Published by :
http://www.ijert.org
International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 8 Issue 07, July-2019

Sensitivity of Dynamics of a Locomotive Suspension to Axle Loading

Sumit Hemant Pitale
Department of Mechanical Engineering
RV College of Engineering
Bengaluru, India

Shivaraj B W
Assistant Professor
Department of Mechanical Engineering
RV College of Engineering
Bengaluru, India

Ravi Kumar Dhulipalla
Engineering Technical Leader
GE-Transportation, A Wabtec company
Bengaluru, India

***Abstract*— The dynamics study for a locomotive mainly revolves around safety and comfort parameters, which include wheel rail lateral forces, derailment quotient, acceleration and ride index, defining the performance of the locomotive. The objective of this project was to achieve modularity in the suspension system of the locomotive to increase the number of applications for a single suspension. A locomotive model with three axles per truck and four spring nests per axle was developed in ADAMS/Vi-Rail. The axle load of the locomotive was varied to determine its effect on the performance parameters. The axle load was varied between 12 tons to 18 tons with an interval of 1.5 tons (12, 13.5, 15, 16.5, 18), for a PS-SS combination of 400-140 N/mm stiffness and dynamic analysis was performed on a six-axle locomotive model using ADAMS/Vi-Rail. Also, the critical velocities for each axle load were analyzed by varying the locomotive speed between 160 to 260 kmph at each axle load condition to perform stability analysis. The analysis results suggest that the lateral forces generated in the wheelsets increase by 2 KN as the axle load increases from 12 to 18 tons. The derailment quotient, acceleration and ride index number show a decreasing trend with increase in the axle load, which indicates that the suspension has better ride quality and comfort at higher axle loads of 16 to 18 tons. In stability analysis, as the load on the spring increases from 12 tons to 18 tons, the value of critical velocity of the locomotive comes down from 220 Kmph for 12 tons to 160 kmph for 18 tons.**

Keywords*— Sensitivity analysis, ADAMS/Vi-Rail, ride index, stability analysis, lateral forces, derailment coefficient, acceleration

I. INTRODUCTION

A spring suspension is a need on rail vehicles because the running rails and the track are not perfectly smooth, and the wheels are also not perfectly round (surface irregularities, wheels not perfectly round, imbalance of mass and inertial forces). Such irregularities on the wheel and the tracks lead to the resulting disturbing forces being transferred to locomotive components. Also, the wheel rail interaction and conical profile causes the system to continuously shift from dynamic equilibrium and thus creating lateral oscillation.

The aim of the suspension system is to provide optimal performance by achieving values of ride quality and other parameters as close as possible to the locomotive design solutions. The ride quality and ride comfort concern equalizing the distribution of weight loads between the wheels, minimizing the dynamic forces caused by the interaction between wheels and rails, realizing the maximum possible tractive and brake efforts for the locomotive/s to haul and control the largest possible train and to minimize and dampen the dynamic forces and natural oscillations received from the train under traction and braking. In addition, the suspension design should create safe and comfortable working conditions for the locomotive crew and minimize adverse impacts on the equipment placed on the locomotive. Suspension components involve:

- Primary spring
- Secondary spring
- Primary/Vertical Damper
- Lateral Damper
- Yaw Damper

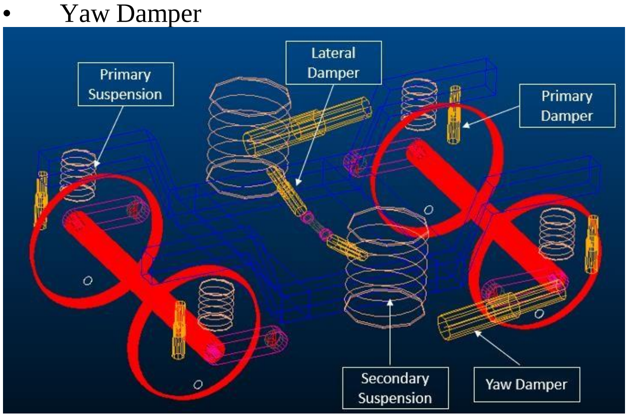

Figure 1. Components in locomotive suspension system
(Source: ADAMS/ Vi-Rail software library)

II. LITERATURE SURVEY ON DYNAMIC ANALYSIS

The vibrations in the railway locomotives becomes complex because of the effect of suspension condition, wheel profile, condition of track sections, including rail profile, rail irregularities, cant and curvature. In the literature, the Sperling ride index is calculated using RMS accelerations to evaluate the ride comfort. The ride index value was improved by 41% at speed of 60 m/s [1]. Lower is the value of the ride index, better is the ride quality of the locomotive. It improves the effectiveness of the suspension system. The wire diameter of the spring is an important parameter for optimization of the ride index of the locomotive. In the literature, ADAMS/Rail software is used to find optimum wire diameter values of springs for ride index minimization. The optimum set of spring wire diameter was found for snubber, outer and inner springs of the suspension system [2]. A good truck design of a locomotive ensures ride comfort and lateral stability during cornering at high speeds. The two parameters ride comfort and lateral stability including hunting and derailment are sensitive to load distribution in dynamic conditions. Improper

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 8 Issue 07, July-2019

distribution of the load and arrangement of primary suspension components are the causes for failure of suspension springs and shock absorbers of the locomotive. Failure of the composite spring assembly was analyzed by applying the forces obtained from dynamic analysis. The critical loading condition was determined at a hunting speed of 132km/h on a curved track [3]. Another literature for dynamic study is based on the modal analysis of the railway bogie, which is carried out using ANSYS software. The study is made on the vibrations which are caused in the bogie and the first few natural frequencies are extracted [4]. A few of the literatures also focus on the recent developments in the dynamics of railway vehicle. The railway vehicle dynamics is studied along with the performance characteristics such as lateral stability, curving, multibody simulation, wheel to track interaction, ride quality and comfort. The advancements in dynamic analysis such as 2D and 3D theories for wheel-to-track interaction is also discussed. Detailed analysis of ride quality and comfort evaluation methods is presented in this paper [5]. Many more new ideas to improve the suspension system of the locomotive are studied in various other papers. One of the journals titled, "A variable stiffness and damping suspension system for trains" briefs about an idea of making the spring and damper coefficients as variable parameter which can change according to the load requirement on the system [6].

III. SENSITIVITY ANALYSIS

Sensitivity analysis deals with studying the effect of change in the locomotive parameter on the dynamic parameters in suspension design. Sensitivity is the behaviour of the dynamic parameters based on the change in the locomotive parameter. In designing of the suspension system for the railway locomotive, the effect of change in axle load is studied on the dynamic parameters like lateral forces, accelerations, derailment coefficient and the ride index. The change in axle load deals with the load acting on the spring system, which has its effect on the suspension components. Hence, the spring behaviour also changes based on the change in axle load. As the load increases the force at the interface of the wheel and the rail is increased, which affects the lateral forces and the derailment coefficient.

For analysis purpose the simulation model used is a ADAMS/Rail model. Adams model mainly consist of various rigid components like carbody, frame, wheels, various suspension components, traction components and joints. Whole model is built based on coordinate references. The model starts with construction of hardpoint and construction frames which acts as a coordinate reference for each component. Various components are connected by the means of I markers and J markers. [7] The present model consists of one carbody, two truck frames, six traction motor assemblies and six wheelsets. The trucks are mounted to the carbody by the means of traction pins separated by distance 15.504 m. Also, each wheelset is at 3.6 m distance from each other. The evaluation of lateral force and derailment takes place at the wheel rail contacts whereas the acceleration and ride index are calculated on the traction pin of each truck.

In the initial model the locomotive weight is 90 tons, which has an axle load of 15 tons. The carbody weight is 60 tons, with two frames weighing 4.8 tons each, each wheelset of 1200 kgs and the traction motor assembly each of 2200 kgs. The primary spring of stiffness 400 N/mm, secondary spring of stiffness 14000 N/mm, primary damping of 70 N-s/mm, and lateral damping of 100 N-s/mm are used in the model are used in the model. Figure 2 shows the model built in ADAMS/Rail software with all the mentioned specifications of the various components.

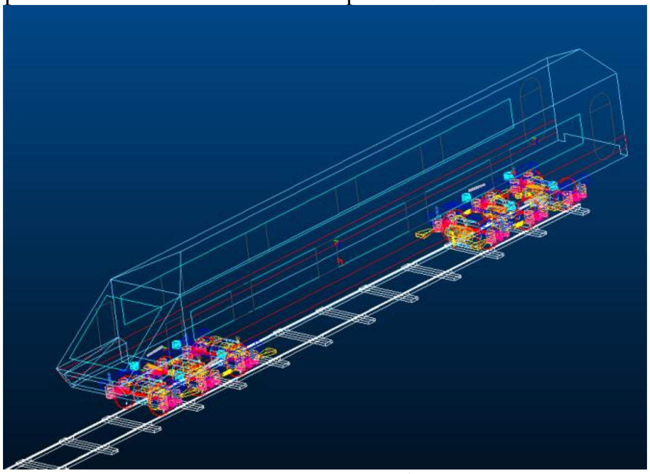

Figure. 2. ADAMS/Rail model of a locomotive

A. *Inputs and Outputs*

Dynamic analysis requires several inputs into the simulation model. Firstly, the initial velocity and the simulation time are entered. Simulation time is calculated by dividing the total length of the track by the locomotive initial speed. Next is the track file which is constructed using the alignment and unevenness data of measured track data on both left and right-side rails. These alignment and unevenness are the lateral (Y) and vertical (Z) irregularities respectively observed from desired profile in the track files. Once the track file is selected, the contact configuration is given, which specifies the friction coefficient between the track and the wheel at the interface.

The sensitivity analysis was performed for different values of axle load of 12 tons, 13.5 tons, 15 tons, 16.5 tons and 18 tons acting on the springs. In the simulation model the axle load is changed by changing the sprung mass of the locomotive which constitutes the car body weight and the truck frame weight. So, when the iteration in the axle load is done the carbody weight and the truck frame weight is changed in proportion. Here, in the simulation runs the speed of the locomotive is kept constant at 110 km/hour i.e. 30.55 m/sec. Also, the suspension system of the locomotive is kept constant at primary spring stiffness of 400 N/mm and secondary spring stiffness of 140 N/mm. Class 5 track file, which is a 2 km straight track with vertical and horizontal irregularities is chosen for this analysis. The simulation end time is calculated as 33 seconds.

In the iterations of the axle load, the carbody and frame weights are changed in proportions based on the initial model. For example, the carbody weight is 67% of the total locomotive weight. So, for 12 tons axle load, the locomotive weight becomes six times the axle load, i.e. 72 tons. Now, 67% of 72 tons is 48 tons. Hence, the carbody weight will be 48 tons. Similarly, for the frame weight, the ratio is 5.33% of the locomotive weight per frame. Hence, weight of one frame will be 5.33% of 72 tons, i.e. 3.84 tons. So, the frame weight in the simulation model will be changed to 3840 kgs.

For each iteration of the axle load, the same procedure is followed to change the carbody and frame weight in the model.

TABLE I: INPUT MATRIX FOR SENSITIVITY ANALYSIS

Axle load (Tons)	Locomotive Wt. (Tons)	Carbody Weight (KGS)	Weight of one Truck Frame (KGS)
12	72	48000	3840
13.5	81	54000	4320
15	90	60000	4800
16.5	99	66000	5280
18	108	72000	5760

The outputs based on the five simulation runs are used to calculate the dynamic parameters. The lateral forces and derailment quotient at wheel rail contact, lateral and vertical acceleration over front and rear truck and lateral and vertical ride index over front and rear trucks at traction pin of each truck are processed. In the simulation results the graphs are obtained which give us the lateral forces or accelerations or the ride index values with respect to time. Also, the results are compared with the track irregularities in vertical and lateral directions to know the cause of the peaks in the results. The maximum value from the graph is noted. The output parameters studied are Lateral forces, Derailment coefficient, Vertical and Lateral accelerations and Ride index.

In the simulation run results, the derailment coefficient is calculated at two different points/interfaces of the locomotive truck, viz. L/V ratio at the wheel and rail interface of each wheel of the locomotive and H/Q ratio is at the axle center of each wheel. This is because, there are some forces which are absorbed by various components, hence there might be change in forces from point to point. So, to validate the results and the trend in the behaviour of derailment coefficient, it is calculated at two different points. In a locomotive we have two trucks, front truck and the rear truck. Accelerations and ride index results are calculated at the traction link interface of the truck. Hence, in calculation of accelerations we get total of four results, viz. Lateral front, vertical front, lateral rear and vertical rear.

In case of ride index calculations, the results are calculated at two points on each truck, which means in total four result sets for each truck, viz. Vertical front, lateral front, vertical pivot front, lateral pivot front. Similarly, for the rear truck the result sets would be vertical rear, lateral rear, vertical pivot rear and lateral pivot rear. The vertical and lateral comfort values are at the traction link interface (primary traction) and the vertical pivot and lateral pivot are calculated at the traction pin interface (secondary traction).

B. Process flow for sensitivity analysis

The model used for simulation is loaded in ADAMS/Rail software, which has its initial specifications setup for first run. Once the model is loaded, following procedure is carried out for sensitivity analysis:

a. Considering the first set of input variables for first simulation run. Axle load of 12 tons, carbody weight of 48000 kgs and frame weight of 3840 kgs.
b. The axle load is changed by changing the carbody and truck frame weight. Once this is done, the weight of the locomotive becomes 72 tons.
c. Before the simulation is done, preloading needs to be done. Preload analysis is done. It deals with applying load on the springs and bushings in static condition so that the simulation starts in loaded condition.
d. Dynamic analysis follows the preload analysis. In dynamic analysis, we input the value for initial velocity as 30.55 m/s, simulation end time of 34 seconds, time step of 0.01 sec, class 5 track file and contact configuration which give the friction coefficient of 0.5.
e. Simulation run is carried out.
f. Once the simulation is finished, post-processing is used to access the result files in ADAMS/Rail, which give us graphs for all the calculated parameters.
g. Six lateral force graphs are plotted for six wheelsets and we need to check for the highest peak amongst the graph for different wheelsets to find the maximum lateral force acting.
h. Similarly, the graphs for other parameters like derailment coefficient (L/V and H/Q ratio), vertical and lateral accelerations and ride index are plotted and the maximum values for each parameter is found.
i. Now, the axle load is changed to next iteration of 13.5 tons by changing the carbody and truck frame weights according to the respective proportions, keeping the locomotive speed and track file same.
j. Preload analysis is again carried out followed by the dynamic analysis.
k. All the graphs are studied, and the results are analyzed.
l. Sensitivity of each parameter, to change in axle load is plotted and relations are derived between the dynamic parameters and axle load.

IV. EFFECT OF AXLE LOAD ON DYNAMIC PARAMETERS

The results of the sensitivity analysis simulations done on the ADAMS/Rail model are discussed in this section. The values of axle load were changed based on the input matrix shown in Table I. Five simulation runs are done to predict the sensitivity performance of the suspension dynamic parameters such as lateral forces, derailment coefficient, vertical and lateral accelerations of front and rear trucks and ride index in lateral and vertical directions.

A. Lateral forces analysis

The results for lateral forces are obtained by plotting the forces generated with respect to time. In the post processing separate graphs for each wheelset are obtained, which in total makes it six plots of force vs time. To study more precise trend of lateral forces generated at different axle loads, the Root Mean Square value (RMS value), is considered, which gives the mean value for all the peaks in the graph. For better understanding of the results the force plot is compared with the track irregularities in lateral (y-direction) and vertical direction (z-direction) shown in Figure 2. From this comparison the reasons for the peaks in the force plots can be studied and conclusions can be derived. This is done for all the five simulation runs for different axle loads. The results matrix in Table II for the lateral forces shows the lateral force maximum value, the RMS value of the lateral forces, the time when the peak was observed and the wheelset number in which the maximum lateral force was observed.

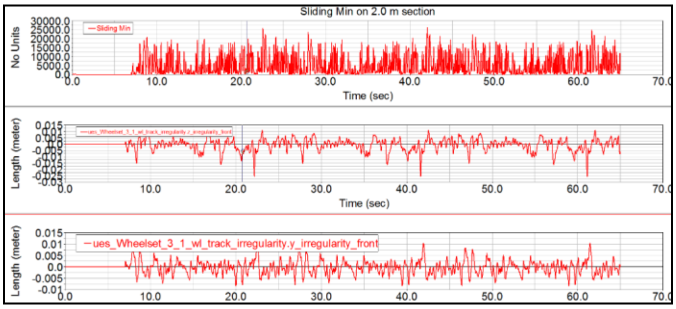

Figure 3. Lateral force vs time plot for 12 tons axle load compared with vertical (z) and lateral (y) track irregularities

Similarly, the graphs are obtained for the rest of the simulation runs at 13.5 tons, 15 tons, 16.5 tons and 18 tons of axle load, for which the output matrix is generated.

TABLE II. LATERAL FORCES SIMULATION RESULTS

Axle load	Lat Force wheelset	RMS	Time of peak	Wheelset no.
12	26267	7151	42.05	3rd front
13.5	25637	7263	42.07	3rd front
15	27073	7470	42.06	3rd front
16.5	27053	7585	42.07	3rd front
18	28156	7787	61.55	3rd front

The lateral forces generated are continuously increasing as the axle load is increased. The peaks at each axle load are happening at a simulation time of around 42.05 seconds, which when compared with the irregularities graph of the track concludes that, one of the reasons for the peak in the lateral forces can be the appearance of maximum deflections in both the directions in the track. The mass and the force have a direct relation as per the Newtons second law.

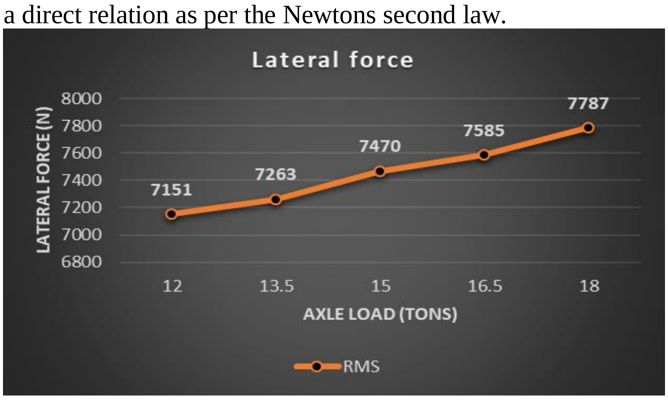

Figure 4. Sensitivity of lateral forces on axle load change

So, as the load is increased the forces generated at the wheelset and rail interface is increased. Also, when the leading axle gets hit by a bump it experiences some lateral forces which are transmitted to the middle and end axle as well. Due to this, when the end axle goes over the bump on the track, the forces generated are added up and the maximum force experienced is in the end axles.

B. Derailment coefficient analysis

The derailment coefficient is determined at two different positions in the locomotive one at the wheel and the rail interface, denoted by L/V ratio and the other at the axle and wheel interface, denoted by H/Q ratio. In this analysis total 12 plots are generated for the 12 wheels of the locomotive which represent the derailment coefficient vs time. The peak in the plot denotes the highest derailment coefficient, which means that at that particular point of time the chances of derailment are maximum.

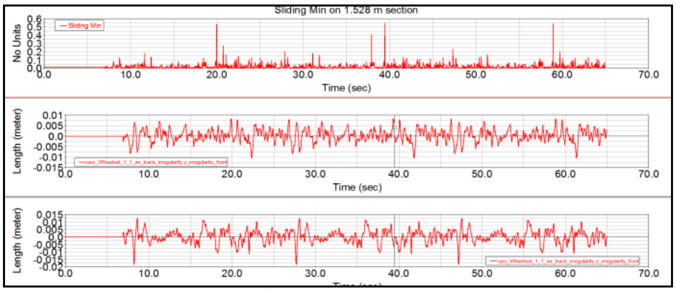

Figure 5. L/V ratio plot for 12 tons axle load compared with track irregularities

Similarly, the plots for 13.5 tons, 15 tons, 16.5 tons and 18 tons are plotted to study the sensitivity of the derailment coefficient to the axle load. The output matrix for the L/V ratio and the H/Q ratio is shown in Table III.

TABLE III. L/V RATIO SIMULATION RESULTS

Axle load	L/V Ratio	RMS	Time of Peak	Wheelset	Wheel
12	0.5437	0.0319	39.46	1st Front	Right
13.5	0.5219	0.0298	39.46	1st Front	Right
15	0.5308	0.0266	39.46	1st Front	Right
16.5	0.4743	0.0251	39.47	1st Front	Right
18	0.4595	0.0243	39.45	1st Front	Right

The derailment coefficient is decreasing as the axle load is increased. The derailment coefficient decreases even if the lateral forces are increasing when the axle load is increased, this is because the vertical forces are increasing more as compared to the lateral forces. The analysis of the results for L/V ratio, it is seen that the peaks are occurring at exactly the same time i.e. 39.45 seconds. On comparison with the irregularities on the track it can be said that the peak in L/V ratio plot is because of the sudden peak in the y-direction irregularities on the track. The y-direction irregularity is attaining it maximum value at this time, hence the maximum lateral movement and maximum derailment coefficient. The results show that the L/V ratio is maximum mostly in the first wheel set of the front truck, this is because when the irregularity appears on the track, the first wheelset experiences the maximum impact, due to which the tendency of derailment is more at this moment. The other two wheelsets adjust themselves according to the movement of the leading wheelset, hence they experience less movement and change in lateral forces. The trend for the derailment coefficient is plotted to determine the sensitivity of axle load on the L/V ratio.

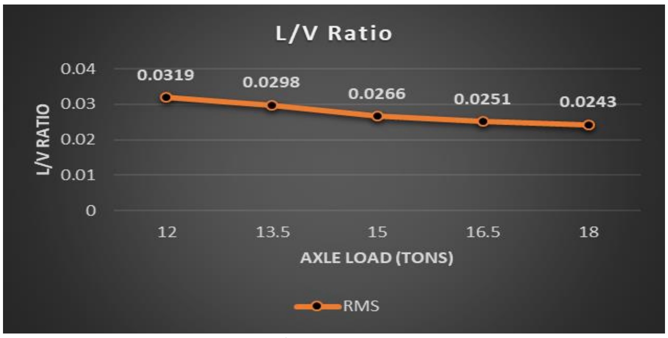

Figure 6. Sensitivity of L/V ratio on the axle load change

C. *Acceleration analysis*

Locomotive experiences acceleration in two directions i.e. lateral and vertical. Lateral accelerations are mainly due to the cornering effect that arises due to the contact between the wheel and the rail due to the alignment of the tracks. In this condition, when the wheel experiences a lateral force and vertical force, the point of interface at the traction link moves in the lateral direction as well as vertical direction. The lateral accelerations depend on the track irregularities in y-directions. Whenever the track irregularities are more, the lateral forces developed are more. As we know that the force is directly proportional to the acceleration according to the Newtons second law F=ma. The output matrix for rear truck accelerations is shown in Table IV.

TABLE IV. REAR TRUCK LATERAL AND VERTICAL ACCELERATIONS

Axle load (tons)	**Locomotive weight (tons)**	**Rear Lateral Acceleration (m/sec^2)**	**Rear Vertical Acceleration (m/sec^2)**
12	72	0.9603	2.9087
13.5	81	0.9296	2.7417
15	90	0.8736	2.6951
16.5	99	0.8323	2.6311
18	108	0.8136	2.6363

The accelerations are depended on the type of irregularities on the track in lateral or vertical directions. It is observed that the upper limit of the vertical accelerations is higher (2.9087) as compared to the lateral accelerations (0.9603), this is because the irregularities in the vertical directions are of higher amplitudes which has more impact on the suspension system. Also, the load on the suspension in the vertical direction is more, hence the accelerations generated are of more amplitudes.

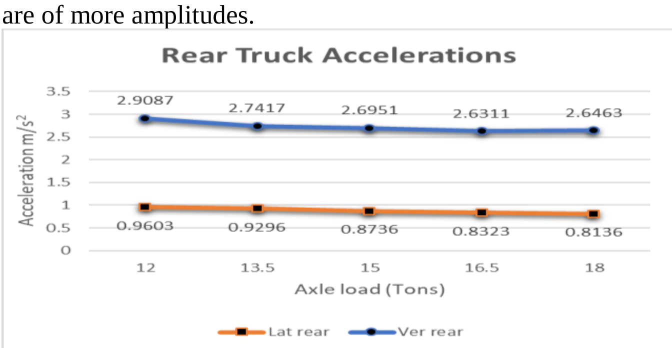

Figure 7. Sensitivity of accelerations on axle load change

The Figure 7 shows the trend in the lateral and vertical accelerations in the rear truck of the locomotive with respect to the changing axle loads. The change in the lateral forces is less as compared to the change in the vertical forces as the axle load is changed. From the previous analysis we already know that the vertical forces developed are more than the lateral forces.

The front truck acceleration shows a similar trend as in the rear truck analysis. In the front truck the accelerations in the lateral direction are higher as compared to the rear truck, this is because the front truck is the leading truck which takes the sudden impact of the irregularities, which is followed by the rear truck which already adjusts according to the movement of the front truck. Due to this reason, the derailment coefficient is also higher in the front truck wheelset.

TABLE V. FRONT TRUCK LATERAL AND VERTICAL ACCELERATIONS

Axle load (tons)	**Locomotive weight (tons)**	**Front Lateral Acceleration (m/sec^2)**	**Front Vertical Acceleration (m/sec^2)**
12	72	1.0601	2.2431
13.5	81	1.0312	2.2264
15	90	1.0391	2.2072
16.5	99	1.0391	2.1787
18	108	1.0352	2.1458

The load on the suspension in the vertical direction is more, hence the accelerations generated are of more amplitudes. The Figure 8 shows the trend in the lateral and vertical accelerations in the front truck of the locomotive with respect to the changing axle loads.

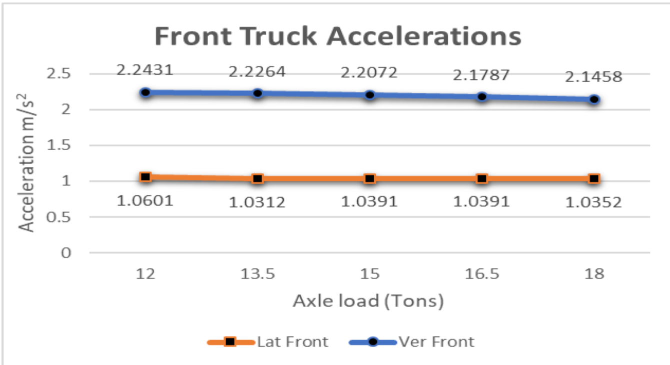

Figure 8. Sensitivity of accelerations on axle load change

D. *Ride Index Analysis*

Ride quality is usually interpreted as the capability of the vehicle suspension to maintain the motion within the range of human comfort and/or within the range necessary to ensure that there is no lading damage. The ride quality of a vehicle depends on displacement, acceleration, rate of change of acceleration, and other factors such as noise, dust, humidity, and temperature. The ride index is calculated at four points in the locomotive, viz. front truck traction link, rear truck traction link, front truck pivot and rear truck pivot. The matrix in Table VI shows the results for ride index in lateral directions, when the axle load is changed.

TABLE VI. LATERAL RIDE INDEX RESULTS

Axle load	Loco Weight	Lat. Front	Lat. Rear	Lat. Pivot Rear	Lat. Pivot Rear
12	72	2.46	2.39	2.36	2.27
13.5	81	2.45	2.38	2.35	2.26
15	90	2.45	2.37	2.34	2.24
16.5	99	2.45	2.36	2.34	2.23
18	108	2.45	2.36	2.34	2.21

Here, the ride index shows a decreasing trend as the axle load is increased. This can conclude that the springs used for the simulation runs might be stiffer at lower loads and acceptable for the higher loads. So, it can be said that, for the spring combination of 400-140 N/mm PS-SS, the ride index gets better when the axle load is increased.

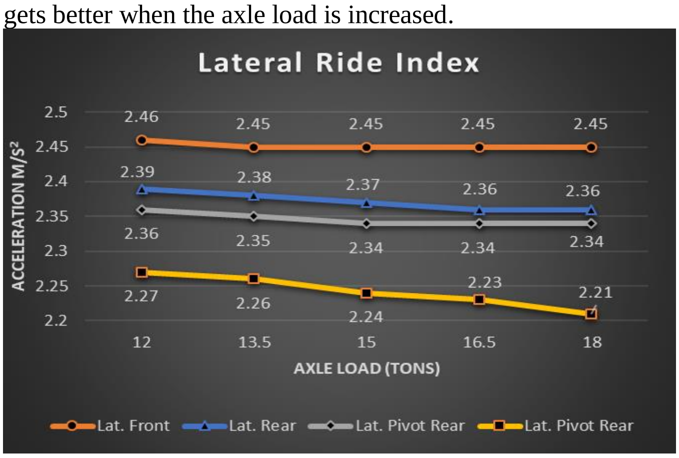

Figure 9: Sensitivity of lateral ride index on the axle load change

The vertical ride index depends on the vertical accelerations developed at the traction link and traction pin joint. As we have analyzed in the previous sections that the vertical forces are higher as compared to the lateral forces, the accelerations generated in the vertical directions are higher than the lateral accelerations in the links.

TABLE VII: VERTICAL RIDE INDEX RESULTS

Axle load	Loco Weight	Ver. Front	Ver. Rear	Ver. Pivot Front	Ver. Pivot Rear
12	72	2.49	2.52	2.38	2.41
13.5	81	2.46	2.52	2.34	2.41
15	90	2.42	2.51	2.31	2.39
16.5	99	2.4	2.51	2.29	2.38
18	108	2.4	2.52	2.29	2.38

In case of the vertical ride index, the change seen in the value is significant in the front truck. As it is seen that in the vertical front and the pivot front the ride index has a significant change. In the rear truck the change is not so significant. This is because the vertical accelerations in the rear truck are higher as compared to the front truck. Hence the ride index of the front truck is lower than the rear truck. We can say that the acceleration and the ride index are inversely related.

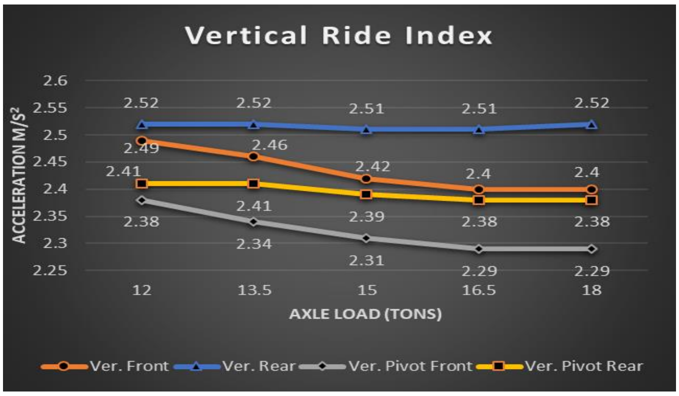

Figure 6.18: Sensitivity of vertical ride index to axle load change

V. CONCLUSIONS AND FUTURE SCOPE

Sensitivity of the dynamic parameters like lateral forces, derailment coefficient, vertical and lateral accelerations and ride index, to the changing load is an important study to develop modularity in the use of suspension system. The lateral forces generated in the wheelsets increased by 2 KN as the axle load increased from 12 to 18 tons. The derailment quotient, acceleration and ride index number show a decreasing trend with increase in the axle load, which concludes that the suspension has better ride quality and comfort at higher axle loads of 16 to 18 tons. The suspension spring of 400-140 N/mm combination shows the results for all the dynamic parameters within the acceptable ranges. Hence, this spring combination can be used for locomotives within the axle load band of 12 to 18 tons. This concept achieves the modularity of the suspension system which can have multiple applications within 12 to 18 tons axle load conditions.

The future work may include, sensitivity analysis on the effect of change in the suspension spring parameters. by changing the stiffness to understand the trend in the performance of the suspension system because of the spring change.

REFERENCES

[1] Mortadha Graa, “Modeling and control for Vertical Rail Vehicle Dynamic Vibration with Comfort Evaluation,” International Journal of Mathematical Models and Methods in Applied Sciences, Volume 11, 2017.
[2] Neha Bajpai, “Suspension Spring Parameter's Optimization of an Indian Railway Freight Vehicle for Better Ride Quality,” IEEE International Conference on Advances in Engineering & Technology Research, August 2014.
[3] T Nikhil Kumar, M Pramod Reddy and Dr P Sampath Rao, “Design and analysis of a freight locomotive suspension spring and redesign of the spring For Durability and ride index,” International Journal of Innovative Research In Technology, Volume 3, January 2013.
[4] Srihari Palli, Dr.Ramji Koona, Venkatesh Muddada, “Dynamic Analysis of Indian Railway Bogie,” International Conference on Newest Drifts in Mechanical Engineering, pp. 627-632, December 2014.
[5] Rakesh Chandmal Sharma, “Recent Advances in Railway Vehicle Dynamics,” International Journal of Vehicle Structures and Systems, pp. 52-63, 2012.
[6] Shuaishuai Sun, Huaxia deng and Weihua Li, “A variable stiffness and damping suspension system for trains,” Proceedings of SPIE: Active and Passive Smart Structures and Integrated Systems, pp. 1-12, 2014.
[7] V. Apoorva, J. N. Hemalatha, W. M. Achyutha and B. W. Shivaraj, "Steps involved while introducing a new product into a healthcare organisation," 2017 International Conference on Computing Methodologies and Communication (ICCMC), Erode, 2017, pp. 180-18