastdown measurements for a car and an HGV. VTI notat 24A-2011. 2011.
13. Kienhöfer F, Na X, de Saxe C, Abdulla R, Venter H, Cebon D. Experimental Evaluation of Low Rolling Resistance Tyres for Heavy Goods Vehicles in South Africa. In: Heavy Vehicle Transport & Technology. 2021. p. 1–11.
14. Na X, Cebon D. Quantifying Fuel-Saving Benefit of Low-Rolling-Resistance Tyres from Heavy Goods Vehicle In-Service Operations. SSRN Electronic Journal. 2022;113(July).
15. Taylor G, Patten J. Effects of Pavement Structure on Vehicle Fuel Consumption. Centre for Surface Transportation Technology (CSTT), National Research Council of Canada (NRC), Ontario, Canada. 2006;
16. Butt AA, Harvey JT, Fitch D, Kedarisetty S, Lea JD, Lea J, et al. Investigation of the Effect of Pavement Deflection on Vehicle Fuel Consumption: Field Testing and Empirical Analysis. 2022.
17. Skar A, Vestergaard A, M.Pour S, Pettinari M. Internet-of-Things (IoT) Platform for Road Energy Efficiency Monitoring. Sensors [Internet]. 2023;23(2756):1–9. Available from: https://doi.org/10.3390/s23052756
18. AzariJafari H, Gregory J, Kirchain R. Potential Contribution of Deflection-Induced Fuel Consumption to U.S. Greenhouse Gas Emissions. Transportation Research Record. 2020;2674(8):931–7.
19. Levesque W, Samson N, Bégin-drolet A, Lépine J. Pavement Effects on Heavy-Vehicle Fuel Consumption in Cold Climate : A Statistical Approach. :1–39.
20. Kang S, Ozer H, Al-Qadi IL, Spencer BF. Stochastic Analysis of Rolling Resistance Energy Dissipation for a Tractor-Trailer Model. Transportation Research Record. 2019;2673(11):593–603.
21. Louhghalam A, Akbarian M, Ulm F-J. Roughness-Induced Pavement–Vehicle Interactions : key Parameters and impact on Vehicle fuel consumption. Transportation Research Record. 2015;2525(1):62–70.
22. Woodman OJ. An introduction to inertial navigation. American Journal of Physics. 2007.
23. Li E, He W, Yu H, Xi J. Model-Based Embedded Road Grade Estimation Using Quaternion Unscented Kalman Filter. IEEE Transactions on Vehicular Technology. 2022;71(4):3704–14.
24. Vishay Precision Group. TN-509, Errors Due To Transverse Sensitivity in Strain Gages [Internet]. 2011. Available from: https://intertechnology.com/Vishay/pdfs/TechNotes_TechTips/TN-509.pdf
25. Vishay Precision Group. TN-511, Errors Due to Misalignment of Strain Gages [Internet]. 2010. Available from: https://intertechnology.com/Vishay/pdfs/TechNotes_TechTips/TN-511.pdf
26. Papagiannakis AT, Masad EA. Pavement Design and Materials. John Wiley & Sons I, editor. Hoboken, Nwe Jersey; 2008.