5th ADAMS/Rail Users' Conference
Haarlem, The Netherlands – May 10th –11th , 2000

Simulation of a scaled roller rig

Bosso N., Gugliotta A., Napoli E., Somà A.

Dipartimento di Meccanica - Politecnico di Torino
Cso. Duca degli Abruzzi 24
10129 Torino Italy
soma@polito.it

Introduction

The railway development has required tests on static vehicles. For this purpose test unit called Roller Rig are used. The Roller Rig is a testing ground (test stand) where the motion of the wheels on the track is simulated by the rolling on the rollers. The first system of this kind was designed by Carter in order to perform tests on locomotives (England, 1920). Roller Rigs are rather expensive systems, because they are provided with a high power electrical engine to give motion to the rollers. Also the control and measurement system is particularly sophisticated. The Deutsche Bahn, the U.S. Department of Transportation (United States, 1978), the Railway Technical Research Institut (Japan, 1989), the Chinese Railway (China, Chengdu) use this test rig. Tests on roller rigs are independent on external climatic conditions and so they are suitable for fatigue tests. For economic reasons, most of roller rigs are designed for tests on scale models. Scaled roller rigs are especially realised in universities and research institutes. They are advantageous because their realisation costs are not very high and their dimensions are quite small. In general terms, all tests made on a bench are influenced by differences of the real behaviour. First of all, the equilibrium of a vehicle on the rail is indifferent stable, whereas on a roller rig the equilibrium is unstable. Also on a roller rig the equilibrium can be stable with constant speed. A anti – drag's system is used to change speed of the test and to support the constant position of the model on roller rig.

Moreover whether on a case of wheel on roller or of wheel on rail, the contact between the two bodies is represented by an area with elliptical form and her dimensions are known by principal bending radius (curvature) around the contact point and around the relative deflection of bodies. The difference between a roller and a rail is the modification of a principal deflection, with the following change of elliptical form. This contact area's variation changes the vehicle's behaviour because in this region are applied the contact forces. It needs to introduce opportune scaling factors for each physical size and fix the phenomenon of the study. The mechanical similarity that is our ground of studying has been treated by several authors. All have expressed various considerations and after to have built the scaling laws, have designed a roller rig. In the present work the three principal scaling techniques are compared, suggested by Pascal, Jaschinski and Iwnicki. Shortly it's only explained the several hypothesis for scaling factors determination.

Pascal's method

For practical and economic reasons it's suitable to built wheel and roller with the same materials used for the wheel/rail system. The scaling factors of Yang's and density's modulus are unitary. The scaling factors of other physical sizes can be obtained by using simple formulas.. Between the physical size the accelerations have a scaling factor equal to the inverse of the scaling length. Also the gravity must be changed on the model. In order to obtain this result the contact force between wheel and rail is increased, by simulating a body of a bigger weight but equal mass, by applying a normal force.

Iwnicki's method

In the building of similarity laws, apart from dimensional scaling factor, various scaling factors can be chosen. For these laws, Iwnicki adopts a scaling factor for the time equal to one and that the building material of wheel and roller is the same for the systems on similarity. This choice introduces a scaling factor on the gravity, but whereas on the described laws, the contact force between wheel and roller increases (bigger weight on equal mass for the gravity increasing), in this case it decreases.

In order to increase the contact force, it's applied a directed force as the gravity on the wheels, whereas the cables is used in order to decrease the gravity acceleration. It's intuitive that the gravity modification influences the dynamic behaviour, but it's negligible if the lateral displacements of the vehicle are small.

Jaschinski's method

In accordance with Reynolds, the similarity studies are corrected if the exact scaling laws is taken into account for all physical sizes. Jaschinski, in accordance with this principle, considers the simplified differential equations that they describe the wheelset dynamic and the contact normal force between wheel and rail. In this way several no – dimensional groups to extract the scaling factors is obtained.. The acceleration's scaling factor is unitary. The creep force's scaling factor can be obtained whether by the motion equations or not by constraint equations, but the value should be the same. Comparing the expression of the creep force's scaling factor, a scaling factor of material density equal to inverse of the length's scaling factor is obtained. It's very difficult to build a model respecting this law. So Jaschinski has suggested two scaling methods, one that allows the dynamic study and the other method for study of contact between wheel and rail.

Methodology of "Jaschinski modified"

The objective of the simulations is to value the realisation possibility of a system for the bogie's test on a curve with constant radius. The scaling laws, that introduce a reduction factor on the gravity, are unsuitable for this aim. The Jaschinski's methodology doesn't allow the simultaneous studying of the contact and dynamic behaviour. In order to eliminate this limit, it's introduced a modification on the Jaschinski's scaling laws. So it's introduced a scaling factor on the material's elastic module.

The principal scaling factor of used physical sizes can be resumed as follows in table I :

Scaling factors	Jaschinski	Pascal	Iwnicki	Jascinski modified
φ_l length	5	5	5	5
φ_t time	$\sqrt{5}$	5	1	$\sqrt{5}$
φ_V velocity	$\sqrt{5}$	1	5	$\sqrt{5}$
φ_a acceleration	1	1/5	5	1
φ_m mass	125	125	125	75
φ_F force	125	25	625	75
φ_ρ density	1	1	1	0.6
φ_E elastic module	1	1	1	3
φ_w weight	125	125	125	75
φ_C stiffness	25	5	125	15
φ_T creep force	125	25	625	75
φ_K damper	$25\sqrt{5}$	25	125	$15\sqrt{5}$
φ_{Ct} torsion stiffness	625	125	3125	375
φ_{Kt} torsion damper	$625\sqrt{5}$	625	3125	$375\sqrt{5}$
φ_I inertia	3125	3125	3125	1875
φ_μ friction	1	1	1	1

Tab 1 *list of scaling factors*

Simulation models

For simulations purpose two models has been used. A first model has been realised in order to compare the results obtained with the use of the scaling techniques. A second model has been designed to describe the behaviour of curving vehicle on a roller rig. All the built models refer to a passenger vehicle with two suspension's stages. The primary suspension is made up by a spring damper acting directly on wheelset axle and by a bushing place on a distance b = 0.26 m by the wheelset centre. Moreover there's a damper distant from wheelset centre c = 0.2 m. The secondary suspension, between bogie and the vehicle's carbody, is formed by 4 bushing elements, where the stiffness (x,y,z) and lateral damper (y) are concentrated. In general terms, on the scaled roller rig it is possible to make tests on bogies and to obtain several information about an entire vehicle due to the structural symmetry. Only half car body has been modeled. The middle of the vehicle doesn't have any yaw and pitch rotation. This joint has been described by bushing with only torsion stiffness. The contact forces between wheel and roller, have been obtained by means of general force (Gforce), applied on the centre of each wheel. Each force is described by two tangential components (x,y) and by the normal force (z). For the analytic expression of the creep forces referred to the level 2a present in Adams/rail. Such formulation has a lot of limits (constant velocity and contact parameters, small lateral displacements).

In the first model, the roller rig is only simulated by reducing the Kalker' s coefficients in the analytic expression of the contact forces. Between wheelset and ground is so defined a planar joint. In this way the pitch rotation is prevented. The pitch is connected with the lateral displacement and if the lateral displacement is small, the bogie pitch can be neglected. This model is used in order to compare the scaling technique but it can't give information about the real behaviour of a roller rig. A second model is used to investigate the roller rig' s behaviour and to proceed in its design. The Roller rig is composed by a roller for each wheel. Each roller is connected with a structure by means of a revolute joint. The structure is connected with ground by bushings. The contact forces' expression is modified to consider the non-linear part of the friction force with the expression:

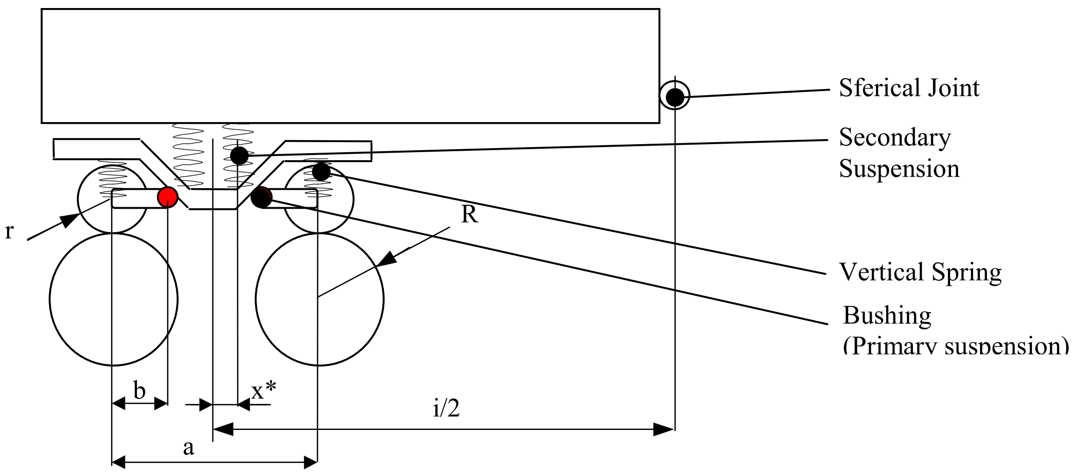
$$F_x = -\frac{f_{11}\mu_x}{\sqrt[n]{1+\left(\frac{f_{11}\mu_x}{fN}\right)^n}} \qquad (1)$$

where f_{11}, is the Kalker coefficient, N normal force, f friction coefficient, μ_x slip quantities.

Wheelbase	A	[m]	2.7
Bogie spacing	I	[m]	19/2
Rolling radius	R	[m]	0.46
Roller radius	R	[m]	0.9
Gauge	S	[m]	0.75
Primary suspension lever length	B	[m]	0.26
Y position axle box. (1)	Y	[m]	1.0
Y position secondary suspension (1)	y*	[m]	0.86
X position secondary suspension (1)	x*	[m]	0.4
Z position body centre mass (2)	ZG*	[m]	0.35
Z position bogie centre mass (2)	ZG+	[m]	2.5

(1) : from bogie centre. (2) : from rail level.

Tab. 2 Geometrical data



fg.1 *scheme of the model*

A longitudinal displacement of vehicle on the roller rig causes a vertical displacement. This fact can determinate the instability. In order to consider this aspect, a kinematic system is used to add this degree of freedom. The kinematics connects the roller centre and the wheel centre and it's composed by two elements without mass: the link and fit. The wheelset is connected with fits by means of an inplane joint and the 2 fits are connected each other by translational joint to allow the wheelset yaw. The links are connected with roller by means of revolute joint. The links are connected with fit by spherical joint useful to test the vehicle's acceleration and curving behaviour. For the Kalker's coefficient calculation is used level 2a of Adams/RAIL and these coefficients has been scaled by using the similarity laws of table 1.

		Scale 1/1	Pascal	Jaschinski	Jaschinski	Iwnicki	Jaschinski modified
Half Wagon Body	M [kg]	18959	151.7	303	151	151	253
	I_{xx} [kgm^2]	25766	8.3	16.5	8.3	8.3	13.74
	I_{yy} [kgm^2]	804283	257	514.7	257	257	428.9
	I_{zz} [kgm^2]	798018	255	510.7	255	255	425.6
Bogie	M [kg]	2313	18.5	37	18.5	18.5	30.84
	I_{xx} [kgm^2]	1551	0.5	1	0.5	0.5	0.83
	I_{yy} [kgm^2]	1655	0.53	1.06	0.53	0.53	0.88
	I_{zz} [kgm^2]	3199	1	2	1	1	1.71
Axle box (4)	M [kg]	71	0.57	1.13	0.57	0.57	0.95
	I_{xx} [kgm^2]	5	1.6E-3	3.1E-3	1.6E-3	1.6E-3	2.6E-3
	I_{yy} [kgm^2]	15	4.7E-3	9.4E-3	4.7E-3	4.7E-3	7.8E-3
	I_{zz} [kgm^2]	15	4.7E-3	9.4E-3	4.7E-3	4.7E-3	7.8E-3
Wheelset (4)	M [kg]	1618	12.95	25.9	12.95	12.95	21.58
	I_{xx} [kgm^2]	753	0.24	0.48	0.24	0.24	0.4
	I_{yy} [kgm^2]	100	0.03	6.4E-2	0.03	0.03	0.054
	I_{zz} [kgm^2]	753	0.24	0.48	0.24	0.24	0.4

Tab. 3 *Inertial data*

		Scale 1/1	*Pascal*	*Jasch ro=0.5*	*Jasch ro=1*	*Iwnicki*	*Jasch - mod*
Primary Suspension							
Primary Suspension Bushing	K_x [kN/m]	6864	1372	549	274.5	54.9	457
	K_y [kN/m]	12748	2550	1019	510	102	849
	K_z [kN/m]	6864	1372	549	274.5	54.9	457
	K_{yy} [Nm/°]	1E11	8E10	3.2E7	16E6	3.2E6	2.7E7
Vertical spring	K_z [kN/m]	6864	454	182	90.8	18.2	151.5
Secondary suspension							
Secondary spring (4 elements)	K_x [kN/m]	61	12.2	4.87	2.4	0.487	4.06
	K_y [kN/m]	61	12.2	4.87	2.4	0.487	4.06
	K_z [kN/m]	231	46.3	18.5	9.2	1.851	15.4
Traction rod	K_x [kN/m]	51012	10202	4080	2020	408	3400
Anti-roll	K_{xx} [Nm/°]	29237	233.9	93.6	46.7	9.36	78
Symmetry connection elements.							
	Kyy[Nm/°]	10E6	8E3	3200	1600	320	2667
	Kzz[Nm/°]	3.1E5	2496	998	500	99.8	832

Tab.4 *Stiffness*

Scaling factors	MMU	Pascal	Jascinski $\rho = 1$	Jascinski $\rho = 0.5$	Jaschinski modified
f_{11}	73	25	125	125/2	40
f_{22}	73	25	125	125/2	40
f_{23}	625	125	625	625/2	80

Tab.5:*scaling factors of Kalker's coefficients*

Comparison between models

The simulations target is a study of the mechanical similarity's influence about the measurable results on a scaling roller rig. The considered parameters are the eigenvalue analysis with a fixed speed and the critical speed (instability threshold). The critical velocities and modal frequencies are compared by multiplying them by the relative scaling factors. On the table 6 are reported obtained values by the simulations with constant speed and equal to v = 20 m/s for the reference's vehicle (scale 1:1).

Scala 1:1	Iwnicki	Jac1	Jac0.5	Pasc	Jascinski Modified	Mode
0.46	0.5	1.23	1.23	2.78	1.22	Body Lateral
0.76	0.76	1.7	1.7	3.79	1.7	Body Yaw
0.99	1.1	2.46	2.46	5.49	2.45	Body Vertical
1.34	1.34	3	3	6.72	3	Body Pitch
1.46	1.48	3.44	3.44	7.75	3.44	Body Roll
2.14	1.92	4.85	4.91	11.1	6.74	Bogie Yaw
10.38	11.17	25.21	25.23	59.97	24.99	Bogie pitch
20.12	20.12	44.98	44.98	100.58	44.97	Bogie Vertical
22.48	23.01	50.41	49.96	112.54	51.97	Bogie Roll

Tab 6: *comparison of eigenfrequency calculated on constant velocity*

The results of the table 6 are not homogenous and therefore are not comparable. It's possible to return the results of table 6 homogenous by multiplying them for the scaling factors of the frequencies. (table 7). The reference values are reported in the first column. In table 8 the differences between the scaling technique is described in terms of perceptual difference.

Table 8 shows that the body yaw and the vertical bogie mode are not influenced by the scaling laws. These frequencies only depend on the mass and on the system's inertia. The positive sign of the per cent error shows an excess assessment whereas the negative sign shows a defect assessment of the eigenfrequency.

Scala 1:1	Iwnicki	Jachinski	Jaschinski $\rho = 0.5$	Pascal	Jascinski Modified	Mode
0.46	0.50	0.55	0.55	0.56	0.55	Body Lateral
0.76	0.76	0.76	0.76	0.76	0.76	Body Yaw
0.99	1.1	1.1	1.1	1.1	1.1	Body Vertical
1.34	1.34	1.34	1.34	1.34	1.34	Body Pitch
1.46	1.48	1.54	1.54	1.55	1.54	Body Roll
2.14	1.92	2.17	2.2	2.21	3.01	Bogie Yaw
10.38	11.17	11.28	11.28	11.99	11.18	Bogie pitch
20.12	20.12	20.12	20.12	20.12	20.11	Bogie Vertical
22.48	23.01	22.55	22.34	22.51	23.24	Bogie Roll

Tab 7: *comparison with principal eigen frequency a railway vehicle*

By table 8 we can conclude that in the majority of the cases the roller rig overestimates the real value of the eigenfrequency. The bogie yaw frequency is influenced by test speed as showed by Klingel's formula, which determines the yaw frequency of a wheelset with rigid wheel and constant conicity:

$$f = \frac{V}{2\pi}\sqrt{\frac{\gamma}{b_0 r}} \qquad (2)$$

where v is the progress velocity of the vehicle, γ the equivalent conicity, r wheel's nominal radius, b_0 the half gauge. This formulation emphasises that the yaw frequency is bound on the kinematic phenomenon of the wheel/roller contact. In order to evaluate the difference a more complex formula, that takes the Kalker's coefficients f_{23}, f_{22} and the gravitational force F_a is used:

$$2\pi f = \frac{1}{2}\left(\frac{f_{23}\varepsilon V}{f_{22} r} - \frac{F_a V \varepsilon}{4 f_{22} s}\right) \pm \sqrt{\frac{1}{4}\left(\frac{f_{23}\varepsilon V}{f_{22} r} - \frac{F_a V \varepsilon}{4 f_{22} s}\right)^2 + \frac{\gamma V^2}{rs}} \qquad (3)$$

The scaling factor of these sizes are not all the same and the result is an error on the scaling value of the frequency. Really it's not much important that the value of this frequency on the roller rig is calculated with precision. Therefore it is possible to underline that, except the bogie yaw mode, the roller rig allows an estimate of eigenfrequency with a small error.

Iwnicki	Jaschinski ro=1	Jaschinski ro=0,5	Pascal	Jaschinski modified	Mode
8.7%	19.6%	19.6%	21.7%	19.6%	body lateral
0.0%	0.0%	0.0%	0.0%	0.0%	body yaw
11.1%	11.1%	11.1%	11.1%	11.1%	body vertical
0.0%	0.0%	0.0%	0.0%	0.0%	body pitch
1.4%	5.5%	5.5%	6.2%	5.5%	body roll
-10.2%	1.4%	2.8%	3.3%	40.2%	bogie yaw
7.6%	8.6%	8.7%	15.5%	7.7%	bogie pitch
0.0%	0.0%	0.0%	0.0%	0.0%	bogie vertical
2.4%	0.3%	-0.6%	0.1%	3.4%	bogie roll

Tab 8. *Percent error of a real vehicle*

	scala 1:1	**Pascal**	**Iwnicki**	**Jsch 1**	**Jsch 0.5**
ADAMS	2.14	1.92	2.17	2.2	2.21
Analytic	2.12	2.11	1.95	2.16	2.11

Tab 9. *Comparison between ADAMS simulations and analytic value of the yaw frequency.*

Scale 1:1	Iwnicki	Pascal	Jachinski $\rho = 1$	Jaschinski $\rho = 1$	Jaschinski Modified	**Test's Condition**
60.9 m/s	**75m/s**	**61 m/s**	**58 m/s**	**55.8 m/s**	**60 m/s**	On Track Vcr
59 m/s	**76 m/s**	**56 m/s**	**51m/s**	**51.8 m/s**	**58 m/s**	On Roller Vcr

Tab 10. *Critical speed*

Table 10 reported the critical speed values. The built model according to the proposed laws by Iwnicki overestimates the stability threshold, whereas the best results are obtained with the Pascal's and "Jaschinski modified" 's laws. We have investigated the possibility of realising a roller rig with the methodology of "Jaschinski modified". The second model has been carried out in order to design machine – members (electrical engine, control system). We present here the achieved results on the curve's simulations. The reference conditions are:

- vehicle in a curve with constant radius
- speed in progress V = 120 km/h
- on each chart are reported the trend lines for curving radius R = 1500 m, R = 1000 m, R = 800 m.

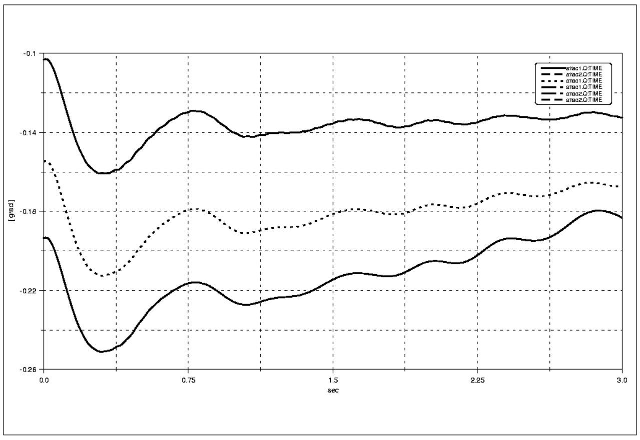


fig 2 *Difference between the tangent curve and the wheel direction for the three considered curves.*

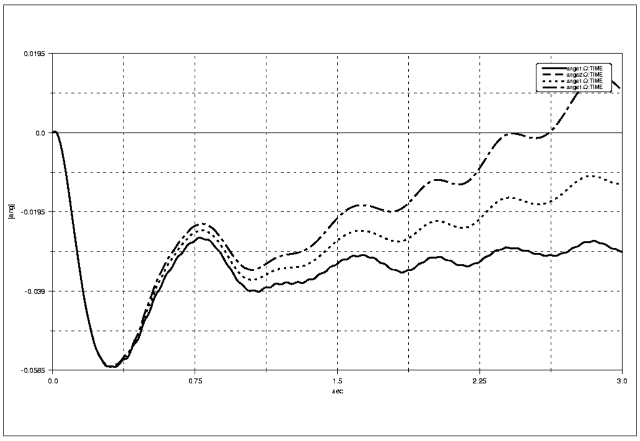


fig 3 *Rotation angle of wheelset.*

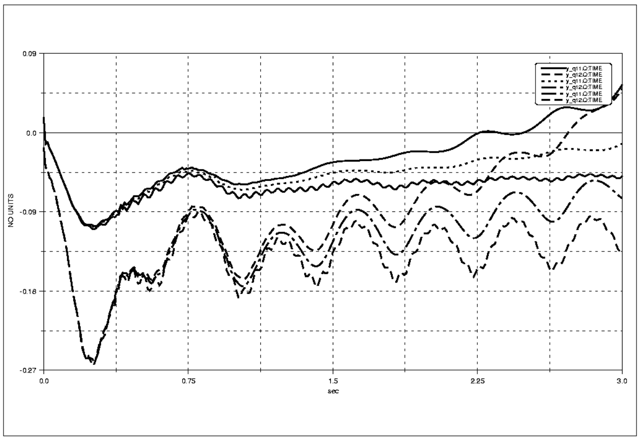


fig 4*Parametrized ratio between the normal force and the lateral creep force on each wheel.*

In this simulations it's possible to see that the vehicle is unstable for small radius of curve and that the design of a roller for stationary curving has to take into account the control device to avoid all these effects.

References

[1] O. Reynolds, On the Dynamical Theory of Imcompressible Viscous Fluids and the Determination of the Criterion, Phil. Trans. R. Soc. 186(1895).

[2] H. Chollet, Etude en similitude mechanique du contact ruoe raie, INRETS(tesi di dottorato, Grenoble) 1995.

[3] A. Jaschiski, On the Application of Similarity Laws to a Scale Railway Bogie Model, Oberpfaffenhofen, 1990.

[4] A. Castelli, Study of the dynamic behaviour of a 1/5 scale Roller – Rig (tesi Politecnico di Torino, Dicembre 98).

[5] S.D. Iwnicki, Roller – Rig scaling European Roller – Rig meeting, Manchester 98

[6] Ming Yan, A Study of the inherent errors in a Roller –Rig model of railway vehicle Dynamic behaviour.

[7] F.C. Carter, On the Action of a Locomotive Driving. Wheel. Proc. Roy. Soc. A112 (1926).

[8] P. J. Vermuelen, K.L. Johnson (1964): Contact of nonspherical bodies transmitting tangetial force, ASME Journal of Applied Mechanics, Vol 31, No. 2, 1964.

[9] Hertz (1882b) , *Ü* ber die Berührung fester elasticher Körper und über die Härte. In: H. Hertz, Gesammelte Werke, Band 1. Leipipzig: J.A. Barth (1895).

[10] Y.P. Chiu, M. J. Hartnett (1987), A numerical solution for the contact problem involving bodies with cilindrical surface considering cylinder effect. Journal of Tribology 109.

[11] J.J. Kalker (1973): Simplified Theory of rolling contact. Delft Progress Report 1

[12] ADAMS, USER'S Manual, MDI.

[13] ADAMS, Solver, MDI.

[14] R. Panagin, La dinamica del veicolo ferroviario, Levrotto & Bella, 1998.

[15] J.J. Kalker, A Fast algorithm for the simplified theory of rolling contact, Vehicle System Dynamics.

[16] E. Napoli, Progettazione e simulazione dinamica di un roller rig in scala 1:5 (Tesi, Politecnico di Torino 1999)