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Simulation of a freight bogie with friction dampers

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1 - Introduction

In the case of freight railways friction elements are often used as dampers for cost and maintenance reason without care of the dynamic performance of the vehicle. On the other hand this technical choice leads to low performance of the vehicle.
The aim of this work is to evaluate the behaviour of an Y25 freight bogie that is the most diffuse in Europe where friction dampers is used to reduce vertical and hunting vibrations.
The purpose is also to improve a bi-dimensional model of friction damper and to test its numerical efficiency on a whole vehicle model in a multibody code.

2 - Friction Model

The representation of the friction forces following the classical Coulumb's law. Often this formulation lead to numerical simulation problem, due to the discontinuity introduced by the friction force behaviour.

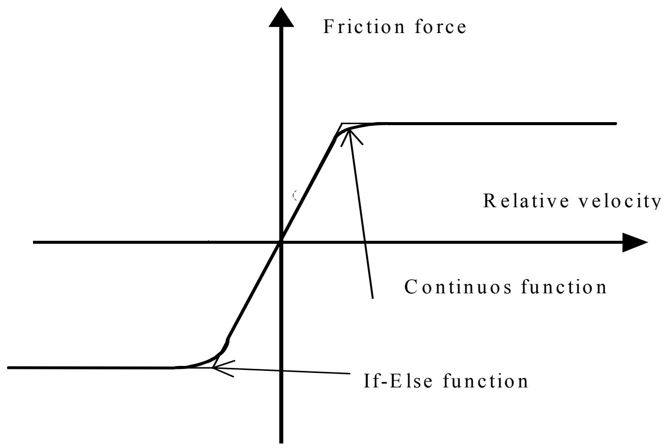


Figure 2.1- Friction force patterns

To avoid this problem the friction force has been modelled with a continuos function of the relative velocity between the friction surface described by the following equation:

$$F_f = \frac{V \cdot \chi}{\sqrt{1 + \left(\frac{V \cdot \chi}{N \cdot \mu}\right)^2}} \qquad (1)$$

$$\lim_{V \to 0} F_f = V \cdot \chi$$
$$\lim_{V \to \pm\infty} F_f = \pm N \cdot \mu$$

Where F_f is the friction force, V is the relative velocity. The χ parameters represent the angle between the velocity axis and the friction force curve around the origin, μ is the kinetic friction coefficient, N is the force normal to the friction surfaces.

In the case of the Y25 bogie the friction surfaces allow the relative motion on the plane which include the vertical and the lateral direction (see fig. 2.2) , so it is necessary to extend the friction force formulation to two degrees of freedom.

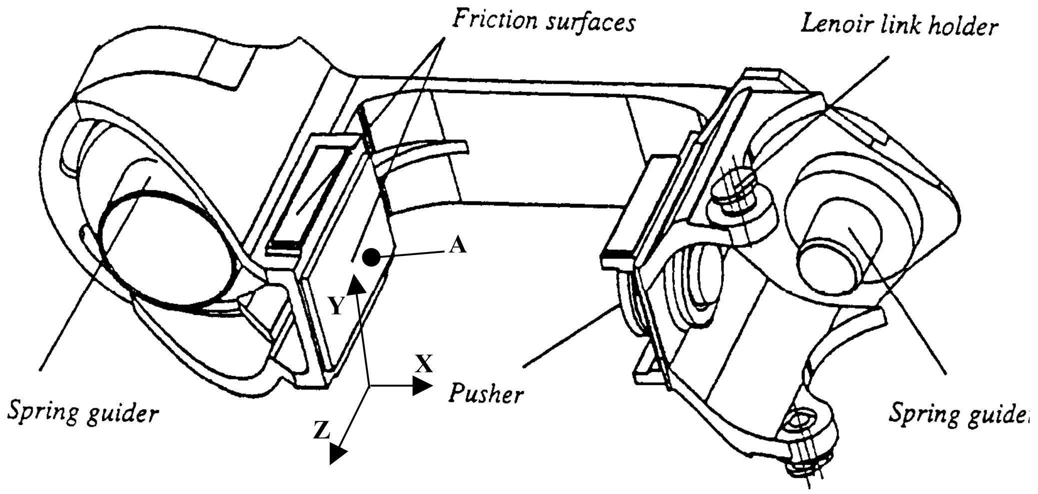
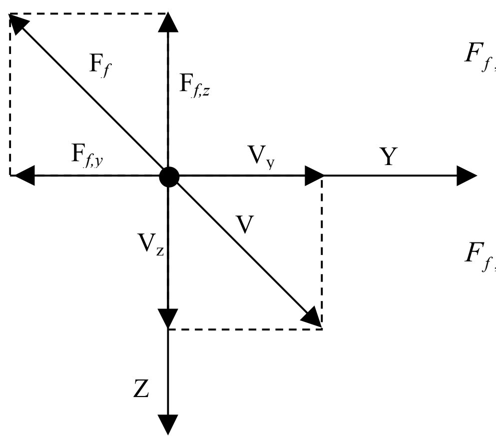


Figure 2.2- Friction surface on the axle-box

The two components of the vertical and lateral velocity are $\dot{z}$ and $\dot{y}$; the absolute value of the relative velocity is :

$$V = \sqrt{\dot{z}^2 + \dot{y}^2} \qquad (2)$$

The components of the velocity are $\dot{z} = V \cdot \cos(\theta)$ and $\dot{y} = V \cdot sin(\theta)$, while the components of the friction force in the z and in the y direction are:



$$F_{f,Z} = F_f \cdot \cos(\theta) = \frac{V \cdot \chi \cdot \cos(\theta)}{\sqrt{1 + \left(\frac{V \cdot \chi}{N \cdot \mu}\right)^2}} = \frac{\dot{z} \cdot \chi}{\sqrt{1 + \left(\frac{V \cdot \chi}{N \cdot \mu}\right)^2}} \qquad (3)$$

$$F_{f,Z} = F_f \cdot sin(\theta) = \frac{V \cdot \chi \cdot sin(\theta)}{\sqrt{1 + \left(\frac{V \cdot \chi}{N \cdot \mu}\right)^2}} = \frac{\dot{y} \cdot \chi}{\sqrt{1 + \left(\frac{V \cdot \chi}{N \cdot \mu}\right)^2}} \qquad (4)$$

Figure 2.3- Friction force components in the ZY plane.

3 - Y25 Model

In the Y25 bogie the damping of vertical motion is given by a mechanical device placed in the primary suspension called *Lenoir link*, that use a portion of the wagon weight to support the normal load to the friction surfaces, so vertical damping is also load sensitive.
The secondary suspension is reduced to a centre pivot with a very high stiffness, and the damping of hunting mode is given by a couple of sidebearers, two friction surface mounted outboard of the bogie pivot and preloaded with springs. The sidebearers also support the rolling stiffness of the car.
Therefore two kind of friction elements are present in the vehicle and both are load sensitive.
Owing to the high complexity of the Lenoir Link substructure, we have reduced it to a transfer function introduced between the axle and the axle-box.
In order to test the function we have simulated a single axle-box with a detailed model realised with ADAMS ("detailed axle-box Model"), then we have introduced the transfer function of the Lenoir-link in a Matlab Model comparing the result.
The transfer function is then adopted to create the model of the entire wagon used to perform the simulations. with Adams/Rail.

3.1- ADAMS detailed model of the axle-box

The ADAMS detailed model of the axle-box is described in the following figure.

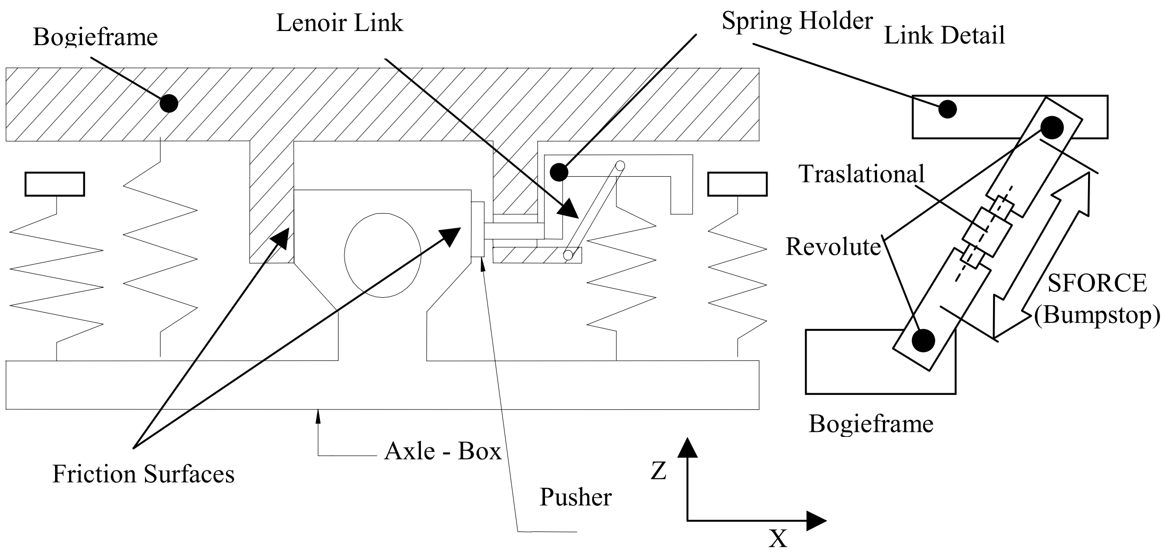


Figure 3.1 : Axle box detailed model.

The model is composed of 7 rigid bodies :

- Bogieframe : the bogiframe is connected to ground through a traslational joint, which allow only the vertical motion (Z).
- Lenoir Link (2 bodies) : The Lenoir link has been built as two separate parts, one linked to the bogie and the other to the spring holder, both with two revolute joints. The two parts are then linked each other with a translational joint and a single force which act as a unilateral bumpstop. When a force is applied to the spring holder by the spring the two part of the link are moved away and the bumpstop operate such that the force is transferred to the bogie.
- Spring Holder : The Spring Holder Keeps the inner spring in the left side of the axle-box; it is connected to the Lenoir-Link as shown above. The Link inclination split the force supplied by the spring in two components in the X-Z plane. The spring-holder is connected to the left side of the pusher with a bumpstop so that the force given by the Lenoir-Link in the X direction is transferred to the pusher itself.
- Pusher : the pusher is connected to the bogieframe with a traslational joint which allow only the relative motion in the X direction. The right side of the pusher is connected to the axle box with a force vector, this element model the first friction surface. In the X direction the force vector act as a bumpstop, the value of the X force is then used as the Normal force for the bi-dimensional friction force implemented in the Y and Z direction.

- Axle-Box : The primary friction surface is located in the left side of the Axle-Box in the Y-Z plane. This surface has been modeled using four force vectors, one to each vertex of the surface. Each Force vector is modeled as the one on the pusher, so that the total friction force is distributed among the four force depending on the normal force acting on each vertex.
  The vertical load is transferred from the Axle-Box to the bogie frame thought four springs, the outer springs have a gap which is closed only in the laden condition, so that in the tare load condition only the inner springs support the load. Moreover the right inner spring act thought the Link as described above. The axle box is connected to the axle with a revolute joint.
- Axle : The axle in this model is connected to ground with a planar joint.

All the bumpstop elements have been modeled with a stiffness of 1e8 N/m and the damping of 1000 N·s/m.

3.2- Equivalent model of the Lenoir Link - Matlab Model

The purpose of the analytical model of the axle-box is to create a transfer function between the bogie and the wheelset.
In the following we will find the equation of the forces exchanged between bogieframe and axle-box due to the link, the friction force and the springs.

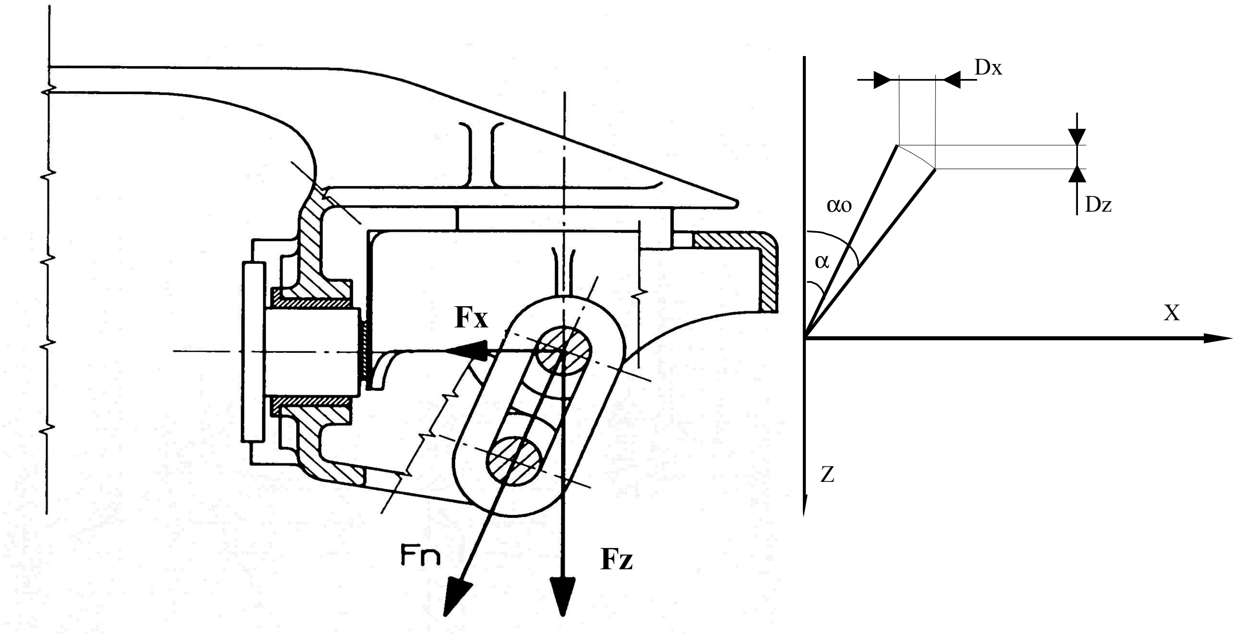


Figure 3.2 :The Lenoir link and the exchanged force

The Lenoir link, due to its inclination with the vertical direction in the vertical-longitudinal plane, couples the stiffness in z and x direction.
The vertical and horizontal components of the force, as it appears in figure n.4 are linked with the relation :

$$Fx = Fz \cdot \tan\alpha \qquad (5)$$

The rotation of the link around the lower hub describe a circumference defined by the following relation :

$$x^2 + z^2 = l^2 \qquad (6)$$

The z displacement can be expressed as :

$$z = \sqrt{l^2 - x^2} \qquad (7)$$

Differentiating eq. (6) :

$$\frac{dz}{dx} = \frac{-x}{\sqrt{l^2 - x^2}} = -\frac{x}{z} = \tan\alpha \qquad (8)$$

For small displacement, when $\Delta x \cong dx$ and $\Delta z \cong dz$, it is possible to write :

$$\Delta z = \Delta x \cdot \tan\alpha \qquad (9)$$

The forces acting between the bogieframe and the axle-box due to the link coupling are :

$$\begin{cases} Fz = kz \cdot \Delta z + kz \cdot \tan\alpha \cdot \Delta x \\ Fx = kz \cdot \tan^2\alpha \cdot \Delta x + kz \cdot \Delta z \cdot \tan\alpha \\ Fy = ky \cdot \Delta y \end{cases} \quad (10)$$

To complete the model is required to consider both the contribution of the friction force, of the left spring and of the two outer springs.
The force normal to the friction surfaces is given by the Fx force reported above. In the following we will set N=Fx. The friction forces are given by (the factor 2 is due to the presence of 2 friction surfaces) :

$$F_{fz} = 2 \cdot \frac{\chi \cdot Vz}{\sqrt{1 + \left(\frac{\chi \cdot V}{N \cdot \mu} \right)}} \quad (11) \qquad F_{fy} = 2 \cdot \frac{\chi \cdot Vy}{\sqrt{1 + \left(\frac{\chi \cdot V}{N \cdot \mu} \right)}} \quad (12)$$

The left spring contribution is given by :

$$\begin{cases} Fzs = kx \cdot \Delta x \\ Fys = ky \cdot \Delta y \\ Fzs = kz \cdot \Delta z \end{cases} \quad (13)$$

The outer springs contribution is given by :

$$\begin{cases} F_{zos} = 2 \cdot kz_2 \cdot (\Delta z - \Delta z_0) \text{ if } \Delta z \geq \Delta z_0 \\ F_{zos} = 0 \text{ if } \Delta z < \Delta z_0 \end{cases} \quad (14)$$

Finally the forces between the bogie and the axle-box are given by :

$$\begin{cases} Fx = kz \cdot \tan^2\alpha \cdot \Delta x + kz \cdot \Delta z \cdot \tan\alpha + kx \cdot \Delta x \\ Fy = 2 \cdot ky \cdot \Delta y + F_{fy} \\ Fz = 2 \cdot kz \cdot \Delta z + kz \cdot \tan\alpha \cdot \Delta x + F_{zos} + F_{fz} \end{cases} \quad (15)$$

The system analyzed with Matlab is described by the following equations:

$$\begin{cases} M \cdot \ddot{x} = -Fx \\ M \cdot \ddot{y} = -Fy \\ M \cdot \ddot{z} = -Fz \end{cases} \quad (16)$$

3.3 - Comparison

The two Model described before have been compared. The free response of the system to a to a vertical and lateral impulse has been investigated.
The maximum value of displacements are reported on table 3.1 and 3.2.
The simulation are made in the tare condition.
The impulse is supplied by the Vertical load itself for the vertical motion, the lateral impulse is supplied instead with an initial velocity of 0.1 m/s.

	ADAMS	Matlab
Max. Vertical displacement [m]	0.0391	0.0387
Mean Vertical displacement [m]	0.0202	0.0203
Frequency [Hz]	3.34	3.33

Table 3.1- Vertical Motion

	ADAMS	Matlab
Max. Longitudinal displacement [m]	0.000071	0.000070
Frequency [Hz]	2.96	2.99

Table 3.2 – Lateral Motion

The simulation are performed using the following parameters :

ADAMS : GSTIFF / digit=7

Matlab : Ode15s / abs. error =1E - 6 / rel. Error = 1E - 8

The two models have the same behaviour, so in the model of the entire vehicle the Axle-Box force model has been implemented. These choice has been adopted since the detailed ADAMS Model require a large computation time for the single Axle-box and makes ineffective its application to an ADAMS/Rail Vehicle Model.

3.4 - Vehicle Model

The vehicle model has been modelled using the function described in par 3.2 to simulate the connection between each axle-box and the bogie. The function of Fx, Fy, Fz have been directly introduced in a single Force Vector element. However has been necessary to introduce also a contribute to simulate the closure of the gap present in the axle-box.

This contribute has been neglected during the previous simulations.

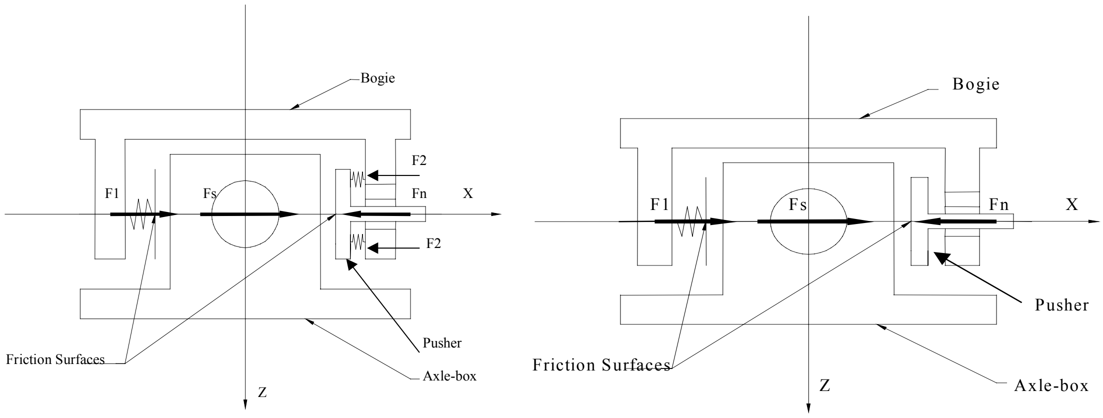


Figure n. 3.5- (a) Axle-box when the clearance is closed. (b) Axle-box when the clearance is open.

The forces indicated as F1 and F2 on figure 3.5 have been added to the force Fx and to the Normal Force, their behaviour is the same of a bumstop with a linear stiffness of 10E8 N/m.

The secondary suspension is made by a centre pivot with a very high stiffness (see table 3.2) in each translational direction, which allow the tree rotation. The roll torsional stiffness is supported by two sidebearers (one for each bogie) placed at a distance of 850 mm from the pivot in the lateral direction and preloaded each with the 31% of the tare load.

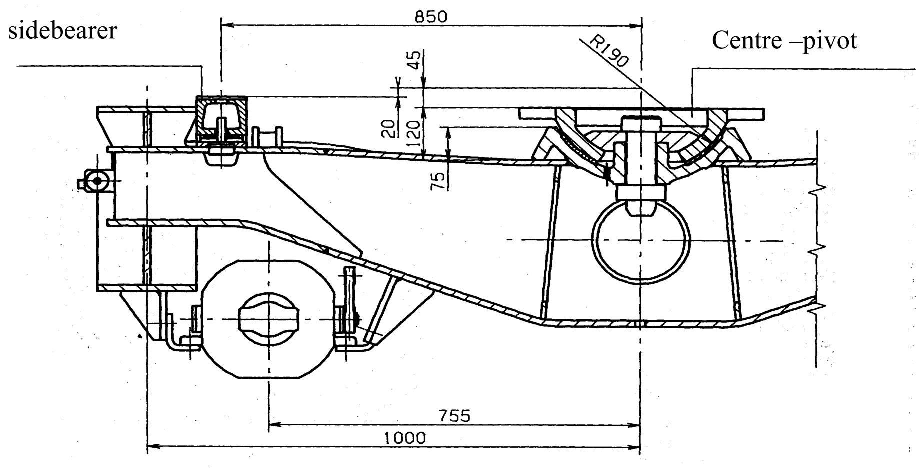


Fig. 3.6 – Transversal section of the bogie.

The sidebearers also supply the Yaw damping acting as friction elements. The same model described on par. 2 as been adopted and implemented on a Force Vector which has a linear stiffness in the vertical direction and a friction damper in the XY plane. The Normal force is the one exchanged by the stiffness in the Z direction.

In the following tables 3.3 and 3.4 are reported the inertial and stiffness data of the model.

Inertial data	Mass	Jxx	Jyy	Jzz
Body	**[Kg]**	**[Kg m^2]**	**[Kg m^2]**	**[Kg m^2]**
Wagon (tare)	12400	8.4E4	1.3E6	1.28E6
Wagon (Laden)	172400	9.7E5	1.5E7	1.4E7
Bogie frame	2070	1400	2100	2400
Wheelset	1225	750	140	750

Table 3.3 – Inertial Data

Stiffness	Kx	Ky	Kz
	[N/m]	**[N/m]**	**[N/m]**
Primary suspension	500000	500000	175000
Internal spring			655000
Secondary suspension			
Centre-pivot	10E8	10E8	10E8
Side bearers		380000	580000

Table 3.4 – Stiffness

4 - Simulations

In the following sections are reported all the simulation performed on the Vehicle Model using ADAMS/Rail.

4.1 - Slant Test

Railway vehicles running on curved track are subject to the altimetrical differences between the two rails imposed by the cant angle, for this reason, vehicle with very high torsional stiffness have serious trouble to cross these kind of defect, cause the vertical load on the wheels may decrease with derailment risk.
In our work we have considered three different event which may cause the wheel unloading and which usually occur together during curving.
The first event is caused by a slant between the two rail extended to the entire side of a wagon, and that lead to a superelevation of only one side (left/right) of a bogie respect to the second bogie. We indicate this events as slant between the pivots (fig. 4.1 -a).
The second event is a slant between the rails with a shorter extension which is applied to a single bogie lifting the first wheel and pulling down the second of only one side of the bogie. This second event is shown in fig 4.1- (b) and is indicated as bogie slant.
The third effect we have considered is the one due to the lateral acceleration non compensated by the cant (anc) which, for freight vehicles, can reach a value of 0.6 m/s^2 during curving. The effect of anc is an unloading of all the wheels of the inner side of the curve and a loading of the wheel of the outer side.
The simulations have been made using the Test-Rigs of ADAMS/Rail 9.1.1. The given superelevation has been chosen according to the reports ORE B 55 RP 6 / RP 8 which stand a limit value for the maximum grade that may be found in the railway for the bogie slant (g+) and for the slant between the pivots (g*) :

$$g^* = \frac{15}{2a^*} + 2.0 = 4\,‰ \qquad g+ = 7.0 - \frac{5}{2a+} = 8\,‰$$

Where 2a* is the distance between the two pivots and 2a+ is the distance between the two axles of a bogie. These inclinations bring the following vertical displacement values :
Dz* = 36.8 mm for the slant between the pivots.
Dz+ = 14.4 mm for the bogie slant.
Next we denote each wheel of the vehicle as “ij” (e.g. 12) where “i” is the number of the wheelset to whom the wheel belong and “j” is the side (1 = right, 2 = left).
The following figure shows the way we lift/lower the rigs :

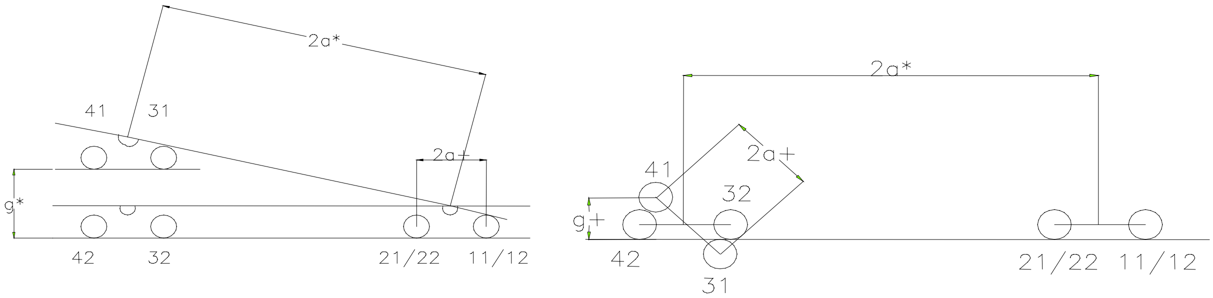


Figure 4.1- (a) Slant between the pivots (b) Slant of the bogie

Usually the skill of a vehicle to cross a slant is checked thought static test, however we thought it right to made dynamic test (simulated) so to keep in account the contribution of the friction force supported by the Lenoir-Link which increase the wheel unloading.

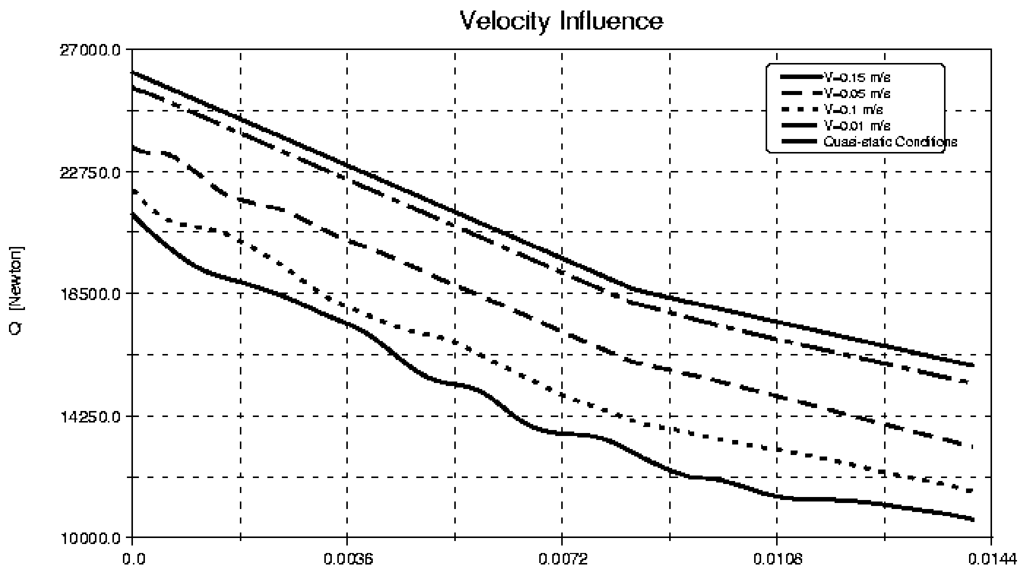


Fig. 4.2 – Effect of the lowering velocity on the wheel unloading.

On Fig. 4.2 is shown the load on the wheel 41 while the rig where this wheel lay is lowered using different values of the velocity. The first curve is obtained thought a quasi static analysis. Is clear that over a certain velocity arise dynamic effects, therefore we have performed all the simulations using a velocity of 0.1 m/s, in this way we consider a quantity of the friction force effort to the unloading without introducing relevant dynamic contributions.

	wheel	41	42	31	32	21	22	11	12
Q_0	[KN]	26.524	26.524	26.524	26.524	26.524	26.524	26.524	26.524
Q_{anc}	[KN]	23.015	30.038	23.015	30.038	23.015	30.038	23.015	30.038
DQ_{anc}	[KN]	3.509	-3.514	3.509	-3.514	3.509	-3.514	3.509	-3.514
DQ_{anc}/Q_0		13.23%	-13.25%	13.23%	-13.25%	13.23%	-13.25%	13.23%	-13.25%
Q_1	[KN]	24.922	28.142	24.904	28.137	28.48	24.57	28.465	24.575
DQ_1	[KN]	1.602	-1.618	1.62	-1.613	-1.956	1.954	-1.941	1.949
DQ_1/Q_0		6.04%	-6.10%	6.11%	-6.08%	-7.37%	7.37%	-7.32%	7.35%
Q_2	[KN]	13.213	40.042	40.166	13.317	26.524	26.524	26.524	26.524
DQ_2	[KN]	13.311	-13.518	-13.642	13.207	0	0	0	0
DQ_2/Q_0		50.18%	-50.97%	-51.43%	49.79%	0.00%	0.00%	0.00%	0.00%
DQ/Q Tot		69.45%	-70.31%	-32.10%	30.46%	5.86%	-5.88%	5.91%	-5.90%

Table 4.1 - Slant -Tare

On the following Table 1 – 2 are shown the unloading of each wheel in the tare and in the laden condition.

Q_0 is the reference load acting on each wheel, Qanc the load measured after the superposition of a lateral acceleration of 1 m/s^2, Q_1 is the load after the application of the bogie slant, Q_2 the load due to the application of the slant between the pivots.
DQ shows the load difference of each case respect the reference load.
The report ORE B 55 RP 8 fix a limit of 0.8 (80%) to the maximum admissible DQ/Q_0 ratio. The total DQ/Q_0 ratio is obtained summing the contribution of Qanc,Q1 and Q2.

	Wheel	41	42	31	32	21	22	11	12
Q_0	[KN]	111.76	111.76	111.76	111.76	111.76	111.76	111.76	111.76
Q_{anc}	[KN]	89.26	134.21	89.26	134.21	89.26	134.21	89.26	134.21
DQ_{anc}	[KN]	22.5	-22.45	22.5	-22.45	22.5	-22.45	22.5	-22.45
DQ_{anc}/Q_0		20.13%	-20.09%	20.13%	-20.09%	20.13%	-20.09%	20.13%	-20.09%
Q_1	[KN]	109.38	114.1	109.39	114.07	105.82	117.75	105.85	117.71
DQ_1	[KN]	2.38	-2.34	2.37	-2.31	5.94	-5.99	5.91	-5.95
DQ_1/Q_0		2.13%	-2.09%	2.12%	-2.07%	5.31%	-5.36%	5.29%	-5.32%
Q_2	[KN]	95.879	126.73	127.91	96.485	111.76	111.76	111.76	111.76
DQ_2	[KN]	15.881	-14.97	-16.15	15.275	0	0	0	0
DQ_2/Q_0		14.21%	-13.39%	-14.45%	13.67%	0.00%	0.00%	0.00%	0.00%
DQ/Q Tot		36.47%	-35.58%	7.80%	-8.49%	25.45%	-25.45%	25.42%	-25.41%

Table 4.2 – Slant - Laden

It is necessary to point out that the bogie in the tare case is very close to the limit imposed by the Norms. In the Laden condition the situation is quite better. The main reason of the behaviour of the bogie during the Slant simulations may be found in a very high value of the torsional stiffness of the bogie primary suspension in the vertical direction. This is clear from the bad loading distribution due to the bogie slant, while the slant between the pivots is not critical.

4.2 - Riding stability

As already say, the vehicle has a number of friction elements in the primary and in the secondary suspension. The presence of these elements, which moreover act a big influence on the stability of the vehicle, make inadequate to find the critical speed trough a series of eigenvalue analysis, cause the heavy non linearity of the friction elements model.
Therefore the simulations to find the critical speed have been made trough transient non-linear analysis on a straight track, giving a lateral impulse to the wheelset to be able to excite the hunting motion.
The criterion to consider as unstable the ride to a certain speed was to watch if the wheelset oscillations led by the impulse were damped or not.

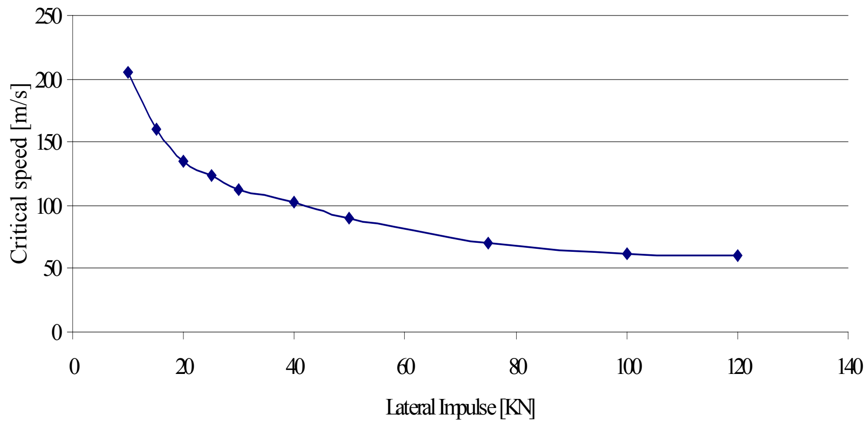


Fig 4.3 – Effect of the impulse force on the critical speed (Laden)

However is known [5],[9], that the impulse value may have a direct influence on the critical speed. Therefore several simulations have been performed changing the impulse force to find a value over which no more critical speed increment are detected.

The figure 4.3 reports this limit for the laden vehicle which is of 100 KN, for the tare condition the value is 25 KN, these value were always used in the following for the simulations.
The contact model adopted was the one defined by the Rail / Level 2a with an equivalent conicity of 0.2.
Several simulation are performed changing the anti-yaw friction dampers parameters (such as the friction coefficient), while regarding the damping supplied by the Lenoir-Link no variations are made since we have detected that this element have no influence on the critical speed.
In the following tables is shown the effect of the variation of the friction coefficient and of the χ parameters (which represents the starting inclination of the characteristic damping force – relative velocity) in both the tare and the laden condition.

Table 4.3 -Effect of the friction coefficient – Laden						
Friction Coefficient	0.4	0.3	0.1	0.05	0	Linear
χ	Critical speed [m/s]					23.55
3500000	61	54	30	26	22.5	
750000	53	44	29	25	22.5	

Table 4.4 -Effect of the friction coefficient – Tare						
Friction Coefficient	0.4	0.3	0.1	0.05	0	Linear
χ	Critical speed [m/s]					25
3500000	65	59	37	32	24	
750000	58	49	35	31	24	

From the obtained values is clear that the inclination χ is as much important as bigger is the friction coefficient. Furthermore the result obtained with an eigenvalue analysis (Linear in the table) are near to the ones obtained in the transient simulation without anti-yaw dampers (friction coefficient =0).
Finally the effect of the wear of the rails and of the wheels is considered, using different values of the equivalent conicity. In the following table are reported the results for the Laden condition (which is the more critical).

Table 4.5 -Effect of the equivalent conicity – Laden				
Eq. Conicity	[rad]	0.05	0.2	0.35
Critical speed	[m/s]	69	61	57

Failure of a sidebearers (anti-yaw dampers)

Since on freight vehicles the maintenance is made seldom, in this section has been considered the effect of a failure to one of the sidebearers (on the 4 installed).
Two failure mode are considered :
Failure mode 1 : heavy reduction of the friction coefficient due to wear, presence of oil or ice on the friction surfaces. This mode is simulated adopting a friction coefficient of 0.01 in the damper.
Failure mode 2 : block of a sidebearer, event which may arise after a long period of inactivity caused by the formation of rust. This mode is simulated with a friction coefficient of 0.7.

Table 4.6 -Effect of the failure of a sidebearers			
Load	Critical Speed [m/s]		
	Failure mode 1: μ=0.01	Failure mode 2 : μ=0.7	No failure
Tare	53	57	65
Laden	47	49	61

Curving stability.

The vehicle in exam is made such that the torsional roll stiffness is supplied by the same element used as anti – yaw dampers. For this reason during curving, due to the lateral non compensated accelerations, the normal load acting on the friction surfaces of the sidebeares has a large variation between the two side of the vehicle. To keep in account this effect a simulation has been performed using a straight track but imposing on the vehicle a lateral accelerations of 1 m/s^2.

The result are shown in the following table :

Table 4.7 -Effect of non compensate acceleration during curving on critical speed.		
Load	Critical Speed [m/s]	
	Curve anc = 1 m/s^2	Straight
Tare	40	65
Laden	29	61

The running of the vehicle subject to an high value of lateral acceleration lead to a drastic reduction of the critical speed, which may be explained by the fact that one of the two sidebearer is locked by an high normal load while the one on the other side is almost unloaded and ineffective.

4.3 Curving

The evaluation of the vehicle behaviour during curving has been made with both the determination of the derailment safety ratio (Y/Q), of the wheelset yaw angle and verifying the maximum lateral force (Ripage).

Derailment safety ratio - Y/Q

The Y/Q ratio has been evaluated running the vehicle on curve with different radius to a speed such that the ANC value was of about 0.6 m/s^2 which is the maximum allowed for freight vehicles in Europe.
We choose to use non canted tracks because often on small radius curve (e.g. 60 m) the track is not canted in the reality, furthermore in non canted tracks the desired value of ANC is reachable at lower speed (far from the critical speed), so that is possible to evaluate the curving behaviour of the vehicle without superposition of dynamic effect and of slant effects which have been considered separately.
In Table 1 are shown the Y/Q value obtained for our vehicle on different curve tracks using the Level 1 contact model.
Obviously the Laden vehicle has a much better performance respect the tare vehicle.
However even in the tare case the limit ratio, which is fixed to 0.8 (depends on the shape of the wheel profile, we use S1002 profile) is respected with a good margin also in the 60 m curves, that are the minimum allowed for this vehicle.

LOAD	Curve radius	Speed	Anc	Y/Q
	[m]	[m/s]	[m/s^2]	[/]
Tare	200	7.5	0.6	0.15
Tare	100	11	0.6	0.20
Tare	60	6	0.6	0.43
Laden	60	6	0.6	0.13

Table 4.8 - Maximum Y/Q value obtained with the Level 1 contact model.

For a curve radius of 200 m have been made the comparison with the Y/Q values obtained trough a Level III simulation. In the following table are reported the maximum value of the Y/Q ratio for each wheel found during the simulation. Both the result are similar and furthermore the Level I method is conservative.

	Level I	Level III left	Level III right	Level III sum
Wheelset 1	0.12	0.37	-0.26	0.11
Wheelset 2	0.09	0.05	0.02	0.07
Wheelset 3	0.15	0.36	-0.26	0.10
Wheelset 4	0.08	0.05	0.02	0.07

Table 4.9 - Differences between level I and III derailment factor

Wheelset yaw angle

These last simulation are performed using the Rail Level III contact model, with the Fastsim 2 algorithm.
The result are reported only for the first wheelset in the running direction which shows the worst performance.

Wheelset 1	Curve		Yaw angles vrs. Track ref. frame[mrad]		
Load Condition	Radius	Curvature (Ψ_0)	Steady-state (Ψ)	Peak	Ψ/Ψ_0
	[m]	[mrad]	[mrad]	[mrad]	[%]
Laden	200	5	4.1	4.1	82 %
Laden	400	2.5	0.9	1.4	36 %
Laden	1000	1	0.3	0.7	30 %
Tare	200	5	4.9	4.9	98 %

Table 4.10 – Level III curving simulation – Wheelset 1 Yaw angle

The table reports the curvature of the considered curve Ψ_0 which is the yaw inclination which should affect the wheelset if no additional constrains were present on it. Since the primary suspension has a very high stiffness, the wheelset is retained to assume this inclination. The difference between the theoretical angle Ψ_0 and the effective wheelset angle is indicated as "Yaw angles vrs. Track ref. Frame" and is reported both for the steady state condition in the middle of the curve and for the maximum value (peak) assumed during the simulation.
The ratio Ψ/Ψ_0 give a measure of the curving performance of the vehicle, the higher is the ratio the less is the performance. It is shown as in tare condition on small radius curve the angle is high; this often lead to heavy wear troubles.

Ripage

The lateral forces between the rail and the wheelset must be limited within a certain value in order to avoid a failure in the rail or in the armature.
The limit for the total lateral force (Ripage) [6] can be found as:

$$Y_{\max} = \sum \left(Y_{left} + Y_{rigth}\right) = 0.85 \cdot \left(1000 + \frac{Q}{3}\right) [daN]$$

Where Q is the maximum vertical load on the wheelset, therefore a limit of 65150 N is found for the laden vehicle and of 19750 N for the tare wagon.
The results reported in table 4.11 show for the wheelset 1 (which is the first in the running direction and have the worst behaviour) that the limit value is not reached.
Is to keep in consideration that the superposition of a slant (not considered in this simulation) could reduce strongly the Ymax limit.

Wheelset 1	Curve	Lateral Force			
Load Condition	Radius	Left wheel	Right wheel	Sum	Ymax
	[m]	[N]	[N]	[N]	[N]
Laden	200	43800	29500	14300	65150
Laden	400	10300	350	9950	65150
Laden	1000	7900	3100	4800	65150
Tare	200	11500	6900	4600	19750

Table 4.11 - Level III curving simulation - Lateral forces between rail and wheels [N].

5 – Final Remarks

The vehicle in exam shows a satisfying behaviour in the laden condition although the axle load reaches 22 t. The main limitation is related to the high torsion stiffness of the bogie, which arise in the troubles to cross the track slants and therefore during curving on canted tracks. This trouble is much manifest in the tare condition. However must be observed that in this work the bogie flexibility has been neglected, while, indeed , it has a favourable influence on the phenomenon.
The second problem that has been observed, is the strong influence of the vertical load acting on the sidebearers on the Anti-Yaw damping and therefore on the critical speed. This sets a limitation for the maximum speed reachable in the curving behaviour.

After all, has been detected that the vehicle is suitable for the most critical working condition (Laden), while when the vehicle is unloaded, instead of having better performances, these are heavily reduced.
This fact is extremely restrictive considering that working with low axle load and using different suspension systems should be possible to run to higher speed reducing the travel time.

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