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Computational study on the effect of a conical spring on handling of buses at low speed

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Abstract. Handling and ride characteristic are dependent to a large extent on the characteristic of a vehicle's suspension system. This work explores the effect of the use of conical spring in place of conventional cylindrical profiled helical spring design in the handling of a bus at low speeds through full vehicle multi-body simulations. The bus was modelled using standard template available in ADAMS™ software package. The vehicle inertial properties were verified against properties in literature. The conventional spring characteristic (L) from ADAMS™ database was taken as reference and compared it with a non-linear characteristic (NL) based on literature data. The planned maneuover was to execute a right turn based on standard road dimension inputs from IRC 86:1983 at a constant speed of 30 km/hour with acceleration controlled by software module. Chassis displacements, displacements of spring were tracked to understand handling and ride quality. The variation of chassis displacements showed a significant improvement in ride characteristic of vehicle with most vibrations being damped in NL at time lower than the L characteristic suspension. All through the study, lateral acceleration was well within the rollover threshold and tire interaction forces did not exhibit any significant changes.

Keywords: Non-Linear Spring, Progressive Spring, Conical Spring, Bus, multi-body dynamic simulation

1 Introduction

The handling of a heavy vehicle has remained the focus of studies since the early 1950s and more documented research is available in public domain from tests conducted in USA from 1960s. Vehicles were tested in a standardised test track and the significant parameter influencing the dynamic stability, handling and ride were identified. The data collected as part of these studies formed the basis of development of an integrated software tools using multi body technique to analyse vehicle handing. Of the parameters identified, the suspension systems are known to influence both the ride characteristic and the handling of the vehicles [1].

The handling of the vehicle is dependent on the suspension subsystem, as it connects the vehicles sprung mass with un-sprung mass which comprised of the tyre, thus resulting in loading of tyres. Modern development have identified the use of conical spring in the place

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of conventional helical spring as the conical spring having potential of more favourable characteristics than helical spring like desirable buckling characteristic, with a compromise on space occupied in packaging of vehicle. The Automotive Research Association of India (ARAI) is the nodal body framing rules for vehicle standardization in India and roads in India are constructed based on standards developed by the Indian Road Congress (IRC). The rules framed by these two nodal bodies form major input to vehicle design limitations for vehicles in India. The objective of this work was to examine the variations of the behaviour of a reference vehicle when the conventional spring is replaced with conical spring behaviour. In order to compare the results, the variations in the geometry of the springs is limited by modelling two springs closer to one another and a standardized road manoeuvre was chosen.

2 Nomenclature

IRC	-	Indian Road Congress
L	-	Linear Suspension behaviour
NL	-	Linear Suspension behaviour
FN	-	Normal Force on tyre at the tyre-road interaction point
FL	-	Lateral Force on tyres at the tyre-road interaction point
FL	-	Front Left tyre to driver
FR	-	Front Right tyre to driver
RL	-	Rear Left tyre to driver
RR	-	Rear Right tyre to driver

3 Methodology

A multi-body simulation of a bus model using ADAMS™ was planned for this study. A standard template of a bus readily available in ADAMS™ was modified with the properties from literature [2]. The bus model was assumed to be un-laden.

3.1 Suspension Model

The vehicle has an independent suspension, and the vehicle inertial properties were altered to values as per literature [2] for conventional vehicles. Due to this change a theoretical springs were needed to simulate the behaviour of the linear leaf spring system. The pre-load in the system is 36kN in the front axle and 17671N in the rear axle.

3.1.1 Mathematical model for Linear Behaviour:

This type of spring predominately obeys the equation, $P = k*y$, where y is the deflection of the spring, P is the axial load on the spring and k is the stiffness of the spring, as a result Force vs. Displacement plot for this spring is a straight line as show in Fig. **1**. The spring was designed using the formulae given in[3] by using leaf spring material as AISI 5160 alloy steel[4].

For the shown plot it was noted that the stiffness of the spring was much closer to that of the required spring rate. For any force exceeding the force of 82kN it noted that the deformation is large, in order to consider this behaviour, the following plot was programmed into the system.

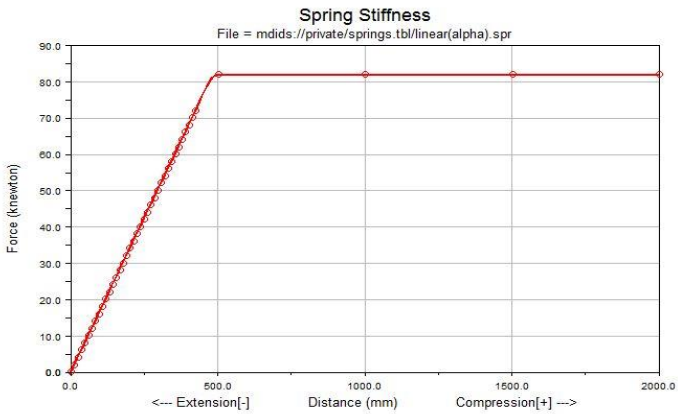

Fig. 1 Linear spring characteristic

3.1.2 Mathematical model for Non-Linear behaviour

The Non-Linear spring characteristic, Force vs. Displacement plot is a curved plot as show in the Figure 2. The spring was designed using the formulae given in [5] by using the material in reference [4]. Similarly like in the case of Linear system the model was capped at 82kN. To keep the variation to a minimal limit both the L and NL spring design were made with the same material and the diameters of the spring was close by.

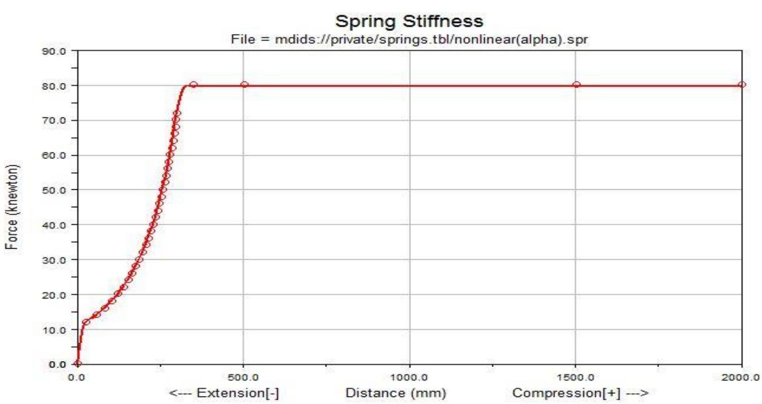

Fig. 2 Non-Linear Spring Characteristic

3.2 Vehicle Description

A full vehicle model was input in ADAMSTM. The vehicle had an engine with gear box attached, brakes in front and rear wheels. An overview of the vehicle model used is summarized in Table 1.

Table 1. Vehicle Specification

Engine	
Type	Rear Mounted Engine
Idle RPM	1000
Max. RPM	2200
Brakes	
Max. Front Brake Torque	5880Nm
Max. Rear Brake Torque	3670Nm
Tyre Specifications	
Front tyre radius	0.32m
Rear tyre radius	0.32m
Tyre file	msc truck pac 2002:tir
Power Train	
Gear ratio	6 speed configuration

3.3 Simulation Conditions

The track of the test is designed in such a way so as to allow the settling time for bus to settle down to the applied linear acceleration and to stabilise its steering characteristics, after a steady state is reached, the bus is turned to a radius specified as per IRC guidelines [6]. The analysis was carried on for 30 seconds, which was in three phases as shown in Figure 3.
Phase I: consists of a straight line driving of the vehicle for 10 seconds. The objective of this segment is to ensure that the entire vehicle reaches a steady state linear motion, where in there are no unnecessary fluctuations in the output parameters.
Transition Phase: It is in this phase that an external steering input is given to the steering wheel which causes the bus to turn. In this phase all the fluctuations are bound to happen and the reading for peaks and falls are present.
Phase II: In this phase the vehicle has turned into the turn and has stabilised to the new condition of motion, thus causing a nearly steady state values in all the measured output parameters. The IRC-86:1983, states the speed of the vehicle for the dimension and purpose of road.

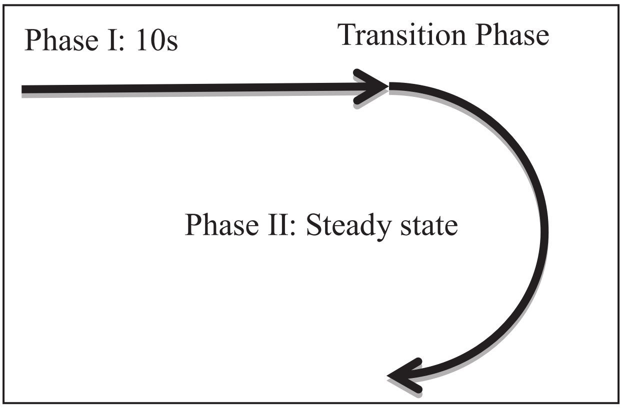

Fig. 3. Description of right hand turn manoeuvre

4 Results and Discussion

In order to understand the variation caused by changing the spring, all important suspension related parameter were studied in steering, chassis, springs and tires as per reference.

4.1 Chassis and Spring Displacement:

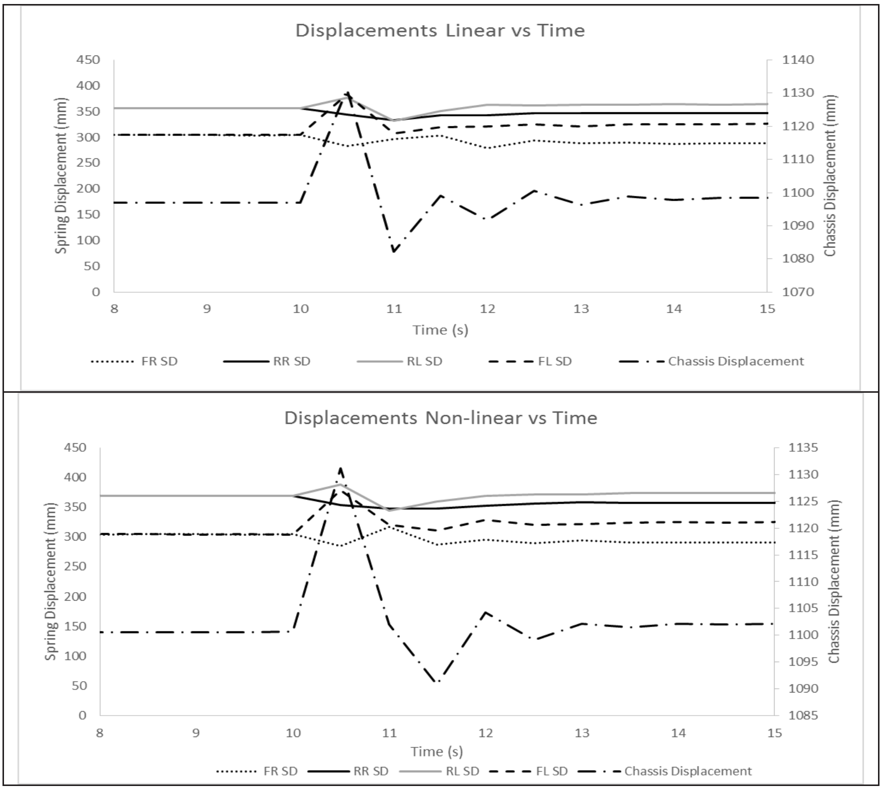

Fig. 4. Variation of chassis and spring displacements for linear and non-linear with respect to time.

The results summarized in Figure 4 indicate that, after the commencement of the turn, the FR and the RR spring displacements are lower that the FL and the RL spring displacements. This phenomenon implies weight transfer to the right, which is an expected behaviour of bus taking a turn. From the Chassis displacement plot it is observed that the Non-Linear model settles at 13.5s for an excitation at 10s whereas the Linear model settles at 14s for the same, moreover form the plot is noted that after the excitation there are 7 peaks whereas in Non-Linear model there are only 5 peaks of excitation before settling. The amplitude of initial excitation however has not changed between the models but the successive excitation amplitudes have reduced considerably.

From the above two graphs it is clear that there is no significant difference in the amplitude of the start of the vibration, but in the case of NL model, the damping was faster. Post turn fluctuation in the individual spring displacement is lower in NL model which could result in the better damping of the overall chassis system. Due to the graded stiffness nature of the NL spring the overall ground clearance of the vehicle is dropped down by a margin of 2mm.

4.2 Chassis Lateral Acceleration:

The maximum lateral acceleration observed was 0.478m/s^2. From literature, the limit for a static rollover condition was 0.4g (3.924m/s^2) acceleration [7]. It was be noted that the variation of difference in the variation of L to NL model is too low to say that their behaviour is different form one another. From the Figure 5, the lateral acceleration in all three phases was within threshold.

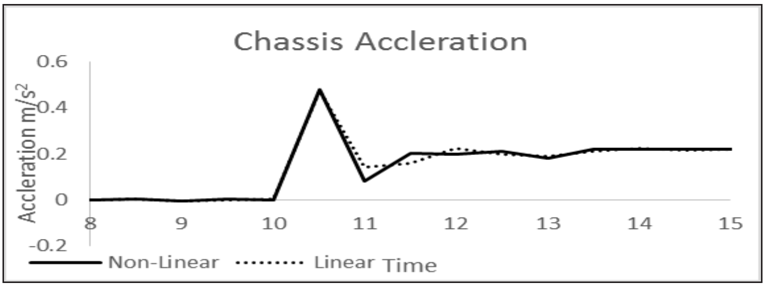

Fig. 5. Chassis lateral acceleration variation during manoeuvre.

4.3 Normal Force and Lateral Force:

Lateral force and Normal forces for the model was studied at the tyre contact points of the vehicle. Knowing the interaction of forces at each wheels helps in determining the motion of each wheel thus the motion of the overall vehicle.

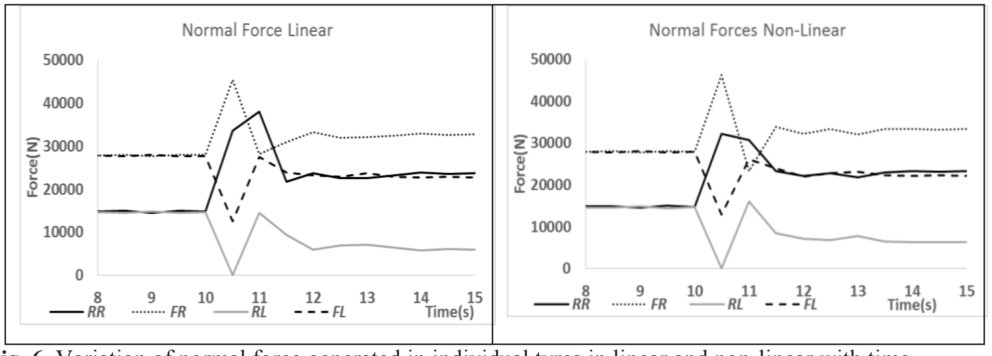

Fig. 6. Variation of normal force generated in individual tyres in linear and non-linear with time

From Figure 6, it was noted form the FR and RR wheels in both the models have a higher value than their counterparts, this is a proof that the vehicle model is in accordance to the physical models. It noted form the above graphs that the RL and FR wheels have a higher Normal force and there is a marginal drop in the RR and the FL wheels in the non-linear model compared to the linear mode. It was observed that normal wheel forces did not change direction (i.e. no lift-off or Rollover).

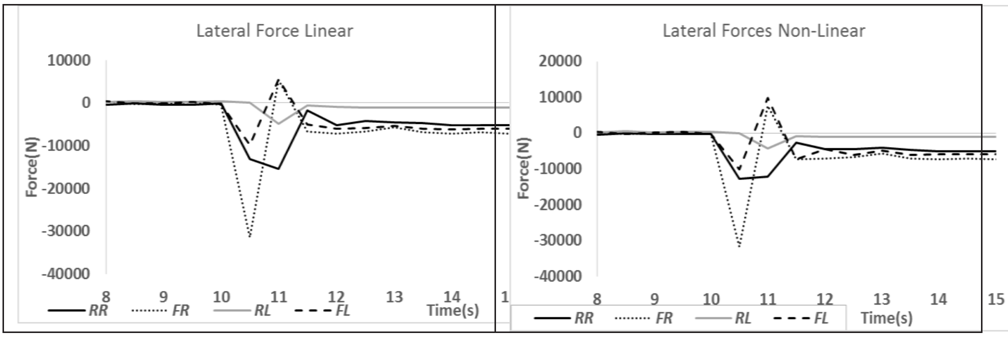

Fig. 7. Variation of lateral force developed in individual tyres in linear and non-linear with time.

Comparing the lateral force graphs (Figure 7) of both the models it is observed that there is no significant changes in the lateral force patterns. The lateral forces in the FR and FL reach a positive value before returning to the negative domain, this is an indication for that fact that the vehicles steering is adjusted to ensure the vehicle is in the proper path.

Overall the lateral force generation at the tyre in both the models is relatively the same. To study tyre loads differently, the ratio between the Lateral Force and Normal Force is studied for determining the overall handling characteristics of the tyre as per reference[8], summarized in Figure 8.

Even for significant changes in normal forces the lateral force to normal force ratio relatively remains constant throughout the study. Form the graphs it is noted that the FL and RR diagonal has a higher lateral force compared to RL and FR diagonal in case of the Non-linear model.

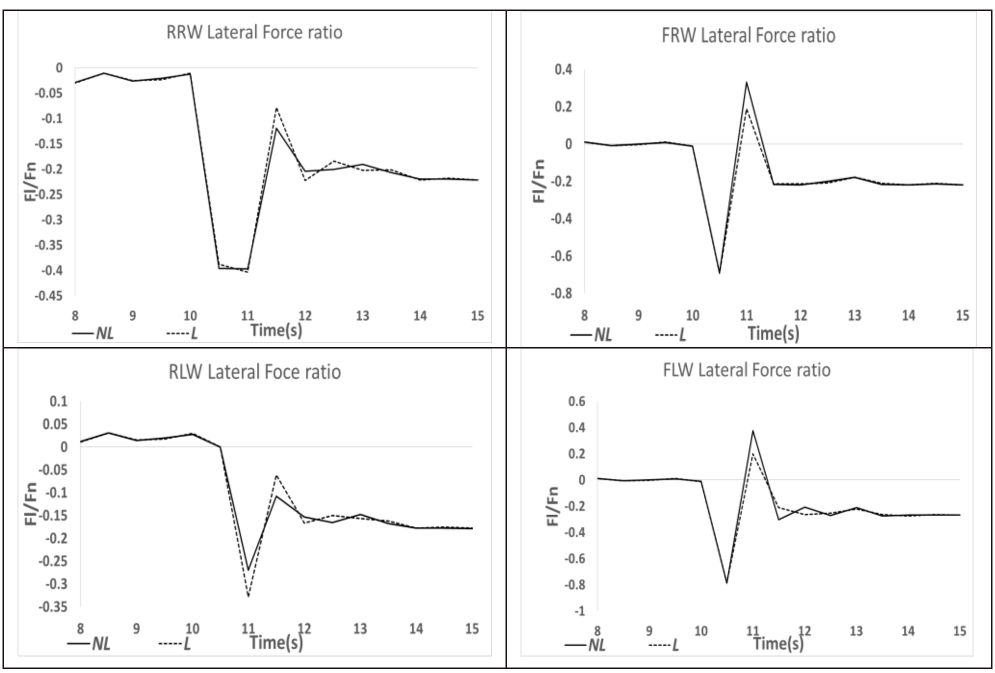

Fig. 8 FL/FN ratios comparison L vs. NL.

4.4 Limitations:

The models have been evaluated for low speed turn with lateral acceleration well below roll-over threshold to examine a typical driving under low speed of vehicle. No extreme manoeuvres were considered. The spring properties have been assumed for a generic material and it remains valid for a comparison study.

5 Conclusions

It was concluded that changing of cylindrical to conical profile spring provided a smoother ride without compromising on the levels of handling and safety as provided in cylindrical profile spring.

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