A FEM MODEL TO COMPARE MEASUREMENT LAYOUTS TO EVALUATE THE WHEEL-RAIL CONTACT FORCES.

Andrea Rindi, Luca Pugi, Susanna Papini Stefano Falomi

*Dipartimento di Energetica, Sez.Meccanica Applicata, Università degli Studi di Firenze
Via S.Marta 3, 50139 FIRENZE Italy

e-mail:susanna@mapp1.de.unifi.it, luca@mapp1.de.unifi.it

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Abstract.
International standards require the measure of wheel-rail contact forces, to characterize the railway dynamics in terms of safety and comfort of the passengers [1],[2]. Several systems have been developed to measure these quantities during the testing of railway vehicles[3],[3],[5]. These systems measure the deformation of wheelset and wheels, or the relative displacement between the wheel and a non-rotating component. However reliability and repeatability of proposed methods is often a critical matter also considering tolerances and errors introduced in the assembling of the system.
Also in braking conditions, heat rate and additional loads introduced by brake discs and pads (installed on the wheelset) may negatively affect reliability and precision of sensors.
In this work authors propose an application case on which sensitivity and optimization analysis are performed.
In particular in this case is proposed the application to a bogie with a layout, characterised by an additional shaft on which brake discs are installed. This mechanical solution successfully applied on some existing locomotives like Siemens E190. The brake force is transmitted to the wheelset by a compact gear unit. The absence of discs on the wheelset allows to have an higher extension for the sensors’ placement and reduce the risk of thermal induced disturbances after prolonged brakings.
A FEM model of the whole bogie was developed, to compare different layouts and methodologies for the contact forces measure. A procedure able to evaluate the contact forces using the strain or displacement measurements has been performed on the system. In this it has been possible to evaluate an optimal layout for strain gage positioning and the resulting sensitivity against position tolerances of the sensing elements. Also a modal/dynamical analysis has been performed in order to evaluate possible bandwidth limitations arising from the flexible behaviour of the system.
1 INTRODUCTION

As stated by regulation in forces\cite{1,2}, in order to evaluate the ride quality of a railway vehicle vertical and lateral contact forces has to be measured. In this work it's adopted the typical convention visible in figure 1 where the three components of the measured contact force are indicated with the following nomenclature:

- the longitudinal com force $X$, direct as $x$-axle rail;
- the lateral force $Y$, direct as $y$-axle;
- vertical force usually indicated with Q.

For the evaluation of the ride quality according\cite{1}, measurements $Y$ and $Q$-components with a minimum bandwidth of 20 Hz is needed since the acquired signals before postprocessing have to be filtered with a filter with a cut off frequency of 20Hz. Also the measurement of the $X$-component of the contact force can be interesting both from a scientific point of view in order to identify the behavior of the wheel-rail friction factor especially when degraded adhesion condition occurs. Identification and modeling of wheel rail adhesion factor has also important industrial consequences for development, testing and homologation of safety relevant subsystems like Odometry algorithms for on board ATP-ATC systems, WSP (Wheel Slide Protection System), Traction controls.

Unfortunately calibration of $X$ force measurements is quite difficult and also the application of braking torques to the axle is associated to thermal and mechanical disturbances which negatively affect reliability and precision of the measurement.

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{Figure1.png}
\caption{The wheel rail contact forces, conventions adopted in this work}
\end{figure}

In order to reduce negative influence of braking forces a customized bogie, as the example visible in figure 2, may be designed, or modified for this kind of activities:

The proposed bogie has a standard “H” shaped steel frame inspired to a widely diffused design adopted as example also on coaches of ETR500 High Speed Train. In order to avoid/reduce the application of disturbances on measurements caused by braking the disks are flanged over an auxiliary shaft connected through a suspended gearbox to the axle.

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{Figure2.png}
\caption{Layout of bogie}
\end{figure}
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This mechanical solution is usually adopted on some well diffused locomotive like Siemens E190, typically running with a maximum service speed of about 200Km/h and a 22.5 t of axle weight so the proposed layout is considered reliable and feasible even considering an augmented values of bogie unsuspended masses/inertia. Use of an efficient WSP system is mandatory since a wheel flat can compromise/vanify calibration and precision of the contact force measurements. Also the solution proposed in figure 2 involve the availability of space to place sensor and other telemetry device on the shaft.

For the aim of this study a classical layout in which the three contact force components are acquired independently on its own sensor system is supposed:

- **Longitudinal force $X$ measurements**: longitudinal forces are reconstructed in terms of torque exchanged along the axle length.
- **Lateral force $Y$ Measurements**: lateral force measurement is performed by instrumenting the lateral deformation of the wheel using methods available in bibliography [5].
- **Vertical force $Q$ measurements**: vertical forces exchanged on wheels are measured through the estimation of the shear stress in different sections of the axle; the shear stress is evaluated by comparing bending stresses on adjacent instrumented sections.

## 2 LONGITUDINAL FORCE $X$ MEASUREMENT

Longitudinal forces $X$ are estimated from torque measurements on two instrumented sections on the axle which are located between wheels and the torque load applied by braking or traction system as visible in the scheme of figure 3.

Torque is measured using the Wheatstone bridge visible in figure 3, spurious load due to axial forces, bending, thermal disturbances are rejected.

![Strain Gauges Layout on a measurement section](image)

Figure 3: strain gauge on axle for longitudinal force measurements

From the measured torque $M_{tor}$ on the i-th section the longitudinal force $X_i$ of the i-th wheel is measured assuming a known constant rolling radius $r_w$ according relation (1)

$$X_i = \frac{M_{tor}}{r_w}$$

## 3 VERTICAL FORCE $Q$ MEASUREMENT

The $Q_i$ vertical component of the contact force on the i-th wheel is evaluated imposing the corresponding equilibrium relation (2) calculated according the simplified scheme of figure 4.

$$Q_i = F_{10} + F_{11}$$

Forces $F_{10}, F_{11}$ are the shear loads applied on two adjacent sections of the wheel; $F_{10}, F_{11}$ are evaluated calculated comparing the measured bending torques $M_{ijk}$ on two adjacent sections according (3)

$$F_i = \frac{M_{ii} - M_{ij}}{y_{ij} - y_{i0}}$$

where $y_{ijk}$ is the corresponding axial position of the measuring section.
Bending torques $M_{ijk}$ on each measurement section are estimated using the strain gauge layout visible in figure 4; since the axle is rotating an absolute position sensor able to measure the angular position of the axle is needed.

The measurement of $F_{10}$ and $F_{21}$ may be performed also using a load cell able to measure the load vertical load on bearings. This solution should be preferred if the design of the axle-bogie involves encumbrances limitations.

In this case load cells cylindrical/toroidal load cells can be inserted under the springs of the primary suspension system or as alternative the same springs may be used as sensible/deformable element of the suspension.

4 LATERAL FORCE Y MEASUREMENT

Lateral forces are measured through an array of 4 strain-gauges bridges placed on different angular positions over the wheel as visible in the simplified schemes of figure 5/a/b; the four voltage signals $E_1, E_{II}, E_{III}, E_{IV}$ are "vectorially summed" according (4) in order to obtain a mean tension $E_Y$ which is roughly proportional to the lateral force $Y$.

$$E_Y = \sqrt{(E_1^2 - E_II^2) + (E_{III}^2 - E_{IV}^2)}$$

The radius values $r_1$ and $r_2$ of the two circumferences on which strain gauges are placed have to be optimized in order to increase the sensitivity of the sensors to the lateral forces and to reject/eliminate cross sensitivity effects against spurious forces as the $Q$ component.
With the complete sensor layout used to calculate vertical $Q$ and lateral $Y$ components of the wheel rail contact forces is also possible to estimate the contact point position [6] assuming as a single contact point the point of application of the contact forces which satisfy the equilibrium of the wheels.

5 UNCERTAINTY ANALYSIS FOR LONGITUDINAL $X$ AND VERTICAL $Q$ FORCES

Uncertainty analysis was performed according the standard “UNI CEI ENV 13005”[7] and the corresponding nomenclature definitions according UNI CEI 70099[8]. In particular for $X$ and $Q$ considered as measurands are defined by the explicit relationships/functions (1) and (3) of a known set of independent variables/quantities $x_i$; As consequence it’s possible to define $X$ and $Q$ according functions (5),(6) considering the axle as a Bernoulli Beam:

$$Y = f_X(x_1, x_2, ..., x_N)$$  \hspace{1cm} (5)

$$Q = f_Q(x_1, x_2, ..., x_N)$$  \hspace{1cm} (6)

Each quantity $x_i$ is subjected to a standard uncertainty $u_i$ deriving from measurement errors or by natural tolerances considering system parameters/properties supposed constants.

The combined standard measurement uncertainty $u_c$ of a generic quantity $y$ defined by a function $f$ of independent variables $x_i$ affected by uncertainties is defined according (7)

$$u_c^2 \cdot y = \sum_{i=1}^{N} \left( \frac{\partial f}{\partial x_i} \right)^2 u_i^2 \cdot x_i = \sqrt{\sum_{i=1}^{N} \left( \frac{\partial f}{\partial x_i} \right)^2 u_i^2}$$  \hspace{1cm} (7)

Supposing that independent variable are affected by a Gaussian distribution of uncertainties with a coverage factor equal to 2 the relation (8) is used to calculate the expanded measurement uncertainty:

$$U = k \cdot u = \sqrt{\sum_{i=1}^{N} \left( \frac{\partial f}{\partial x_i} \right)^2 k_i \cdot u_i} = \sqrt{\sum_{i=1}^{N} \left( \frac{\partial f}{\partial x_i} \right)^2 k_i \cdot u_i} = \sqrt{\sum_{i=1}^{N} \left( \frac{\partial f}{\partial x_i} \right)^2 k_i \cdot u_i}$$  \hspace{1cm} (8)

In table 1 some results concerning the uncertainty analysis for vertical forces are shown: in particular results shown in table 1 are calculated considering a braking maneuver corresponding to a deceleration of about 1-1.2 m/s$^2$ and consequently a tangential effort on each axle of about 10-15KN.

In fact the measurement of longitudinal forces is affected by errors which are strongly dependent on the measurand value and in particular are functions of the ratio $f_{impugnata}$ (9) defined as the ratio between $X$ and $Q$ forces exchanged between rolling surfaces of wheel and rail.

$$f_{impugnata} = \frac{X}{Q}$$  \hspace{1cm} (9)

Dimensionally $f_{impugnata}$ represents an adhesion factor; from a physical point of view neglecting the contribution of rotating inertias of the vehicle $f_{impugnata}$ is the minimum value of the wheel-rail friction factor that assure the transmission of the tangential force $X$ and it’s a good approximation of modulus of the carbody acceleration during braking/traction maneuvers.

As a consequence the relative precision of the $X$ measurements rapidly decreases considering degraded adhesion conditions or more generally conditions in which small longitudinal efforts are exchanged as visible in figure 6. Unacceptable precision performances for low values of $f_{impugnata}$ are due to the fact the torsional stiffness of the axle is quite high compared to the applied torques and consequently to the measurand $X$. In order to measure low values of longitudinal forces for the purpose of degraded adhesion tests, brakes or the auxiliary shaft of the scheme proposed in figure 2 would be instrumented . In this way it’s possible to estimate accurately total force $X$ exchanged between both wheels and rails. For low adhesion test which are usually performed on straight line this kind of results/measurements of the total $X$ force are valuable also considering that current testing/simulation activities of on board equipments like WSP or odometry systems [9] are mainly related to planar vehicle and line models.
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Table 1: Estimated uncertainty of X estimation/measurement during a braking maneuver

<table>
<thead>
<tr>
<th>Symbol</th>
<th>UNIT</th>
<th>Value</th>
<th>uncertainty (abs)</th>
<th>uncertainty %</th>
<th>Derivative</th>
<th>( \left( \frac{\partial y}{\partial x_i} \right) \cdot U_i )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shaft external diameter</td>
<td>( D_e )</td>
<td>mm</td>
<td>165.00</td>
<td>0.5</td>
<td>0.3%</td>
<td>-281</td>
</tr>
<tr>
<td>Shaft internal diameter</td>
<td>( D_i )</td>
<td>mm</td>
<td>65.000</td>
<td>1</td>
<td>1.5%</td>
<td>22.7</td>
</tr>
<tr>
<td>Elastic modulus</td>
<td>( E )</td>
<td>MPa</td>
<td>206000</td>
<td>6000</td>
<td>2.9%</td>
<td>517315</td>
</tr>
<tr>
<td>Shear modulus</td>
<td>( G )</td>
<td>MPa</td>
<td>79845</td>
<td>2326</td>
<td>2.9%</td>
<td>-0.187</td>
</tr>
<tr>
<td>Strain gauge deformation</td>
<td>( E )</td>
<td>mm/m</td>
<td>56.6</td>
<td>2</td>
<td>3.5%</td>
<td>-0.187</td>
</tr>
<tr>
<td>Wheel radius</td>
<td>( R )</td>
<td>mm</td>
<td>520.00</td>
<td>0.75</td>
<td>0.1%</td>
<td>732E-10</td>
</tr>
<tr>
<td>X-force ( X )</td>
<td></td>
<td>N</td>
<td>14977</td>
<td>700</td>
<td>4.7%</td>
<td>700</td>
</tr>
</tbody>
</table>

Table 1: Estimated uncertainty of X estimation/measurement during a braking maneuver

Figure 6: relative uncertainty on X measurement as a function of \( f_{impegnata} \)

For the measurement of vertical loads \( Q \) the same procedure is adopted, in table 2 some results of the analysis are shown: results of table 2 are referred to a configuration in which is supposed a straight line and a coasting maneuver, this involves an equal load distribution between wheels and the absence of applied lateral and longitudinal efforts \( X,Y \).

Table 2: Estimated uncertainty of Q estimation/measurement in a straight line/coasting configuration

Tests are repeated considering different loads and positioning of the applied forces on wheel profiles corresponding to different line design and maneuvers.
In order to obtain more realistic testing conditions loads are obtained from multibody simulations performed with Intec Simpack™.

In particular sensitivity analysis is repeated considering a symmetrical vertical load discharge factor $\Delta Q_r$, defined according (11).

$$
\Delta Q_r = \Delta Q_{r_i} = \frac{\Delta Q_i}{Q_i} = -\Delta Q_{r_2} = \frac{\Delta Q_2}{Q_2}
$$

$\Delta Q_i =$ absolute vertical load variation on the $i$-th wheel

$Q_i =$ vertical load on the $i$-th wheel

In figure 7 the graphical behavior of the maximum estimated uncertainty between the two wheels, as a function of $\Delta Q_i$, is shown: it’s clear noticeable that for higher values of load transfer between wheels the expected error increase.

![Figure 7: relative uncertainty on Q measurement as a function of $\Delta Q_i$](image)

### 6 UNCERTAINTY ANALYSIS FOR LATERAL FORCES $Y$

For the measurement of lateral forces $Y$ it’s not possible to use an analytical relation between the independent quantities and the measured ones. As a consequence, the sensitivity/uncertainty analysis is performed by evaluating on a complete FEM model of the axle and bogie developed in MSC Nastran-Patran™.

In using the FEM model is performed and error sensitivity analysis considering the following uncertainties:

- Linear and Angular error of strain gage positioning
- Application of spurious forces different from the measurand $Y$

Since the strain measurements are defined only on FEM model nodes a smart technique to obtain a smart interpolation of calculated stress/strain along the wheel surface has to be used.

Considering a polar reference system centered on the rotation axis of the wheel, results have to be interpolated over a grid of 936 different nodes corresponding to 26 different radial distance $r$ values and 36 different angular positions $q$, as visible in the simplified scheme of figure 8/a. Also radius $r_1$ and $r_2$, corresponding to the nominal positioning of strain gauges, are shown.

In this work two different interpolation techniques have been used:

- Standard Triangular Interpolation: the generic properties $y_p$ calculates for a point $p$ is calculated as the weighted sum of the corresponding calculated values $y_{p1}, y_{p2}, y_{p3}$ on the three nearest nodes $p_1, p_2, p_3$ as visible in the simplified scheme of figure 8/c.

- IDWI (Inverse Distance Weighted interpolation) defined according [10]: the interpolation is performed on a subset of the complete population of nodes, with a weighting function which is inversely proportional to the squared distance between the interpolation point $p$ and the corresponding node $p_i$. The subset population is chosen among the $n$ nearest nodes respect to $p$ where $n$ is the size of the chosen subset population. In figure 8/b it’s shown a diagram which demonstrates how the IDWI interpolation gradually converge to a very high precision increasing the size $n$ of the population on which is interpolated the solution.

After several test IDWI was preferred since produce smoother results on the polar grid used for the FEM model of the bogie.
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Figure 8/a: Finite Element model of the bogie and node discretization on the wheel

Figure 8/b: radial deformation on a circumference (r=0.403m)

Figure 8/c: Standard Triangular Interpolation, simplified principle

In table 3/a/b some results concerning sensitivity to alignment and positioning errors of strain gage are shown; these results are referred to a reference loading condition described in table 3/a/b on a single patch corresponding to the contact point of a centered axle with an ISO ORE S1002 wheel profile and a UIC 60 rail (1/40). Since force is measured by an array of strain gauges and positioning errors can be applied in different way the corresponding uncertainties in table 3/b are shown in terms of statistical quantities like minimum, maximum and mean value.

<table>
<thead>
<tr>
<th>Positioning Error</th>
<th>Value</th>
<th>Load Component</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radial Error ( \Delta r )</td>
<td>±1 mm</td>
<td>X</td>
<td>22500 N</td>
</tr>
<tr>
<td>Tang. Error ( \Delta \theta )</td>
<td>±1°</td>
<td>Y</td>
<td>30000 N</td>
</tr>
<tr>
<td>Alignment ( \Delta \alpha )</td>
<td>±2°</td>
<td>Q</td>
<td>55000 N</td>
</tr>
</tbody>
</table>

Table 3/a: positioning errors of strain gauges and loading conditions

<table>
<thead>
<tr>
<th>Positioning Error</th>
<th>Corresponding Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radial Error ( \Delta r )</td>
<td>Max 1.02%</td>
</tr>
<tr>
<td>Tang. Error ( \Delta \theta )</td>
<td>Max 0.88%</td>
</tr>
<tr>
<td>Alignment ( \Delta \alpha )</td>
<td>Max 0.60%</td>
</tr>
<tr>
<td>Total</td>
<td>Max 2.54%</td>
</tr>
</tbody>
</table>

Table 3/b: uncertainty on lateral force measurements introduced by errors on strain gauges positioning

The same uncertainty measurement is performed considering measurement uncertainty introduced by different loading conditions and positioning of the contact point always supposed single.
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In particular in table 4/a some data concerning different loading conditions supposed for uncertainty analysis are shown; corresponding results are visible in table 4/b.

<table>
<thead>
<tr>
<th>Contact Point</th>
<th>N.1 (111%)</th>
<th>N.2 (113%)</th>
<th>N.3 (111%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>N.1</td>
<td>30000N</td>
<td>30000N</td>
<td>30000N</td>
</tr>
<tr>
<td>N.2</td>
<td>55000N</td>
<td>75000N</td>
<td>55000N</td>
</tr>
<tr>
<td>N.3</td>
<td>22500N</td>
<td>22500N</td>
<td>100N</td>
</tr>
</tbody>
</table>

Table 4/b: Relative uncertainty of Y measurement against different loading configurations and contact positions

7 PRELIMINARY EVALUATION OF THE FREQUENCY RESPONSE

Using the FEM model of the bogie it’s possible to evaluate the transfer functions between contact forces and the corresponding measurements. In particular forces are applied in contact point corresponding to the n.2 of table 4/a. In particular for both Y and Q measurements the corresponding $G_Y(\omega)$ and $G_Q(\omega)$ transfer functions are calculated as visible in (12) and (13):

$$G_Y(\omega) = \frac{f_Y(\omega)}{Y(\omega)}$$ (12)

$$G_Q(\omega) = \frac{f_Q(\omega)}{Q(\omega)}$$ (13)

Defining $G_Y(0)$ and $G_Q(0)$, the static values of $G_Y(\omega)$ and $G_Q(\omega)$ it’s possible to define the normalized transfer functions $P_Y(\omega)$ and $P_Q(\omega)$ defined as (14),(15):

$$P_Y(\omega) = \frac{G_Y(\omega)}{G_Y(0)}$$ (14)

$$P_Q(\omega) = \frac{G_Q(\omega)}{G_Q(0)}$$ (15)

In figures 9/a/b the calculated behavior of $P_Y(\omega)$ and $P_Q(\omega)$ is shown: it’s interesting to notice that the bandwidth of $P_Y(\omega)$ is strongly influenced by the a mode visible in figure 9/c which corresponds to a rolling motion of axles at about 22.5Hz. In the finite element model the mechanical impedance of the rail is modeled using an array of lumped elasto-viscous elements.

The equivalent stiffness of the rail deeply affects the frequency positioning of the structural mode visible in figure 9/c demonstrating an important cross coupling between line and vehicle modal behavior which may heavily influence the real bandwidth of the measurement system.

Obviously in a real applications deriving from electrical noise and other limitation introduced by signal transmission and amplification have to be evaluated.
CONCLUSIONS AND FUTURE DEVELOPMENTS

In this work the sensitivity of contact force measurements against several uncertainty factors. Among the various factors investigated the position of the contact point seems to be one of the most important. Also from the calculation of the frequency response of the system it’s clearly recognizable a strong coupling/interaction between the flexible behavior of rails and vehicle. As consequence the real bandwidth of the contact force measurement system has to be verified taking count of the dynamical response of both systems. Research/study activities will be further extended in order to introduce in the FEM model of the system a more accurate model of the rail flexible behavior.

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