International Journal on
"Technical and Physical Problems of Engineering"
(IJTPE)
Published by International Organization of IOTPE

ISSN 2077-3528
IJTPE Journal
www.iotpe.com
ijtpe@iotpe.com

December 2015 | Issue 25 | Volume 7 | Number 4 | Pages 75-85

DYNAMIC, STABILITY AND SAFETY ANALYSIS OF WAGONS ON MD52 BOGIES WITH MODIFIED SUSPENSION SPRINGS

M.R. Saviz

Mechanical Engineering Department, Azarbaijan Shahid Madani University, Tabriz, Iran, saviz@azaruniv.ac.ir

Abstract- This paper presents a novel design procedure for the springs of wagons with two-stage suspension system, which satisfies the static, strength, and dynamic requirements as well as standard preferences. A relatively precise and detailed method is proposed for modeling the suspension elements for dynamic simulations. Three vehicle types are modeled by using the vehicle dynamics simulation package ADAMS/ Rail, providing details of the vehicles, bogies, suspension elements, constraints and kinematic connections. The suspension springs of bogie MD52 with a 1 m track gauge are modified. The wagons are subjected to different load conditions. The spring sets are calculated with the aim of satisfying static equilibrium condition, kinematic constraints, wagon height and vertical balance. Finally, the designed springs are used in simulations and results are compared with the standard permissible values. Lateral stability (hunting), critical speeds, safety (derailment ratios) and dynamic behavior (ride comfort) of each wagon type, subjected to standard track conditions and irregularities are evaluated.

Keywords: Dynamic Simulation, ADAMS/Rail, Bogie, Spring Design, Ride Quality Index, Derailment Safety.

I. INTRODUCTION

A passenger wagon bogie consists of a frame, bolster, wheelsets, primary and secondary suspension systems containing spring sets, dampers and connecting parts such as friction pads, bumpstops, axle boxes and centre pivot, which are in kinematic and dynamic interaction with each other. In bogies of type MD52, manufactured by Bombardier company, some new design considerations and modifications have been applied to increase the allowable and critical speeds of related wagons, such as load bearing side bearers, suspension system and axle box springs [1].

In this paper, the suspension springs of bogie MD52 with 1000 mm line gauge have been designed to satisfy both the static and dynamic conditions. Three new wagons are subjected to different load conditions and for each wagon a proper set of springs has been suggested. First, the suggested spring sets have been checked to satisfy the static equilibrium conditions and kinematic constraints, wagon height and balance. Thus, a computer code was developed for this purpose. The inputs are mass properties and important dimensions of car body and bogie components, predicted spring dimensions and material and kinematic constraints of suspension system. The outputs are displacements of wagon at different points, forces and maximum stresses in springs. Kinematic constraints are determined by bogie structural and geometric constraints given in original MD52 drawings and standard [3] for impact loaded and closed spring conditions. Next, the springs have been used to evaluate the lateral stability (hunting), critical speeds, safety (derailment ratios) and dynamic behavior (ride comfort) of each wagon type.

Because of high degrees of freedom and complexities of mathematical model for dynamic of a wagon and difficulties in modeling friction surfaces, bump stops and clearances, it is preferred to use the ADAMS/ Rail software. The following analysis steps have been taken:

- Preload analysis, which can be used to verify the static analysis.
- Stability analysis by means of Routh-Hurwitz criteria, which can be used to obtain the critical speeds.
- Dynamic analysis on different lines, which can be used to assess the Sperling ride index and derailment ratio, [2].

The main contribution of the present work is building a complete wagon model that contains all the kinematic and dynamic elements and constraints in bogie structure, suspension and connections between bogies and car body, such as bolster (a new component), friction in side bearers and axle boxes, anti-roll (torsion) bar, clearances between bolster and frame, center pivot as a cylindrical joint and axle box (leaf) springs, in order to comply with the standard requirements. For this purpose, the models of bogie components and car body have been assembled to create the wagon model. The Kalker linear theory for creep forces between rail and wheel, [4] and Wickens wheel- rail contact geometry [5, 6] are applied. The results are discussed and critical speeds, lateral derailment coefficient in different speeds are compared with permissible values given by related standards such as [7], as well as designer or manufacturer's constraints.

II. SPRING DESIGN PROCESS

In a two-stage bogie suspension, the car body weight is transferred through four side bearers (friction pads) to two bolsters, the sum of this load with each bolster weight is transferred to the outer springs of secondary suspension, which have longer free lengths than inner springs, then the load is supported by both inner and outer springs.

The sum of these loads and frame weight is transmitted to the primary suspension springs, where the same mechanism as secondary suspension is seen. In each bogie, there are eight primary spring sets and four secondary spring sets. An important element in MD52 primary suspension is the axle box leaf spring, which acts like a cantilever beam with one side on axle box and the other side (fixed) on the bogie frame.

The adjustment of this fastener is vital in operation of primary suspension. In static design process of suspension springs, the height of wagon body, bolster and frame in tare and different loaded conditions, as well as maximum shear stresses in springs must be checked and the spring deflection limits should fulfill the bogie structural and geometric constraints given in bogie drawings and related standard [3] in impact load and closed spring conditions. Bogie MD52 with metric gauge was designed by Bombardier Company [1]. The mass properties of original car body together with new wagon bodies are listed in Table 1.

Table 1. Weights of three new car bodies, [1]

Wagon Type	Symbol	Unit	(I)	(II)	(III)	Original
Loaded Wagon Body Weight	QewL	kg	29984	26286	29627	27080
Tare Wagon Body Weight	QewT	kg	22844	21066	27299	16400
Tare Center of Gravity Position	ax	mm	-45.6	-39.4	150.6	0
	ay	mm	-8.4	-9.3	9.1	0
Loaded Center of Gravity Position	ax	mm	-34.8	-499.2	331.7	0
	ay	mm	-6.4	-10.7	5.0	0
	az	mm	1856.0	1784.5	1645.2	-

In new car bodies, whose mass properties are very different from the original one (37% to 66% heavier), springs are re-designed, so that the car body height of unloaded wagons are equally constant and for loaded condition are in permissible range. Having car body mass properties, wagon dimensions like bogie base (Figure 1) and important dimensions of suspension system listed in Table 2, forces on side bearers can be calculated.

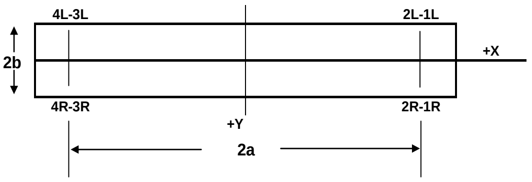

Figure 1. Geometrical dimensions of wagon, [1]

Transmitted weights to left and right side bearers of the rear and front bogies of each wagon are calculated by solving the static equilibrium, given as Equations (1):

Left set, Front bogie: $$1\text{-}2L = \frac{W}{2}\left[\left(1+\frac{a_x}{a}\right)\left(1-\frac{a_y}{b}\right)\right] \quad \text{(1a)}$$

Left set, Rear bogie: $$3\text{-}4L = \frac{W}{2}\left[\left(1-\frac{a_x}{a}\right)\left(1-\frac{a_y}{b}\right)\right] \quad \text{(1b)}$$

Right set, Front bogie: $$1\text{-}2R = \frac{W}{2}\left[\left(1+\frac{a_x}{a}\right)\left(1+\frac{a_y}{b}\right)\right] \quad \text{(1c)}$$

Right set, Rear bogie: $$3\text{-}4R = \frac{W}{2}\left[\left(1-\frac{a_x}{a}\right)\left(1+\frac{a_y}{b}\right)\right] \quad \text{(1d)}$$

where, W is car body weight. The calculation results are given in Table 3. For loaded and tare original wagon, the forces on the side bearers of the bogies are 40225.1 N and 66420.5 N, respectively.

Then, forces in the left and right spring sets of secondary suspension systems of front and rear bogies for tare and loaded conditions are calculated by using the following equations for bolster static equilibrium

Left set, Front bogie: $$\frac{(1\text{-}2L)+(1\text{-}2R)+W_b}{2}\left(1-\frac{a_y}{2bz^*}\right) \quad \text{(2a)}$$

Left set, Rear bogie: $$\frac{(3\text{-}4L)+(3\text{-}4R)+W_b}{2}\left(1-\frac{a_y}{2bz^*}\right) \quad \text{(2b)}$$

Right set, Front bogie: $$\frac{(1\text{-}2L)+(1\text{-}2R)+W_b}{2}\left(1+\frac{a_y}{2bz^*}\right) \quad \text{(2c)}$$

Right set, Rear bogie: $$\frac{(3\text{-}4L)+(3\text{-}4R)+W_b}{2}\left(1+\frac{a_y}{2bz^*}\right) \quad \text{(2d)}$$

where, W_b is the bolster weight. The forces in the left and right spring sets of primary suspensions for front and rear bogies are calculated, using the following equations for frame static equilibrium:

Left set, Front: $$\frac{(1\text{-}2L)+(1\text{-}2R)+W_b+W_f}{2}\left(1-\frac{a_y}{2bz^+}\right) \quad \text{(3a)}$$

Left set, Rear: $$\frac{(3\text{-}4L)+(3\text{-}4R)+W_b+W_f}{2}\left(1-\frac{a_y}{2bz^+}\right) \quad \text{(3b)}$$

Right set, Front: $$\frac{(1\text{-}2L)+(1\text{-}2R)+W_b+W_f}{2}\left(1+\frac{a_y}{2bz^+}\right) \quad \text{(3c)}$$

Right set, Rear: $$\frac{(3\text{-}4L)+(3\text{-}4R)+W_b+W_f}{2}\left(1+\frac{a_y}{2bz^+}\right) \quad \text{(3d)}$$

where W_f is the weight of bogie frame. By solving Equations (1) to (3) for tare and loaded conditions, numerical results are given in Tables 4 and 5, respectively.

Table 2. Dimensions of wagon and suspension system, mm, [1]

Description	Symbol	Value
Distance between Wheels	2bw	1065
Rail gauge	2br	1000
Distance between Primary Springs	2bz+	1473
Distance between Secondary Springs	2bz*	2030
Distance between Side Bearers	2b	930
Wheel Base	wb	2200
Primary Suspension Spring Play	pp	45
Secondary Suspension Spring Play	sp	77

Table 3. Calculated forces on the side bears of the bogies, N

Calculated Loads on Side Bearers:			Wagon Type		
			(I)	(II)	(III)
Tare Cond.	Front Bogie	Left	57717.4	53378.7	68525.0
		Righ	55669.1	51287.6	71260.6
	Rear Bogie	Left	58566.0	54055.6	65298.8
		Righ	56487.6	51938.0	67905.6
Loade d Cond.	Front Bogie	Left	75568.5	61860.1	77159.3
		Righ	73516.6	59069.6	78840.1
	Rear Bogie	Left	76414.9	72602.7	69379.6
		Righ	74340.0	69327.6	70890.9

When car body is assembled on bogies, some portion of body weight would be absorbed due to length difference between the outer and inner springs as

$$F_{Oi} = K_O^i \left(L_O^\circ - L_i^\circ - T_P \right) \quad (4)$$

where, K_O^i is the stiffness coefficient of ith outer spring and $L_i^\circ, L_O^\circ, T_P$ are the free lengths of the inner spring, outer spring and thickness of sheet plate between two springs, respectively. The inner and outer springs are parallel and reduce in length together, until balance of body weight minus $\sum_{i=1}^{24} F_{Oi}$ is achieved.

Table 4. Calculated forces on spring sets of an empty wagon, N

Calculated Loads for Spring Sets (Tare Cond.)			Wagon Type			
			(I)	(II)	(III)	original
Secondary Springs	Front Bogie	Left Bearer	60090.9	55737.1	72715.4	43315.6
		Right Bearer	59595.6	55229.3	73370.3	43315.6
	Rear Bogie	Left Bearer	60927.9	56403.8	69439.5	43315.6
		Right Bearer	60425.7	55889.9	70064.9	43315.6
Primary Springs	Front Bogie	Left Bearer	68582.1	64235.8	80890.0	51507.8
		Right Bearer	67804.3	63430.6	81895.7	51507.8
	Rear Bogie	Left Bearer	69420.4	64903.6	77619.7	51507.8
		Right Bearer	68633.1	64090.1	78584.7	51507.8

Table 5. Calculated forces on spring sets of a loaded wagon, N

Calculated Loads for Spring Sets (Loaded Cond.)			Wagon Type			
			(I)	(II)	(III)	original
Secondary Springs	Front Bogie	Left Bearer	77937.5	63951.1	80949.4	69510.9
		Right Bearer	77447.6	63278.6	81350.0	69510.9
	Rear Bogie	Left Bearer	78775.1	74506.9	73104.4	69510.9
		Right Bearer	78279.9	73723.4	73466.2	69510.9
Primary Springs	Front Bogie	Left Bearer	86416.4	72489.1	89195.3	77703.1
		Right Bearer	85668.7	71440.6	89804.1	77703.1
	Rear Bogie	Left Bearer	87254.9	83065.9	81357.6	77703.1
		Right Bearer	86500.0	81864.4	81913.0	77703.1

So, the body height of the empty wagon and body displacements for wagon under impact load is adjusted through changes in the stiffness of springs [8]. Stiffness coefficient of springs is defined as

$$K = \frac{Gd^4}{8D^3 n} \quad (5)$$

where, G is the shear module of the spring steel (for hot rolled steel [9]), d is the diameter of spring wire (mm), D is the average diameter of spring and n is the number of active spring coils = total number of spring coils -1.5.

It is obvious that for a constant material, by increasing d or decreasing D and n, the stiffness of the spring increases. To decrease D, it's necessary to change the form of springs and supports. Therefore, changing the other two parameters has been chosen.

There are two limitations for increasing the spring stiffness. The former is rigidity of suspension and reducing the ride quality in dynamic behavior and the latter is high shear stress in spring wire. Another solution is to increase the free length of springs without any change in stiffness. On the other hand, buckling of the springs curbs the increase in free length. The maximum shear stress in spring wire is calculated as follows, [8]:

$$\tau_{\max} = A \frac{8FD}{\pi d^3} \quad (6)$$

where, A is the Wahl coefficient:

$$A = \frac{4C-1}{4C-4} + \frac{0.615}{C} \quad , \quad C = \frac{D}{d}$$

By using the trial and error method, an initial wire diameter is guessed, along with a coil number and free length of the springs taken from the properties of original bogie, then, the stiffness is calculated. Equations (5) and (6) result in spring displacements (heights of car body and frame), forces and stresses in the springs at different loading conditions. If the results are not in the permissible ranges, the guessed values must be changed in order to find the proper springs.

A. Spring Calculation Results

Final results for proper inner and outer springs of the primary and secondary suspension systems, containing free length l_0 and d are obtained for the original wagon and three new wagon types, by using an Excel computer program, which are listed in Table 6. Stiffness coefficients are calculated, by using Equations (5) for the material used in the original springs 50CrV4, with mechanical properties given in Table 7, [9]. Stiffness coefficients for outer and inner springs of each set are shown in Table 8. Maximum shear stresses, occurring in the buffer block length (completely closed or solid spring), obtained from Equation (6) are given in Table 9.

Table 6. Final results of the MD52m suspension springs, mm

Wagon Type		Free Length (L_o)		Wire Diameter (d)	
		Outer	Inner	Outer	Inner
(I)	Secondary	425	401	31	22.5
	Primary	236	212	25	17
(II)	Secondary	431	407	31	22.5
	Primary	238	214	25	17.5
(III)	Secondary	438	414	34	22.5
	Primary	244	220	25	17.5

Table 7. Material properties of 50CrV4 steel springs, [9]

Standard . No.	S_y, Mpa	S_{ut}, MPa	Reduction of Area %	Impact energy J	G, Mpa
1.8159	1175	1495	40	21	78500

It is observed that none of stress values exceeds 65% of material yield strength. According to Figure 2, for a hot worked manufactured spring, permissible shear stress at buffer block length decreases with wire diameter, [3].

Table 8. Calculated stiffness for each spring set, N/mm

Wagon Type		(I)	(II)	(III)
Secondary	Outer Spring	186.4	186.4	269.7
	Inner Spring	122.2	122.2	122.2
Primary	Outer Spring	231.7	231.7	231.7
	Inner Spring	151.4	134.8	151.4

In Table 9, the maximum shear stress occurs in inner spring of (I) secondary suspension spring set (d=22.5 mm), which is 808 MPa and is smaller than permissible value (825 MPa), similar checks have been conducted for all springs. In the next step, the displacements in primary and secondary spring sets have been compared with kinematic constraints (185 and 330 mm for primary and secondary, respectively) these final values given in Table 10, determine the car body height. The sum of displacements in primary and secondary suspensions gives deviation of body height from its reference value.

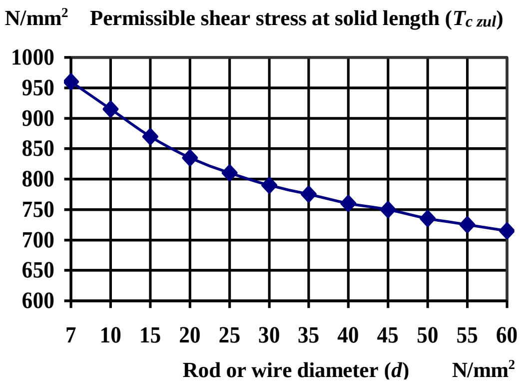

Figure 2. Permissible shear stress versus wire diameter, [3]

Table 9. Maximum shear stress for each spring set, N/mm^2

Wagon Type		(I)	(II)	(III)	Original
Primary Suspension	Outer Spring	708.0	695.6	745.2	679.4
	Inner Spring	764.4	751.5	809.7	729.5
Secondary Suspension	Outer Spring	773.4	753.3	755.2	718.8
	Inner Spring	808.4	785.4	785.4	746.8

Table 10. Deflection of primary and secondary spring sets, mm

Wagon Type			(I)	(II)	(III)	original
Primary Suspension	Front Bogie	Left	-0.10	-0.67	-1.80	-0.36
		Right	0.35	-0.18	-2.38	-0.36
	Rear Bogie	Left	-0.58	-1.07	0.10	-0.36
		Right	-0.13	-0.58	-0.46	-0.36
Secondary Suspension	Front Bogie	Left	-0.85	0.07	-1.00	0.04
		Right	-0.06	0.88	-1.82	0.04
	Rear Bogie	Left	-2.18	-0.99	3.10	0.04
		Right	-1.38	-0.17	2.32	0.04

Table 11. Required amounts of shimming, mm

Wagon Type		(I)		(II)		(III)		Original
Shimming under Spring		Left	Right	Left	Right	Left	Right	
Front Bogie	Secondary	1	0	0	0	5	6	0
	Primary	0	0	0	0	3	3	0
Rear Bogie	Left	3	0	3	0	0	0	0
	Right	0	0	0	0	0	0	0

Negative values indicate the spring deflection is over allowable limit, while smaller deflections have positive values. The left sprig set of (III) rear bogie secondary suspension has the greatest deviation from allowable displacement, because of tolerances in spring parameters and initial dimensions. These deviations never practically vanish, [10]. To balance the car body level, thin metal plates with 3 to 5 mm thickness are placed under spring sets to compensate the lack of height. Required shim values for wagon suspensions are exhibited in Table 11. As it is observed, (III) wagon needs serious shimming.

III. MODELING AND SIMULATION

To evaluate the dynamic performance of suspension springs, the dynamic behavior of wagons such as lateral stability (hunting), derailment safety (wheel- rail forces) and dynamic performance (comfort) of each wagon type is studied. Thus, the models of bogies and car bodies are provided and assembled to create wagon models with general identifications given in Table 12. In this work, the bolster element, side bearers (friction pads), clearances between bolster, frame and axle boxes in bogie model are created to consider their effects on dynamics, which has not been studied in the past. The sample bogie and wagon models in ADAMS/Rail are shown in Figures 3 and 4. The steps for performing the dynamic simulation are given in the following flowchart (Figure 5).

Table 12. Main dimensions of wagons, [1]

Description	Symbol	Value	Unit
Distance between Wheels	$2bw$	1065	mm
Rail gauge	$2br$	1000	mm
Distance between Primary Springs	$2bz^+$	1473	mm
Distance between Secondary Springs	$2bz^*$	2030	mm
Distance between Side Bearers	$2b$	930	mm
Wheel Base	wb	2200	mm
Primary Suspension Spring Play	pp	45	mm
Secondary Suspension Spring Play	sp	77	mm
Permissible Axle Load	-	10	tons
Wheel Diameter	wd	725	mm

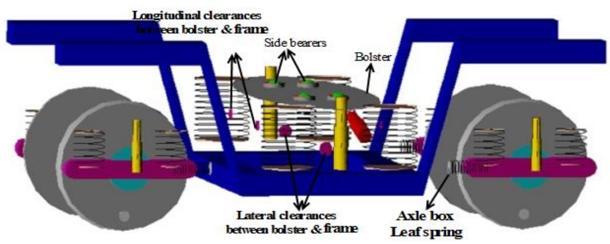

Figure 3. Modified Md52 bogie model

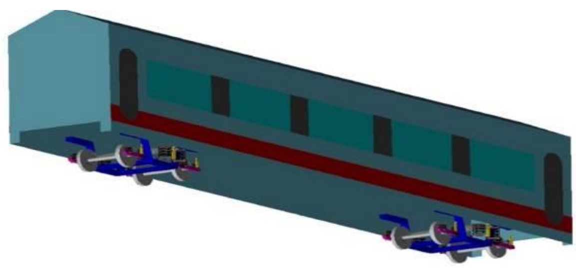

Figure 4. Assembled wagon model

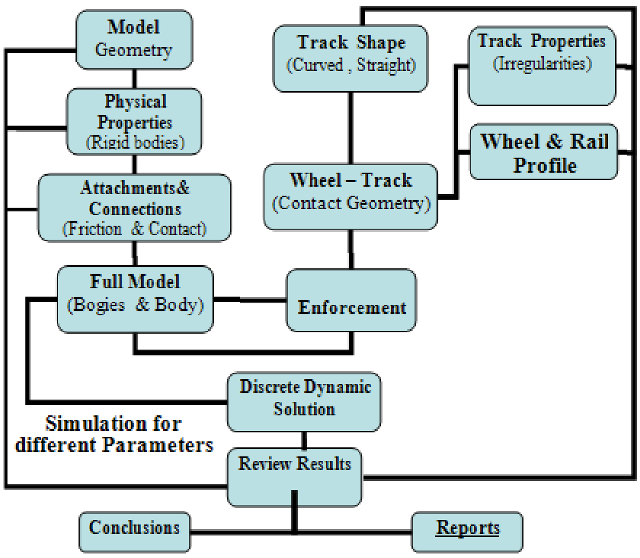

Figure 5. Basic steps of dynamic simulation

The mass properties of basic wagon elements required for dynamic analysis are calculated by using SolidWorks software based on available 3D models, listed in Table 13. The moments of inertia should be transferred into the center of gravity coordinates. These values for three car bodies are listed in Table 14.

Table 13. Calculated mass properties of wagon elements

Name of the Rigid Body	Mass (kg)	Moment of Inertia, kg m^2			Center of gravity, m		
		Ixx	Iyy	Izz	x	y	z
Wheel Sets	1300	593	100	593	0	0	-460
Frame	800	900	1700	2560	0	0	-646
Bolster	860	408	22	420	0	0	-817
Axle Box	100	2.1	5.6	5.6	-8	-3	-461
Car Body	According to Table 14						

Following analyses have been carried out during the vehicle dynamic simulation:

- Calculation of pre-loads on side bearers and springs.
- Stability analysis against hunting, and obtaining the Critical Speed by using Routh-Hurwitz criteria.
- Dynamic response analysis (measuring accelerations recommended by UIC 518, [7] on a straight track with ERRI dislocation, [11], and a curve with R = 300 m.
- Calculating the Sperling ride index at wagon maximum speed, [2].

Table 14. Calculated Moments of inertia in C.O.G coordinate

Wagon type		(I)	(II)	(III)
Moments of inertia in loaded cond. Relative to COG, kg m^2	Ixx	25795.5	21575.1	26827.2
	Iyy	172386.14	415350.3	173417.8
	Izz	154033.9	405608.2	154033.9

The wheel profile is taken S1002, [12]. All the parts are assumed rigid bodies with mass properties at the centre of gravity. The clearances between inner and outer springs (in spring sets) and also, between body center pin and its position on bogie have been ignored.

The main clearances of bogie (between bolster, frame and axle box) are modeled with bump stop elements. The friction between car body and side bearers has been modeled by using an external force developed by Bosso et al [13], as follows:

$$F_x = \left(\frac{v_x \cdot \chi}{\sqrt{\left(1 + ((v_m x) / (N.\mu))^2\right)}} \right)$$
$$F_y = \left(\frac{v_y \cdot \chi}{\sqrt{\left(1 + ((v_m x) / (N \mu))^2\right)}} \right) \quad ; \quad F_z = 0 \qquad (7)$$

v_x, v_y: Relative velocities in x, y directions, $v_m = \sqrt{v_x^2 + v_y^2}$: Relative velocity between carbody and bolster, N: Normal force, $\chi = 3.0e6$ N.s/m , and $\mu \approx 0.35$: Friction coefficient.

IV. SIMULATION RESULTS

Static preloads on bushings and suspension elements are found by using static equilibrium solver of ADAMS/ Rail. In this way, the loads on side bearers and spring sets are obtained. The preloads on secondary and primary suspension spring sets for loaded wagons are given in Table 15 for all wagon types. By comparing simulation results with calculated ones (Table 5), a good agreement is witnessed, while, the maximum difference between them is about 5.6%, which belongs to wagon (II).

The eigenvalue analysis is used to find natural frequencies of the sprung wagon components vibrations. The natural frequencies and damping ratios for nine first modes (seven modes for carbody and three for bogies) are listed in Table 16. As expected, frame bounce would be hardly stimulated. The difference between frame yaw and car body yaw modes, which is about 10.3 Hz for three wagon types is satisfactory. Also, lower and upper sway frequencies are between 0.6 and 1.5 Hz, which is in the acceptable range [4].

The wagon stability on the straight ideal track is investigated by evaluating the hunting critical speed. The open loop stability analysis for each wagon is executed in a velocity range (for instance 30 to 55 m/s), to obtain the eigenvalues of the system. Critical velocities are determined from the root locus results.

Table 15. Preloads results on wagon suspension elements, N

Preload results of Spring Sets (Loaded Cond.)			Wagon type					
			(I)		(II)		(III)	
			Sim.	Calc.	Sim.	Calc.	Sim.	Calc.
Secondary Springs	Front Bogie	Left Bearer	78040.5	77937.5	64885.1	63951.1	80730.6	80949.4
		Right Bearer	77344.6	77447.6	64401.0	63278.6	81446.5	81350.0
	Rear Bogie	Left Bearer	78863.1	78775.1	73472.1	74506.9	73346.4	73104.4
		Right Bearer	78191.9	78279.9	72701.8	73723.4	73346.5	73466.2
Primary Springs	Front Bogie	Left Bearer	86685.8	86416.4	69622.1	72489.1	88812.0	89195.3
		Right Bearer	84911.4	85668.7	67307.6	71440.6	89797.6	89804.1
	Rear Bogie	Left Bearer	88050.6	87254.9	86854.9	83065.9	81604.9	81357.6
		Right Bearer	86192.1	86500.0	85075.4	81864.4	82055.6	81913.0

Table 16. Natural frequencies and damping ratios for 9 mode shapes

Mode No.	Mode Description	Natural Frequency (HZ)			Damping Ratio %		
	Wagon Type	(I)	(II)	(III)	(I)	(II)	(III)
1	Upper Sway	0.64	0.69	0.66	20.1	18.5	18.00
2	Car-body Yaw	0.83	0.83	0.83	8.9	9.25	9.00
3	Car-body Pitch	1.00	1.02	1.05	31.0	29.1	29.00
4	Car-body Bounce	1.40	1.45	1.41	16.0	19	20.00
5	Lower Sway	1.48	1.61	1.52	31.0	32	38.00
6	Hunting In Phase	9.50	9.70	10.01	34.0	34	30.00
7	Hunting Out of Phase	9.55	9.60	10.00	33.8	33.8	29.00
8	Frame Yaw	11.1	11.1	11.0	1.8	1.9	2.00
9	Frame Bounce	13.6	13.5	14.0	46.5	46.8	46.90

The eigenvalues with positive real parts (negative damping ratio) indicate the instability at that velocity. As an example, the root locus for loaded wagon (II) is presented in Figure 6. The effect of frictional side bearers on critical speed is studied by comparing the results with bogie model without side bearers where, carbody is directly connected to secondary suspension. The critical speed results for three wagon types on straight track are listed in Table 17. The sim1 refers to the results for bogie with frictional side bearers and sim2 denotes the results for bogie without side bearers. It is obvious that the smallest critical speed for bogie with side bearers belongs to loaded wagon (II), with 130 Km/h, while loaded WEC has the largest speed (147 Km/h).

Conventionally, the practical critical speed is considered 10% below the theoretical critical one. That is, at service on straight track, the train should run below 117 Km/h, but, some other provisions should be considered when the wagons are running in curved or irregular track. The critical speeds for wagons with and without side bearers in a curve with R= 310 m are also listed in Table 18. It is induced that on the curved track, the train should run below 102 Km/h, the theoretical critical speed.

Table 17. Critical speed on straight track

Wagon Type	(I)		(II)		(III)	
Critical Velocity	sim1	sim2	sim1	sim2	sim1	sim2
	142.2	136.5	130.0	120.7	139.2	132.8

Table 18. Critical speed on curved track R = 310 m, km/ h

Wagon Type	(I)		(II)		(III)	
Critical Velocity	sim1	sim2	sim1	sim2	sim1	sim2
	115.5	105.3	114.0	96.8	117.0	105.8

A. Tracks and Lines for Running Simulation

If a vehicle is dynamically stable, all displacement and forces should be damped on a straight and smooth track, after leaving the section with irregularities. Three track cases have been distinguished, that meet a standard of benchmark for wagon simulations in the worst conditions of the test line. The standard tracks are as follows

Track 1- The first track consists of a single irregularity. This test is used by ERRI [11], as hunting assessment irregularity. The track gauge is widened about 3.5 mm (out of phase) from given profile and deepens about 5 mm (in phase), at the irregularity. The irregularity is located about 60 m from start of track and is not the same for left and right rails. The length of whole track is 350 m.

Track 2- This is a straight track consisting of a distributed irregularity, used by ERRI [11], as hunting irregularity for ride quality assessment. The track starts with 60 m of smooth track, followed by a 500 m of measured irregularity (both vertical and lateral) then there is a smooth and straight track. The track gauge is widened about 5 mm from given profile and deepened about 6 mm in place of irregularity.

Track 3- This track with a constant gauge, two straight and a curved section without any irregularity is used for safety examination of wagons. This track is extracted from ADAMS/Rail Benchmark. Radius of small curve is 310 m. The length of whole track is 1500 m. The preliminary straight track is 50 m and the length of curve is about 750 m. The maximum super elevation is 12 mm.

B. Dynamic Analysis

Dynamic analysis has been achieved on mentioned tracks. The results required to assess the dynamic performance are presented, such as, longitudinal, vertical and lateral accelerations on bogie frame and car body, ride quality in vertical and lateral directions, derailment ratios in each wheel set. The running velocity on straight track is well above the nominal velocity (about 33 m/s = 118.5 Km/h). On curved track of radius 310 m (L= 2 Km), the running velocity is well below the minimum allowable velocity on this track (about 30 m/s =108 Km/h). UIC 60 rail profile and S1002 wheel profile, available in the ADAMS Rail system library have been used. The number of integration steps mainly depends on the wagon type, running speed, the rail and track conditions. Generally, the number of integration steps are chosen for the best accuracy and therefore convergence criteria has been checked for all of the simulations. For more confidence, the responses on all bogie components like wheelsets and bolster have also been observed.

Generally, there are two methods for assessment of wagon safety against derailment. A flange climb situation emerges when the ratio of lateral contact force to the normal one reaches the maximum value for each wheel, which is described by Gilchrist and Brickle [14]. The limiting ratio (Y/Q) is a function of the flange angle and the friction coefficient. The friction coefficient between wheel and rail is assumed 0.36. The time histories of vertical and lateral forces between wheels and rail, as well as Y/Q are calculated by ADAMS/Rail.

For curve radius above 250 m, this ratio should be less than 0.8 [7]. The critical derailment factor results for front and rear bogie wheelsets of wagons (II) and (III) on Track 1 are given in Figures 7 and 8. Similar results for wagon (II) on Track 2 are shown in Figure 9 and the results of wagon (III) on Track 3 are shown in Figures 10 and 11 in terms of front and rear axles of front and rear bogies, respectively. To avoid a crabbed graph, in Figure 10, left and right wheels on front and rear axles are illustrated separately. It is observed that in all wagon types and tracks, the derailment ratio is well below 0.8, showing running safely.

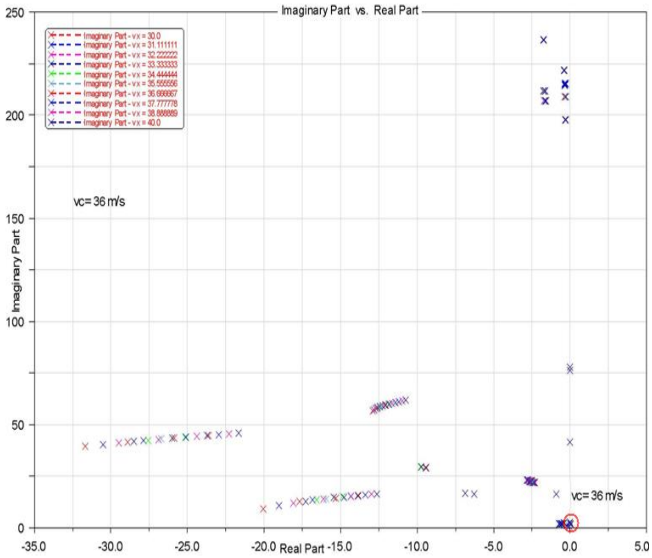

Figure 6. Root locus for wagon (II)

It is observed that for track 1, the largest derailment ratio is about 0.05, which belongs to wagon (I). Also, on track 2, wagon (II) has the largest derailment ratio, about 0.08, for track 3, the greatest derailment ratio is nearly 0.35, which belongs to wagon (III).

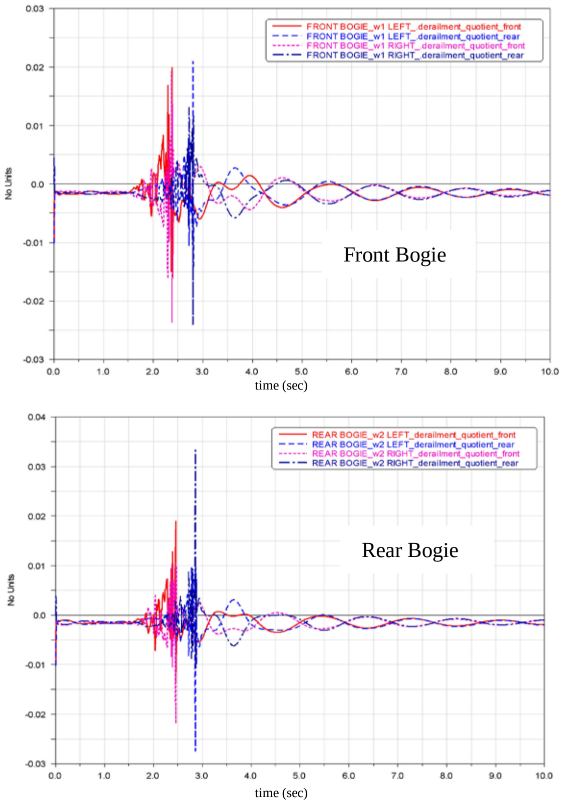

Figure 7. Derailment factor wagon (II), track 1

Another way to study running safety is comparing the vertical and lateral accelerations at the center of wagon body floor and on the bogie frame at the position of each outer wheel set with standard permissible values, [7] as

1. Lateral acceleration of bogie frame:

$(\ddot{y}_s^+)_{\lim} = 12 - \frac{M_b}{s}$, M_b is the total bogie weight in tons.

Therefore, $(\ddot{y}_s^+)_{\lim} = 12 - 1 = 11$ m/s^2.

2. Lateral acceleration of car body center:

On straight track and large radius curve: $(\ddot{y}_s^*)_{\lim} = 3$ m/s^2,

Small radius curve, $250 \text{ m} \le R \le 400 \text{ m}$: $(\ddot{y}_s^*)_{\lim} = 2.6$ m/s^2.

3. Vertical car body acceleration: $(\ddot{z}_s^*)_{\lim} = 3$ m/s^2.

Although, numerous results can be presented for accelerations of different wagon parts, only sample filtered accelerations on bogie frame and car body floor of wagons, which have critical time responses are shown in Figures 12 to 19. The lateral and vertical accelerations on the frame over axle boxes of wagon (III), on tracks 2 and 3 are given in Figures 12 and 13, which are smaller than 11 m/s^2. The lateral and vertical accelerations at the center of car body floor of wagon (III), on three tracks are presented in Figures 14, 15 and 16. The maximum acceleration approaches 0.34 m/s^2, which is below the standard limit values. This, in turn complies with the derailment safety for running in the allowable speed range.

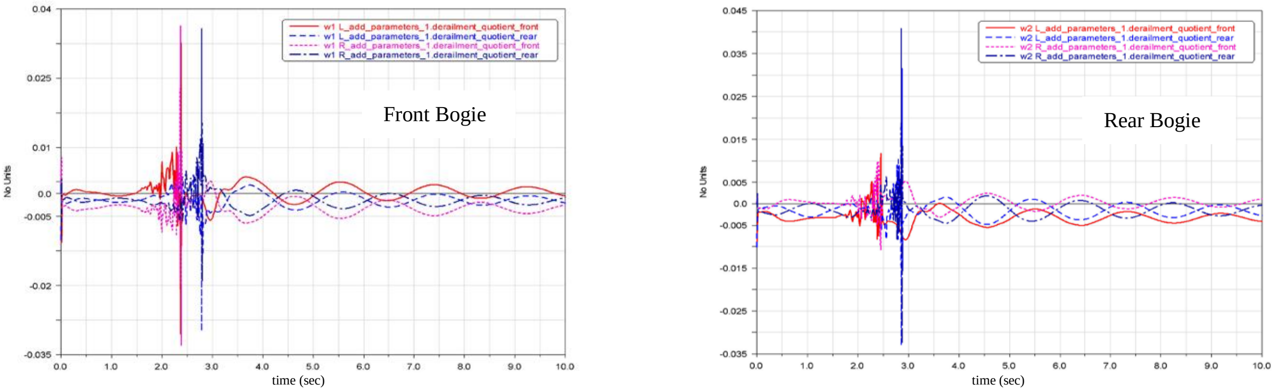

Figure 8. Derailment factor, wagon (III), track 1

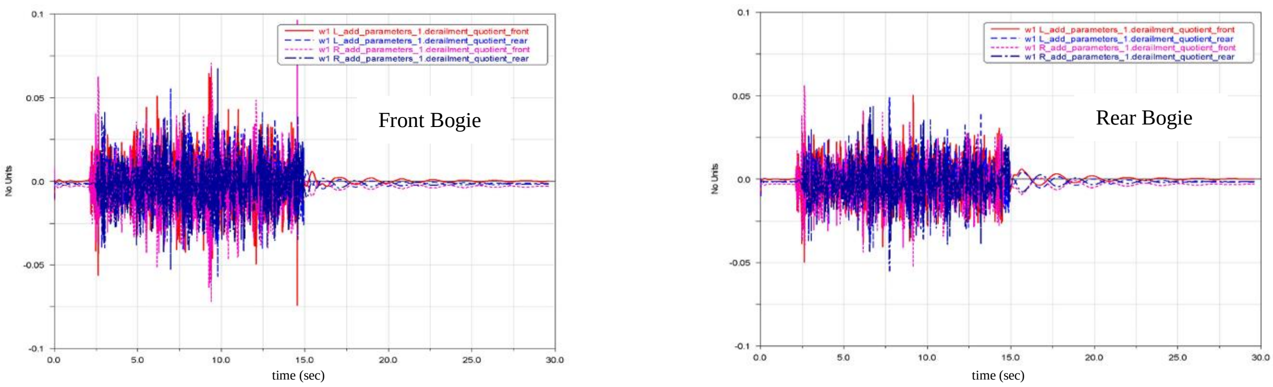

Figure 9. Derailment factor, wagon (II), track 2

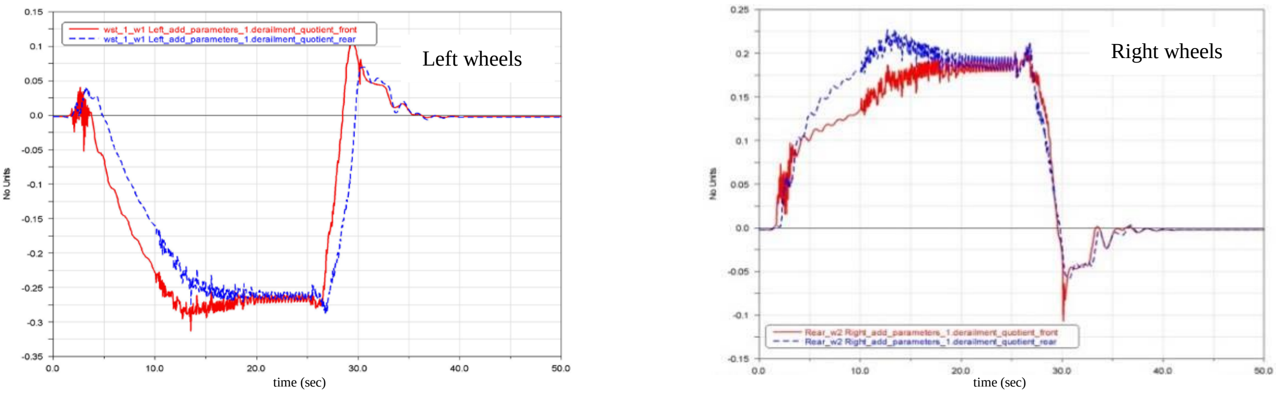

Figure 10. Derailment factor, front bogie, wagon (III), track 3

C. Ride Comfort Analysis

Comfort analysis is done on three mentioned test tracks. The lateral and vertical accelerations at the points, recommended by [7], i.e. $(\ddot{y}_s^*, \ddot{z}_s^*)$ at body floor center and lateral acceleration of frame over the axle boxes $(\ddot{y}_s^+)$ are exposed to the Fast Fourier Transform (FFT) and a proper frequency filtering range (0.1-100 Hz) is chosen. The Sperling ride index is defined as follows, [4, 7]

$$Wz = 6.67\sqrt{\int_{0.5}^{30} a^2.B^2.df} \tag{8}$$

where, $a(f)$ is the acceleration amplitude in frequency domain and $B(f)$ is a function describing the human sensitivity to vibrations, as follows:

$$B_{vert} = 58.8\sqrt{\left[\frac{1.911 f^2 + (0.25 f^2)^2}{(1-0.277 f^2)^2 + (1.563 f - 0.0368 f^3)^2}\right]} \tag{9}$$

$$B_{Later} = 1.07 B_{vert}$$

Filtered accelerations at car body centers are used for evaluation of ride quality index. Sample results for all three wagon types, with higher values, on three tracks are shown in Figures 14 to 19. It is observed that the most critical accelerations at the car body centers of wagon (I) on track 1 and for wagon (III) on track 2 are about 0.6 and 1.2 m/s^2, respectively. Ride indexes for all wagons are calculated by means of Equations (8) and (9). The results for different number of integration steps are listed in Table 19. By increasing the number of integration steps the results converge to the final values. The greatest vertical ride index belongs to (III) and then (II) on track 2. On track 3, the lateral ride index of (III) and (I) are largest ones. None of the values of these results exceed the limit values, which are defined as follows, [7]

- *Wz*= 1~2: Ride quality is excellent
- *Wz*= 2.5: Ride quality is good
- *Wz* > 3: Ride quality is bad (uncomfortable)

Another method to assess the ride quality is using the accelerations on the car body floor, over the front and rear bogie center pins. According to UIC513, [7], the standard permissible values are $(\ddot{y}_q^*)_{\lim} = 2.5$ m/s^2 and $(\ddot{z}_q^*)_{\lim} = 2.5$ m/s^2.

To assess the filtered accelerations, sample results for the worst case, i.e. wagon (III), at floor level are given in Figures 18 and 19. As it is observed, the maximum values of lateral and vertical filtered accelerations are on track (1) front: -0.6, and rear: -0.7 m/s^2; on track (2) front: 0.45, and rear: 1.3 m/s^2; which are in the acceptable range. However, the maximum values on track 3 are 2.5 and 2.6 m/s^2, which means that the lateral ride quality of (III) is not proper.

Table 19. Ride quality indexes

Track No.	Ride Index	Wagon Type					
		(I)		(II)		(III)	
n =		142	100	193	100	158	100
1	Vertical	1.98	1.83	1.92	2.07	1.99	1.81
(33 m/s)	Lateral	1.30	1.34	1.18	1.22	1.10	1.19
n =		164	130	222	130	213	130
2	Vertical	2.36	2.29	2.44	2.17	2.50	2.23
(33 m/s)	Lateral	1.82	1.66	1.78	1.56	1.79	1.71
n =		253	160	261	160	289	160
3	Vertical	1.70	1.54	1.80	1.72	1.93	1.60
(31 m/s)	Lateral	2.08	1.89	1.56	1.31	2.14	1.85

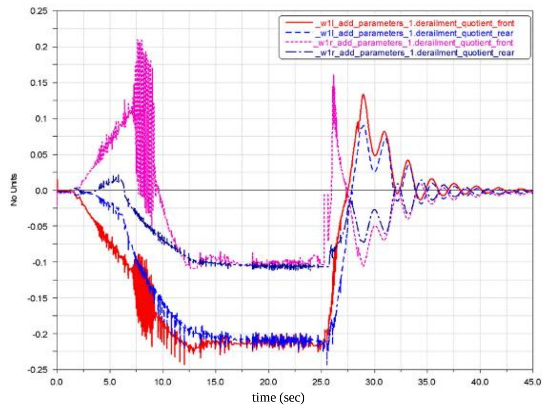

Figure 11. Derailment Factor, Rear Bogie, wagon (III), track 3

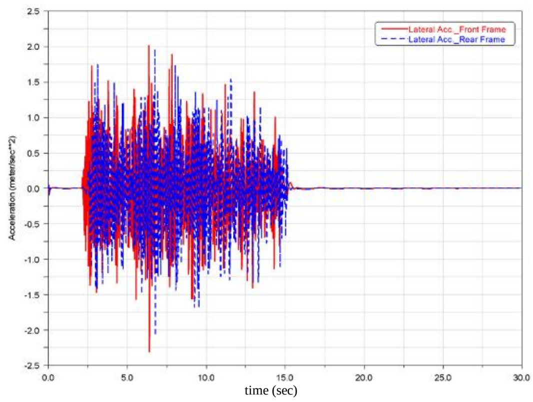

Figure 12. Filtered lateral acceleration, bogie frame, (III), track 2

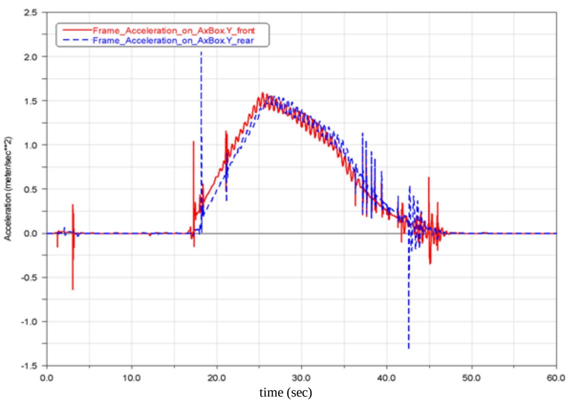

Figure 13. Filtered lateral acceleration, bogie frame, (III), track 3

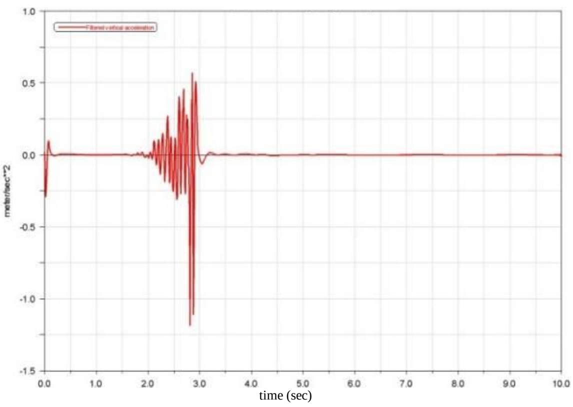

Figure 14. Filtered vertical acceleration, body center, (III), track 1

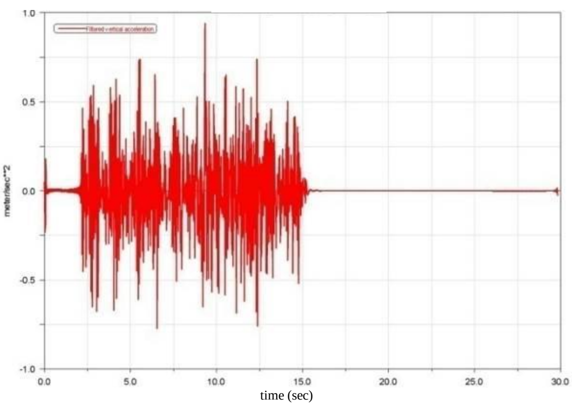

Figure 15. Filtered vertical acceleration, body center, (III), track 2

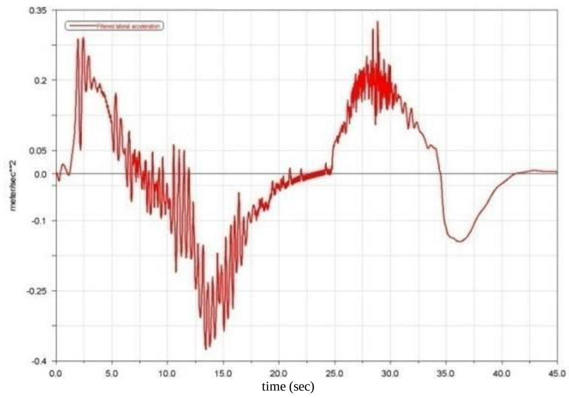

Figure 16. Filtered lateral acceleration, body center, (III), track 3

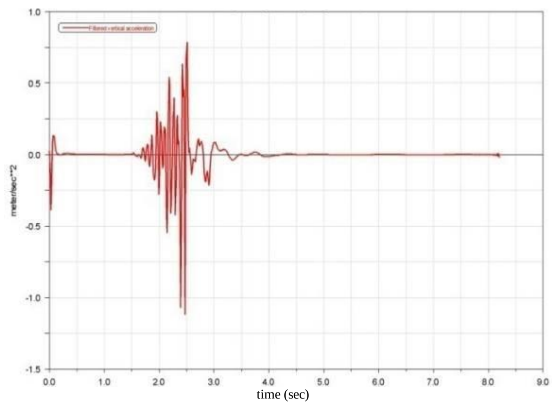

Figure 17. Filtered vertical accelerations, body center, (I), track 1

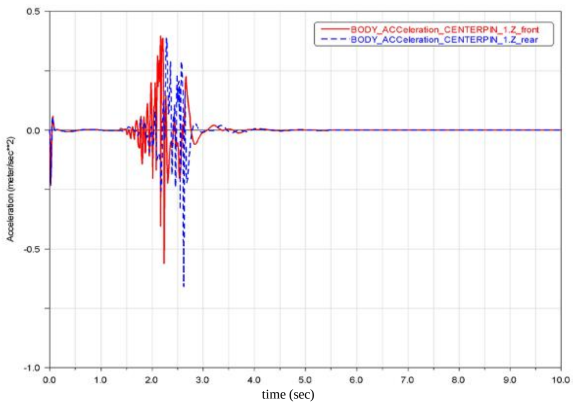

Figure 18. Filtered vertical acceleration, on bogies, (III), track 1

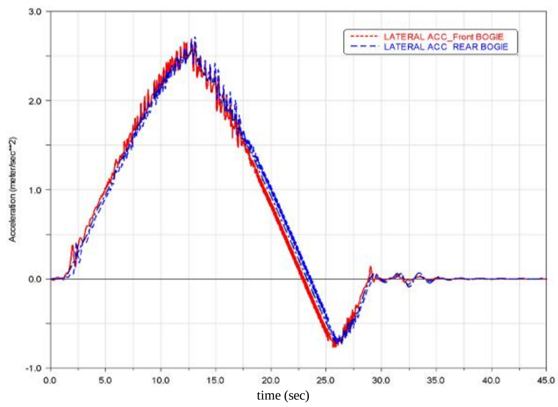

Figure 19. Filtered lateral acceleration, on bogies, WPC, track 3

V. CONCLUSIONS

The springs of bogie MD52 are re-designed for three new wagons, based on maximum shear stresses and structural constraints of bogie suspension. The calculated loads for suspension elements are verified by using ADAMS/Rail. Then, dynamic simulation results, such as eigenvalues and critical velocity are obtained. Secondly, the dynamic performance parameters, vertical and lateral accelerations on bogie frame and car body floor, Sperling ride quality in different directions and derailment factors on irregular straight and curved tracks are investigated. All the results are inside the standard allowable ranges. The following conclusions can be made from this study:

- According to the stability analysis of dynamic system (with at least 73 D.O.F) for different wagon types, the maximum allowable train speed was obtained 117 Km/h for straight line and 102 Km/h for a small radius curve.
- According to dynamic responses of wagon components, the designed springs provide a stable dynamic behavior.
- From derailment analysis, it is concluded that all wagon types are safe against derailment in all tracks. Not only, in curved, but also, in straight track with rough irregularities (track 2) the derailment ratio increases.
- The simulation results show that major suspension parameters are suitable and in the allowable design range.

The models of bolster, frictional side bearers and clearances between bolster, frame and axle boxes in bogie are created as new elements in ADAMS/ Rail, which give more precise simulation results and the effect of these elements has not been investigated by others previously.

REFERENCES

[1] "Bombardier Transportation", www.bombardier.com.
[2] S. Sharma, A. Singh, R. Bhushan, "Design Development of Meter Gauge Bogie", 16th European Mechanical Dynamics Users' Conference, pp. 114-126, Nov. 2001.
[3] German institute for Standardization, "Helical Springs Made of Round Wire and Rod", DIN 2089, 1984.
[4] V.K. Grag, R.V. Dukkipati, "Dynamics of Railway Vehicle System", Academic Press, 1994.
[5] A.H. Wickens, "The Dynamics of Railway Vehicles - from Stephenson to Carter", Proceedings of the Institute of Mechanical Engineering, Part F: J. Rail and Rapid Transit, Vol. 212, No. 3, pp. 209-217, 1966.
[6] A.H. Wickens, "Fundamentals of Rail Vehicle Dynamics: Guidance and Stability", Swets & Zeitlinger B.V., Lisse, The Netherlands, 2003.
[7] International Union of Railways, "Testing and Approval of Railway Vehicles from the Point of View of their Dynamic Behavior - Safety - Ride Quality", UIC 513, 1st Edition, UIC 518, 2nd Edition, 2003.
[8] Sh. Budynas-Nisbett, "Mechanical Engineering Design", McGraw-Hill, 8th Edition, 2006.
[9] German institute for Standardization, "Hot rolled Steels for Springs Suitable for Quenching and tempering - Technical Delivery Conditions", DIN 17221, 1988.
[10] International Union of Railway, "Technical Specification for the Supply of Helical Compression Spring, Hot or Cold Coiled, for Tractive and Trailing Stock", UIC 822, 5th Edition, 2003.
[11] European Rail Research Institute, "B176/3 Benchmark Problems - Results and Assessment", B176/ DT 290, Utrecht, 1993
[12] International Union of Railways. "Trailing stock: wheels and wheelsets", UIC 510-2, 4th edition, 2004.
[13] N. Bosso, A. Gugliotta A. Soma, "Simulation of a Freight Bogie with Friction Damper", 5th ADAMS/Rail Users Conference, Harlem, 2000.
[14] A.O. Gilchrist, B.V. Brickle, "A Re-Examination of the Proneness to Derailment of a Railway Wheelset", J. Mech. Sci., Vol. 18, No. 3, pp. 131-141, 1976.

BIOGRAPHY

Mohammad Reza Saviz was born in Tehran, Iran in 1973. He received the Ph.D. degree on Mechanics from Amirkabir Technical University, Tehran, Iran in 2008. Now, he is a member of scientific board in Mechanical Engineering Department, Azarbaijan Shahid Madani University, Tabriz, Iran.