SIMULATION OF WHEEL–RAIL CONTACT CONDITIONS ON EXPERIMENTAL EQUIPMENT

Abstract. The paper is focused on equipment used in experimental research in the field of rail vehicles. Such equipment often replace the wheel or the rail (or both) with substitutive bodies, e.g. discs of different dimensions and profiles. The main aim of this study is to present an analysis of contact conditions of such bodies with the purpose to reduce (or at least identify) the differences from the conditions of actual railway operation. Firstly, an overview of properties of experimental equipment is given, together with theoretical basis of the most important differences with the use of Hertzian contact for comparison. This is followed by analysis of three selected situations encountered in research work of the author; these include substitution of straight rail with a roller (rotating rail), influence of pressure between contacting bodies upon coefficient of friction and the problem of inducing full sliding with respect to the torque of the driving motors. In conclusion, it is stated that selected quantities may be kept at the values typical for real operation, but not all of them at the same time. It appears particularly suitable to maintain the correct value of pressure (normal stress) in the contact area because of material loading, frictional conditions as well as slope of adhesion characteristics.
Keywords: Adhesion, friction, rotating rail, test equipment, wheel–rail contact

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Introduction

Railway research has made a significant progress since the origins of this mode of transport, and has provided detailed theories of physical phenomena important for motion of railway vehicles. Relevant calculations and predictions may be made on the basis of these theories, particularly now that high-performance computational equipment is available. Nothing of that, however, reduces the importance of technical experiment which is still an inseparable part of solution of current problems not only in rail vehicles.

This paper is focused on equipment for experimental work connected with wheel–rail contact mechanics. According to the relationship of the experimental device to real operation, four cases may be distinguished (see Fig. 1):

1) **real vehicle, real track**: in this case, experiments are performed with a real vehicle in operation or on a test track;
2) **real vehicle, track substituted**: this is a quite demanding possibility of testing whole vehicles on large roller rigs where each wheel is supported by a roller (rotating rail);
3) **real track, vehicle substituted**: various small vehicles, rollers, tribometers pushed by hand or borne by other vehicles on railway tracks;
4) **vehicle and track substituted**: this includes a great variety of test rigs of different mechanical structure and scale.

If any object is substituted with a model, the experimental device is equipped:

- **instead of the vehicle**: by a part of the running gear – bogie, wheelset, assembly of the wheel with primary suspension or, which is common, a wheel alone;
- **instead of the track**: by a roller (rotating rail) for each wheel; or a straight rail segment which makes repeated linear movments.

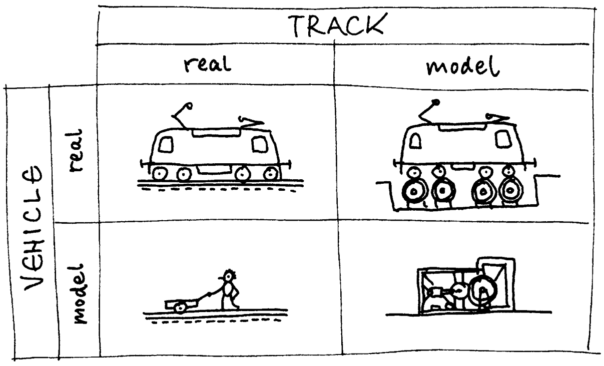


Fig. 1 Illustration to the possibilities of experimental equipment in the field of rail vehicles

Overview of selected designs and examples of experimental devices all over the world, together with some theoretical considerations, may be found e.g. in (Jaschinski 1999), (Iwnicki 2006), (Kalivoda 2014). For Czech and Slovak research institutes, we may mention the roller rig of the CTU in Prague (Kalivoda 2011), Rail Wheel Test Stand of the University of Pardubice (Culek 2015) and the RAILBCOT machine of the University of Žilina (Gerlici 2014).

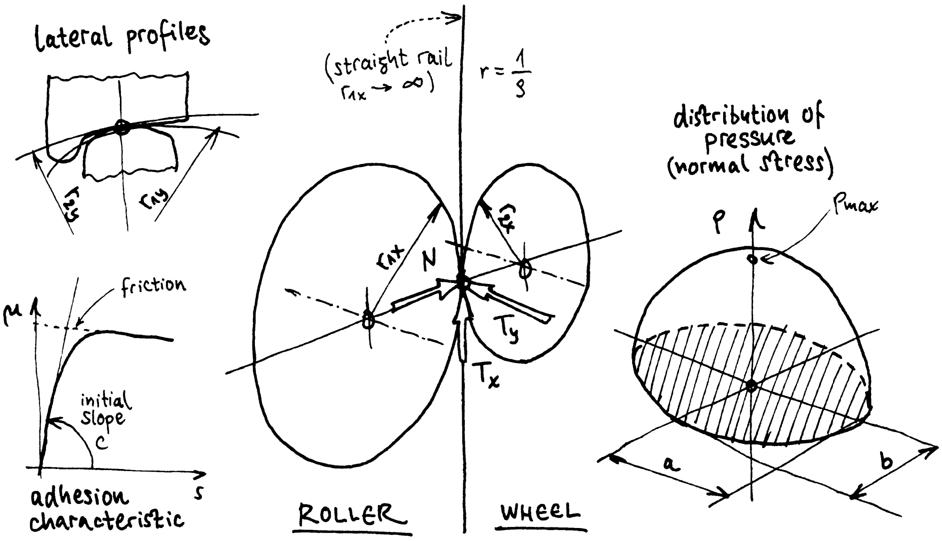


Fig. 2 A sketch to the contact of wheel with a (rotating) rail

Design of roller rigs might be the subject of extensive studies, as well as the issues of model similitude, see e.g. (Čáp 1997); investigation of running behaviour on roller rigs was dealt with in (Kalivoda 2013). The focus of the present paper is, however, aimed directly at the contact of wheel and rail or the bodies which represent them in the experimental device. The use of substitutive bodies brings about a change of contact conditions. Careful examination of these, based on rolling contact theory – see e.g. (Kalker 1990), (Polách 2005) – helps to answer essential questions such as:

- what is the nature and extent of the effect of differences between the experimental device and real operation?
- what design or setup of the experimental device might minimize the effect of the differences?

1. Differences of experimental equipment from real operation

1.1. Source of the differences

Observing the parameters which have influence on wheel–rail contact conditions (illustrated in Fig. 2), the following appear to act as main factors:

1) **Geometry** of the bodies in contact; we may furthermore distinguish:
   a) **longitudinal geometry**, consisting in the diameter of the wheels: the experimental device is often down-sized to reduce space and material requirements – this significantly reduces expenses particularly if frequent replacement of test specimens (discs, segments) is necessary;
   b) **lateral geometry**: the wheel and rail profiles may be identical to real operation but do not have to, esp. again in small-scale devices.
2) **Contact forces** including the normal force N, longitudinal force T_x and lateral force T_y. To reduce demands on construction and operation of the experimental equipment, loading forces may be lower than in railway operation.
3) **Mobility**, by which we mean ranges of rolling velocity and of lateral and longitudinal creep velocities. For instance, if no lateral movement or angle of attack is possible, lateral creepage may not be induced.
4) **Frictional conditions**, which means possibility of creating various conditions of surfaces in contact (roughness, supply of contaminants) and control of environmental properties which affect them (humidity, temperature).

1.2. Effects of the differences

The factors listed above include contact geometry and compressing force, hence differences in size and shape of the contact area will generally be present at the experimental device. For the purposes of comparison, the Hertz theory will be used here, to which a brief explanation is given e.g. in (Iwnicki 2006) and (Čáp 1999). In Hertzian contact, the length of contact ellipse semiaxes is

$$a, b = const. \cdot \sqrt[3]{N/\rho} = const. \cdot \sqrt[3]{N/\left(\rho_{1x} + \rho_{1y} + \rho_{2x} + \rho_{2y}\right)} \quad (1)$$

i.e. length of the semiaxes is proportional to cubic root of the ratio of normal force N to combined curvature ρ which is the sum of principal curvatures of both bodies in both directions. This shows that suitable choice of curvatures of the substitutive bodies may theoretically provide the required contact area size even for different (lower) contact force.

Similarly, the ratio of the semiaxes might be preserved, too, as

$$\frac{a}{b} = \mathrm{f}\left(\frac{\rho_x}{\rho_y}\right) = \mathrm{f}\left(\frac{\rho_{1x} + \rho_{2x}}{\rho_{1y} + \rho_{2y}}\right). \tag{2}$$

Furthermore, the formula for maximum Hertzian pressure $p_{\max}$ may be considered, and rewritten in the form

$$p_{\max} = const. \cdot \mathrm{f}(a/b) \cdot \sqrt[3]{N \cdot \rho^2} \ , \tag{3}$$

which is different from (1). Therefore, if limited compressive force is available for the experimental device, choice of curvatures may still allow maintaining true contact size or contact pressure but not both at the same time. Preservation of contact pressure typical for real operation may be, without doubt, regarded as more important since it is a measure of loading to which the material is subjected – and whose high value is typical for the wheel–rail contact. The experimental device may then provide the same contact pressure as in real operation, however on smaller area.

The quantities a, b and $p_{\max}$ also appear in formulae related to calculation of tangential forces. Specifically, the initial slope of adhesion characteristic $\mu = \mathrm{f}(s)$ is

$$c = \left.\frac{\mathrm{d}\mu}{\mathrm{d}s}\right|_{s=0} = const. \cdot \frac{C_{ij}}{p_{\max}} \ , \tag{4}$$

where C_{ij} is Kalker's coefficient for the given creepage direction (C_{11} longitudinally, C_{22} laterally). The value of the coefficient depends on the a/b ratio – see e.g. (Iwnicki 2006), (Kalker 1990) – but the sensitivity is not very high. Therefore the contact size is not of primary importance here but contact pressure has got a major influence. For devices with low compressive force, higher slope of adhesion characteristic may be expected; its peak moves to lower values of relative creepage.

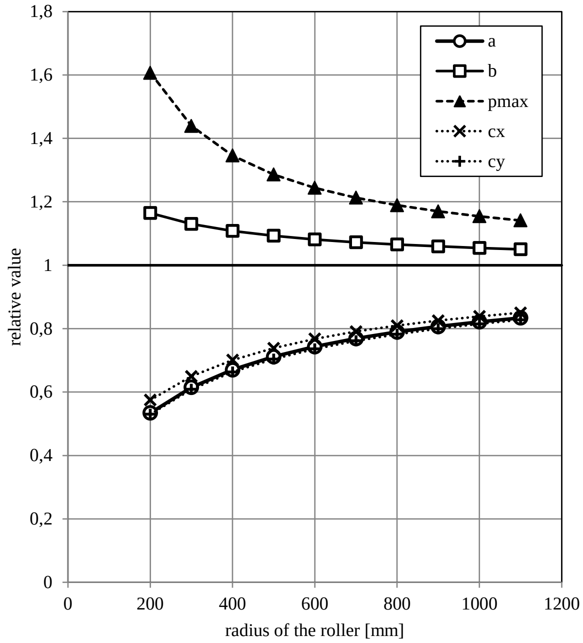
The contact conditions are also constituted by other factors whose theoretical description is not so trivial, namely the phenomenon of friction and the effect of the state of contacting surfaces (rougness, third-body layer). This also changes the conditions of transmission of forces described by the adhesion characteristic.

As a result, at any rate, the adhesion characteristic changes. The following sections describe selected analyses of observed effects which we encountered when dealing with tasks of applied research.

2. Analysis of selected cases

2.1. Substitution of a linear rail with a roller

The substitution of a straight rail with a roller (rotating rail) constitutes change of contact geometry in the longitudinal direction. The change is more significant for smaller roller radius. The top plot in Fig. 3 shows quantitative representation of this effect for a wheel with 920 mm diameter, –450 mm (concave) lateral radius, and a rail/roller with 300 mm lateral radius. This contact geometry is close to conditions of the S1002/60E1 1:40 contact in centered position, which is actually non-Hertzian, but Hertz theory is used here for comparability. The material parameters are E = 210 GPa, ν = 0.3, normal force is constant 100 kN. The vertical axis shows the length of the semiaxes a, b, maximum Hertzian pressure $p_{\max}$ and initial slopes of longitudinal and lateral adhesion characteristics c_x, c_y in relative values with respect to these valid for a linear rail.



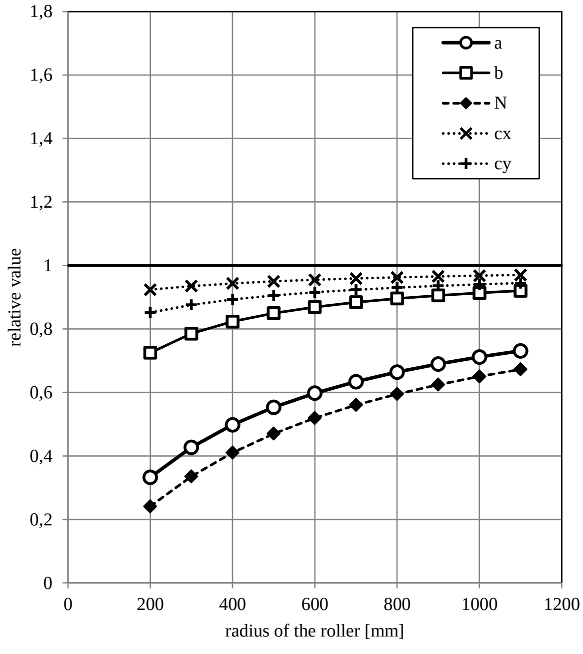


Fig.3 Influence of the roller radius on size of the contact ellipse, contact loading and slopes of adhesion characteristics – top: for N = *const*; bottom: for $p_{\max}$ = *const*.

It is seen that, by influence of the roller curvature, the contact area becomes shorter, somewhat wider, and contact pressure increases (by ca. 25 % for 600 mm roller radius). At the same time, initial slope of adhesion characteristics decreases, as explained by theory in the section 1.2. Better agreement with the conditions of straight rail may be reached by increasing the dimensions of the roller; however for radius above ca. 0.6 m, a practical advantage, consisting in the possibility of manufacturing the roller by turning a railway wheel, is lost.

Seeing that the usage of a roller leads to increase in pressure, one might consider decreasing the normal force. The bottom plot in Fig. 3 shows the situation where normal force is adjusted to get constant Hertzian pressure. This has a negative influence on preserving the contact ellipse size (the contact area especially becomes shorter), however slope of the adhesion characteristics is much closer to that for straight rail. If, for instance, a roller of 600 mm diameter is pressed towards the real wheel by a force of 60 kN, similar material loading and similar slope of adhesion characteristic is attained as for real vehicle on linear rail with 100 kN wheel force. This is advantageous also with respect to forces acting on the components of the experimental equipment.

2.2. Influence of normal force on coefficient of friction

Coefficient of friction (COF) in wheel–rail interaction models is often considered constant, or exponentially decreasing in dependence on creep velocity (Polách 2005). Dependence of COF on normal force is not included in the theories. Actually, its absence is assumed – this is why coefficient of friction is a coefficient, a constant value by which the normal force is multiplied. Experiments however show that some influence of normal force or pressure on COF (or available adhesion coefficient) does exist.

This trend is shown e.g. in the standard EN 14363 in Fig. 4. This plot is based on lateral adhesion characteristics measured at a test rig in Minden in the 1960s (see also leaflet UIC 510-2). The highest adhesion characteristic belongs to the lowest normal force. In order to get more information about this effect, a study of results of adhesion measurements published in 16 different sources was made, and included in a research report (Voltr 2013) to the project „Technology for measurement of force effects in the wheel–rail contact“. Conclusions of the study are briefly described here.

The studied publications generally indicated that some dependence of COF on loading was recorded (even if it was not the purpose of the work to find it). In order to make a summary, a plot in Fig. 4 was compiled. It is given without the key here; each line or cluster of points stands for one publication or set of measurements. It should be noted that the plot contains results of many experiments under various conditions and that it is inaccurate, e.g. makes no distinction between coefficient of friction and maximum coefficient of adhesion. It is rather intended to give a complex information about

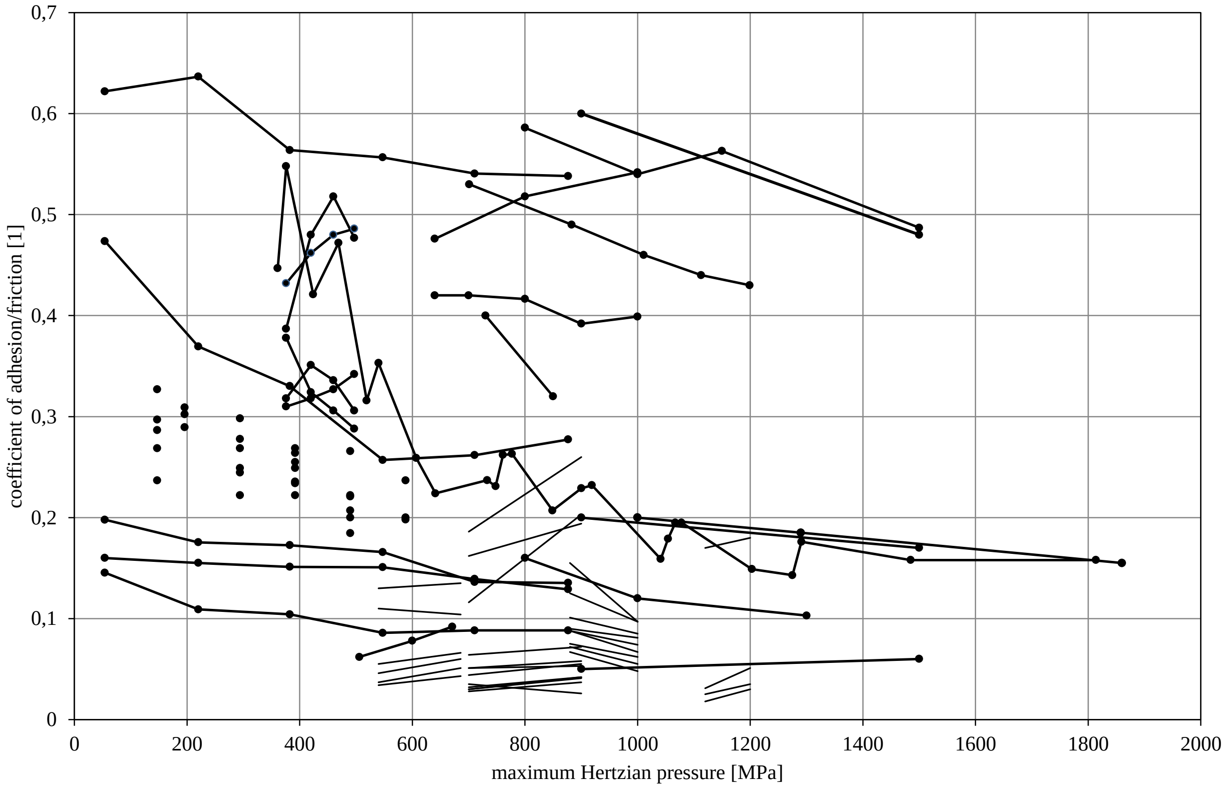


Fig. 4 Trends of dependence of adhesion/friction coefficient on pressure in the contact area from experimental results published in literature

explored regions and recorded trends in pressure–friction dependence.

At a glance, most lines confirm the abovementioned decreasing trend. Taking account of the experimental conditions documented in the published sources, we conclude that:

- for **dry conditions**, COF generally slightly decreases with increasing pressure,
- for **contamination by oil**, by contrast, it increases,
- for **wet surfaces**, it is something in between, and
- if **HPF modifier** is used, the results are similar for dry surfaces.

If one takes the liberty of quantifying such inhomogeneous and inaccurate data, the following indicator may be used to that end:

$$\varphi = \frac{\mathrm{d}f}{\mathrm{d}p_{\max}} \approx \frac{\Delta f}{\Delta p_{\max}}\,, \tag{5}$$

which is a slope of linearized dependence of COF f on maximum Hertzian pressure $p_{\max}$ in the contact. Calculated values of φ seldom exceed, in absolute value, the amount of $2.0 \cdot 10^{10}\ \mathrm{Pa}^{-1}$. This value indicates that, for instance, when contact pressure changes by 100 MPa, the coefficient of friction changes by 0.02. Direction of the change is indicated by the list above.

The conclusion is that one cannot reject existence of the coefficient of friction on contact pressure. If an experimental device maintains unreduced level of contact pressure between the bodies representing wheel and rail, this influence is eliminated.

2.3. Possibility of inducing full sliding

We occasionally encounter questions like „How many percent slip can this machine do?“ or „What slip should be set to represent real operation?“ In answer to this, it must be noted that slip (creepage) is the primary controlled and limiting quantity for devices, where slip is determined by

- **angle of attack** (devices utilizing the principle of lateral slip), or
- **gearing of the machine** (Amsler-type devices),

but otherwise the operating mode is principally limited by the tangential force that can be attained by the driving system ($T_{\max}$, corresponding motor torque $M_{\max}$). The same is limiting for a locomotive in operation. We may define the index

$$u = \frac{T_{\max}}{f \cdot N} = \frac{M_{\max}}{r_{\mathrm{k}} \cdot f \cdot N}\,, \tag{6}$$

where r_{k} is the wheel radius, f is coefficient of friction. If $u > 1$, full sliding can be achieved, thus any value of slip may occur. Otherwise the experimental device is limited to the microslip region, i.e. no more than several per cent slip will be induced. Operational measures to remove this limitation include lubrication of the contact (reducing f) and decreasing the normal force (reducing N, i.e. also the contact pressure). The requirements on slip, force loading and range of frictional conditions are, in this respect, opposing.

Conclusions

1. By suitable choice of parameters of experimental equipment which substitutes wheel and/or rail by different bodies, one might maintain contact conditions comparable to those of real operation but not all at the same time.
2. It appears particularly suitable to set the conditions in such way that the experimental device produces the same level of contact pressure as in operation, since
   - pressure is the principal measure of material loading,
   - pressure has a direct influence on the slope of the adhesion characteristic (see section 1.2, 2.1),
   - pressure can also influence the coefficient of friction (section 2.2).
3. If the substitutive bodies of the experimental device are of greater curvature than real wheel and rail, normal force may be decreased. This effect is not related to small-scale equipment only – it is noticeable also for replacement of the linear rail with a roller at a full-scale roller rig. The decreasing of the compressive force is not only to reduce demands on the roller rig structure but may be really recommended, based on the above explanation.

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References

Culek, B. et al. 2015 Rozvoj technické infrastruktury Dopravní fakulty Jana Pernera, Univerzity Pardubice v oblasti kolejové dopravy /Development of technical infrastructure of the Jan Perner Transport Faculty, University of Pardubice, in the field of rail transport/. *In: Vědeckotechnický sborník ČD, 39/2015.*

Čáp, J. 1997 Možnosti podobnostní metody studia adheze /Possibilities of the similitude method in the study of adhesion/. *In: Scientific Papers of the University of Pardubice, Series B, 3 (1997), p. 105–112.*

Čáp, J. 1999 Teoretický rozbor tečných a interaktivních sil ve styku kolo–kolejnice /Theoretical analysis of tangential and interactive forces in the wheel–rail contact/. *In: Scientific Papers of the University of Pardubice, Series B, 5 (1999), p. 29–47.*

Gerlici, J. et al. 2014 Rail vehicles wheels and brake blocks wear laboratory test stand utilization. *In: Prace naukowe Politechniki Warszawskiej, z. 101, 2014, p. 21–32.*

Iwnicki, S. (ed.) 2006 Handbook of Railway Vehicle Dynamics. 1st edition. CRC/Taylor&Francis. ISBN 978-0849333217.

Jaschinski, A. et al. 1999 The Application of Roller Rigs to Railway Vehicle Dynamics. *In: Vehicle System Dynamics, 31 (1999), p. 345–392.*

Kalivoda, J., Bauer, P. 2011 Railway bogie with active wheelset guidance – computer simulations and roller rig tests. *In: Proceedings of the 22nd International Symposium on Dynamics of Vehicles on Roads and Tracks, 14–19 August 2011, Manchester.*

Kalivoda, J., Bauer, P. 2013 Experimentální výzkum jízdních vlastností kolejových vozidel na kladkových stavech /Experimental investigation of rail vehicle behaviour on roller rigs/. *In: Současné problémy v kolejových vozidlech /Current Problems in Rail Vehicles/, 18.–20.9.2013 Česká Třebová, Sborník příspěvků, p. 49–60.*

Kalker, J. J. 1990 Three-Dimensional Elastic Bodies in Rolling Contact. Kluwer Academic Publishers. ISBN 0-7923-0712-7.

Polách, O. 2005 Creep forces in simulations of traction vehicles running on adhesion limit. *In: Wear 258 (2005), p. 992–1000.*

Voltr, P. 2013 Teoretické řešení problematiky adhezních a třecích poměrů v kontaktu kolo–kolejnice /Theoretical solution of the problems of frictional conditions in the wheel–rail contact/. Report no. 01/13 – DFJP, UPa. Project TA02030776 „Technology for measurement of force effects in the wheel–rail contact“. Pardubice: University of Pardubice.