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To cite this article: K Chinnaraj *et al* 2009 *J. Phys.: Conf. Ser.* **181** 012070

Journal of Physics: Conference Series **181** (2009) 012070 doi:10.1088/1742-6596/181/1/012070

Dynamic Response Analysis of a Heavy Commercial Vehicle Subjected to Extreme Road Operating Conditions

Chinnaraj K.[1], Dr. Sathya Prasad Mangalaramanan[2] and Dr. Lakshmana Rao C.[3]

[1] Research Scholar, Department of Applied Mechanics, IIT Madras, India
Senior Manager, Computer Aided Engineering, Ashok Leyland Ltd, Chennai, India
[2] Section Head, Advanced Engineering, Ashok Leyland Ltd, Chennai, India
[3] Professor, Department of Applied Mechanics, IIT Madras, India

chinnarajk.alvvc@ashokleyland.com, prasad.alc@ashokleyland.com, lakshman@iitm.ac.in

Abstract: Wheel excitations measured on a heavy commercial vehicle by driving it through extreme road operating conditions, are considered as inputs to perform dynamic response analysis in a simulated laboratory and computational environment. From initial modal analysis results using finite elements, critical vehicle frame rail locations are identified for dynamic laboratory strain measurements on a six poster road load simulator that employs dynamic wheel excitations as input. Dynamic stresses calculated from measured strain values are then compared with computationally obtained stress results on each of these locations. This study also points out all geometric locations and vibration modes that may affect the design behavior of the frame members under extreme road operating conditions. The results obtained from this work can be considered for further fatigue life prediction and design optimization of chassis frame rail assembly.

1. Introduction

In the design of automotive structures, if a structural member is to withstand millions of cycles of dynamic load applications, it is important to understand the dynamic stress-time histories due to repeated dynamic loads caused by uneven, extreme road operating conditions. Traditionally in automotive industry, road load specifications for design, verification and validation tests are directly given by Original Equipment Manufacturers (OEMs), which are generated based on an envelope of generic customer usage profiles [1]. These generic road load specifications in many cases are over specified, often leading to failure modes different from those experienced in field. Therefore, in recent times, the road load specification for a particular vehicle is generated based on real-time road load measurements.

The direct strain measurement at various frame rail locations by driving the vehicle on an accelerated road-test may be considered as one of the options for evaluating dynamic response of frame rail assembly. However, carrying out physical testing in accelerated test environment may not be feasible in all cases due to reasons such as complexity in instrumentation, driver fatigue and chances for unnoticed failures during testing, etc; Hence a new approach combining experimental and computational methods, is proposed as an efficient and reliable means for the dynamic response analysis of a heavy commercial vehicle in its early development stage. From the dynamic analysis results, durability of automotive structure can also be performed thereof, for any given road operating conditions.

In the first step of this approach, a prototype vehicle is used once to capture the road load excitations on extreme road operating conditions. In the second step, road load excitations are filtered and processed for giving it as an input signal for six poster road load simulator and finite element model. In the third step, critical frame rail locations are identified for strain-history measurement based on analytical modal analysis results. In the fourth step, transient dynamic analysis is carried out using modal super position technique to predict dynamic stress histories on the identified frame rail locations. This technique is chosen in the analysis due to its applicability to automotive structures especially when the structural mass is large, the stiffness is low and the frequencies of the operating loads pass through number of natural frequencies of the structure [2]. Finally from the strain history measured from six poster road load simulator, dynamic stress histories are calculated and compared with analytically predicted dynamic stress histories on those critical locations.

2. Structural Response in Time Domain and Vehicle Load Specification

It has been established that most vehicle loads can be generally described by a stationary random process [3]. Vehicle surface road loads to automotive structures are typically random in nature, as evidenced from Figures 1 to 6. The stationary random process of a vehicle can be easily described in terms of time or frequency domain format. In this work, data obtained in time-domain format is directly considered for carrying out vehicle structural response analysis.

2.1. Structural Response in Time Domain

Equation of motion of a linear automotive structural system, in general, is expressed in matrix format using equation (1). The system of differential equations can be solved directly in the physical coordinate system, corresponding to each load-time step:

$$[M]\{\ddot{x}(t)\}+[C]\{\dot{x}(t)\}+[K]\{x(t)\}=\{p(t)\}, \qquad (1)$$

where [*M*], [*C*], [*K*] are mass, damping and stiffness matrices, respectively and $\{p(t)\}$is an applied load vector. From the displacement vector $\{x(t)\}$ obtained by solving the system of differential equations (1), the other transient response variables such as acceleration, strain and stress can be obtained for each time step [4,5].

2.2. Road Load Specification for Six Poster Simulator and Computational Model

The dynamic road load excitations at all wheels are measured by driving the prototype vehicle at a constant speed of 30 km/h, on various test road conditions featuring Belgian Pave, Corrugation, Herring Bone, Setts and Pothole roads. The measured road load excitations in the form of acceleration is then filtered and processed using SoMat Ease3 data processing software [6], before taking it as acceleration input signal to six poster road load simulator for experimental strain measurement, and FE model for numerical simulation.

In the current work, only a stretch of 10 sec road acceleration data on Belgian pave road representing extreme road operating conditions is considered as input for the ease of handling of huge computational data. Figures 1 to 6 show time history of acceleration profiles measured using accelerometers at all axle ends of prototype vehicle on Belgian pave test track, available in Ashok Leyland Technical Center.

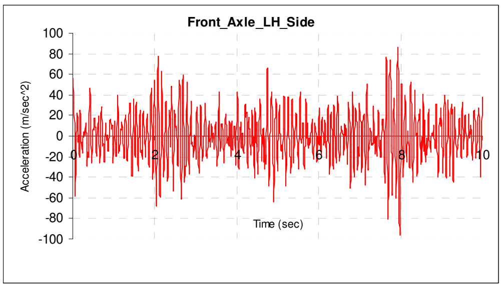

Figure 1. Pave Signal on Front Axle LH Side

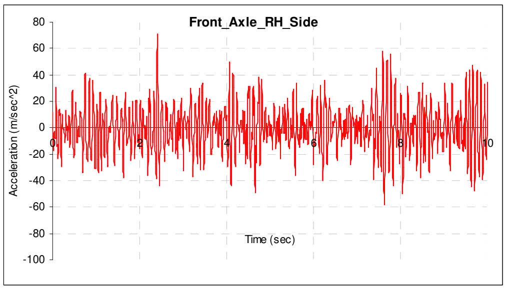

Figure 2. Pave Signal on Front Axle RH Side

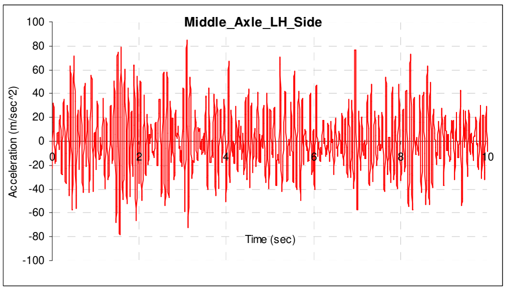

Figure 3. Pave Signal on Middle Axle LH Side

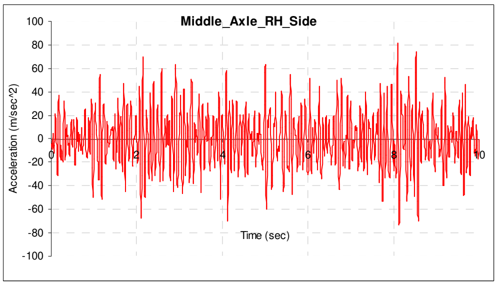

Figure 4. Pave Signal on Middle Axle RH Side

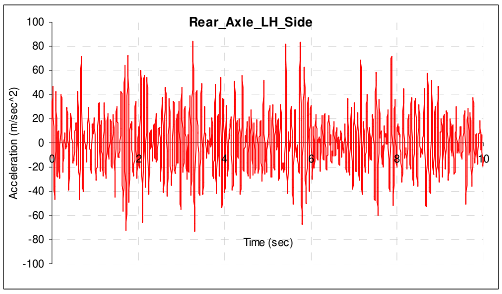

Figure 5. Pave Signal on Rear Axle LH Side

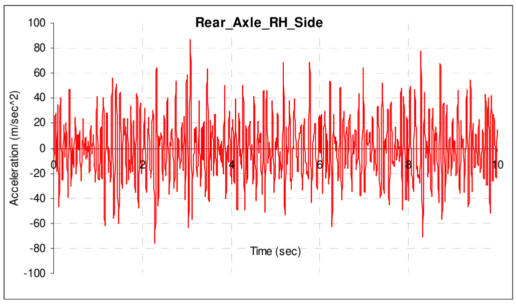

Figure 6. Pave Signal on Rear Axle RH Side

3. Computational Analysis

3.1. Fundamental Vibration Modes and Failure Locations

In order to identify the critical failure locations in load bearing structural members, the truck systems are modeled using finite elements as shown in figure 7. The frame side and cross members are modeled by maintaining smooth flow of shell elements throughout the geometry. Bolt connections in the assembly are simulated by using beam and spider elements. The section and material properties are specified for beam elements to get correct stiffness and realistic mass of bolt joints. While commercial CAD software CATIA [7] is used to model the geometry, Hyperworks [8] preprocessor is employed to mesh the model before taking into MSC Nastran [9] finite element solver.

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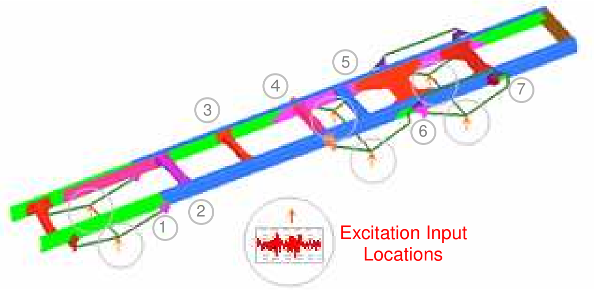

Vehicle Specification

Vehicle Model	Multi Axle Truck
Wheel base	3850mm
GVW	25 Tons
Overall Length	8300mm
Front Overhang	1200mm
Rear Overhang	1800mm
Vehicle Width	2432mm

Figure 7. FE Model with Seven Strain Locations and Six Input Locations

The effect of tyres, leaf springs and bushes are considered in the analysis by measuring the stiffness properties from component tests and idealizing them as linear spring elements in the FE Model. Other non-load bearing structures and major vehicle assemblies such as engine, cab, fuel tank and payload are represented by lumping their respective masses at appropriate center of gravity locations and connecting to relevant bolt holes of load bearing structural members.

The predicted modal frequencies and mode shapes for the first six low vibration modes by employing MSC Nastran code is shown in figures 8 and 9. From the modal analysis results, seven critical locations are identified, where the presence of residual stresses and high loading stresses are expected.

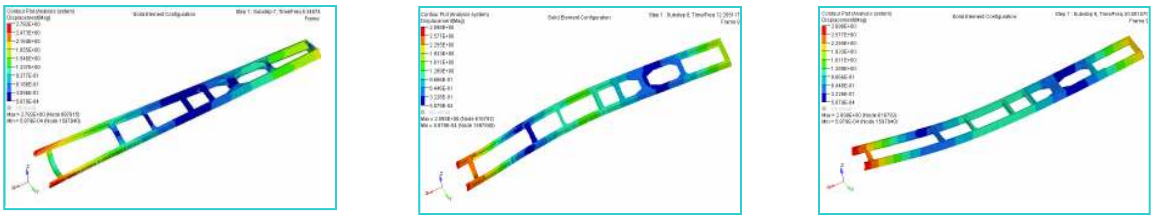

Figure 8. Fundamental Torsion, Lateral Bending and Vertical Bending Modes (1st, 2nd and 3rd natural Frequencies - 5.35 Hz, 12.20 Hz and 20 Hz, respectively)

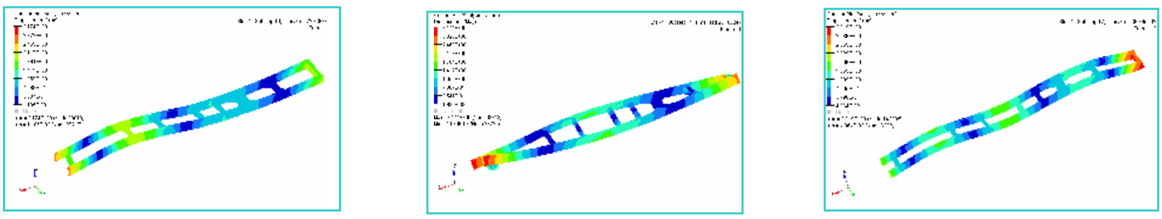

Figure 9. Full Cycle Lateral Bending, Combined Torsion & Bending and 1.5 Cycle Lateral Bending Modes (4th, 5th and 6th natural Frequencies - 22.84 Hz, 25.41 Hz and 39.76 Hz, respectively)

These identified frame rail locations are strain gauged using 45^0 strain rosettes to experimentally measure the strain-history on six poster road load simulator.

3.2. Computational Transient Response Analysis
The finite element model used in modal analysis is supplemented with appropriate constraints and degree of freedoms for giving excitation load in vertical direction on all six wheel locations. Apart from vertical road load excitations, inertia loads of major assemblies are taken into consideration by applying mass and mass moment of inertia of all assemblies at its corresponding center of gravity locations. While the direct transient analysis is not feasible for automotive applications as it uses prohibitive computation time and needs considerable amount of memory to store output results, the modal superposition approach, is used for carrying out dynamic analysis.

Modal superposition approach is an effective technique to obtain the transient response of any structure. This approach evaluates stress histories by linearly combining the modal participation histories of each mode of interest, with the corresponding modal influence coefficient as obtained from the modal analysis. While using this technique, two important parameters that need to be considered are the number of modes to be extracted and accounted for, and the size of the time step to be used when obtaining the participation histories.

4. Experimental Analysis

Figure 10 shows the six poster road load simulator available in Ashok Leyland Technical Center. It consists of 6 identical vertical load modules using servo hydraulic actuators fitted with precision hydrostatic bearings, providing the support to reproduce any vertical road surface events accurately in laboratory environment. Using this simulator, any road simulations such as durability tests could be easily performed on 6 wheel vehicles up to 600 KN GVW (Gross Vehicle Weight). Besides durability tests, it is also used to test vehicles for squeaks and rattles; evaluate the ride behavior of different vehicles with a set of standard road profiles and test vehicles as they roll off the production line.

For vehicle road load simulation using this system, the actual road surface events on vehicle wheels are measured using accelerometers or string potentiometers for displacements. The measured signals are then analyzed and digitized in terms of actuator displacements using remote parameter control software before feeding into servo-controller of the simulator to execute vehicle simulation test.

Figure 10. Six Poster Road Load Simulator

In the laboratory experiments, the identified critical frame rail locations from analytical modal analysis are strain gauged using 45^0 strain rosettes to measure dynamic strain histories on six poster road load simulator. Experimentally measured and processed 10 sec Belgian pave road load data is given as input to six poster simulator and corresponding strain time histories with sampling rate of 100 Hz are recorded on those identified frame rail locations.

From the measured dynamic strain histories, the dynamic stress histories are calculated and compared with the computationally obtained dynamic stress histories at all critical frame rail locations.

5. Comparison of Experimental and Computational Results

The comparison of dynamic stress values obtained from road load simulator measurement and computationally predicted stress values on all identified locations are shown in Figures 11 to 17.

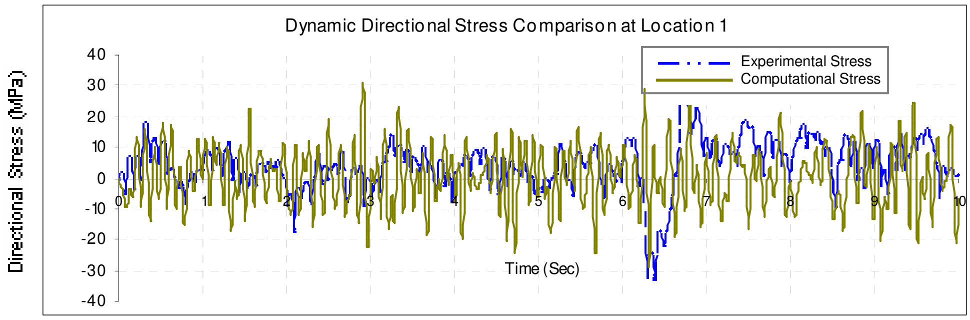

Figure 11. Experimental vs. Computational Dynamic Directional Stress at Location 1

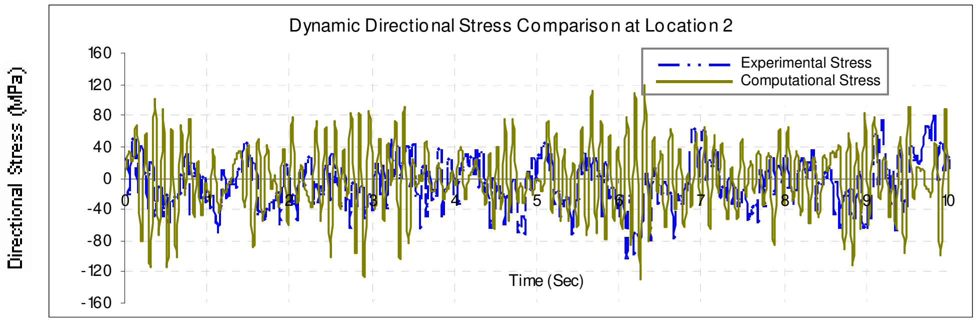

Figure 12. Experimental vs. Computational Dynamic Directional Stress at Location 2

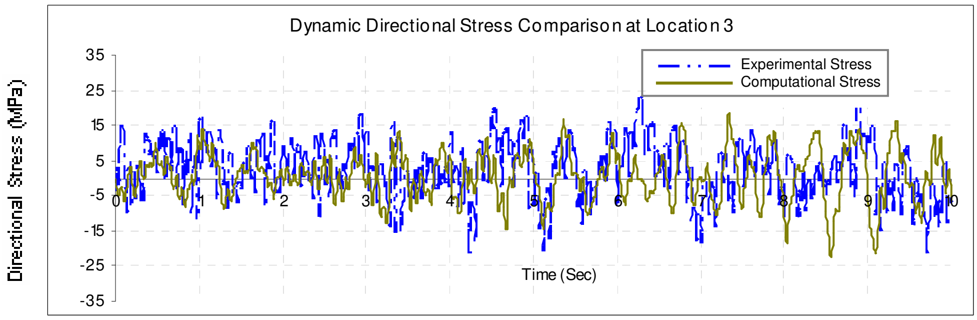

Figure 13. Experimental vs. Computational Dynamic Directional Stress at Location 3

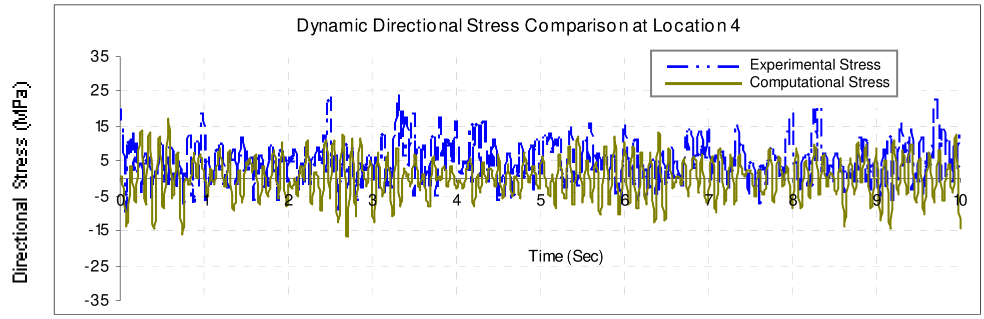

Figure 14. Experimental vs. Computational Dynamic Directional Stress at Location 4

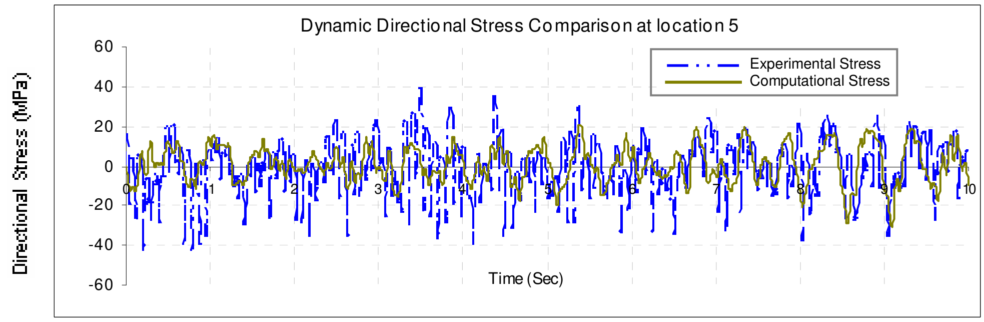

Figure 15. Experimental vs. Computational Dynamic Directional Stress at Location 5

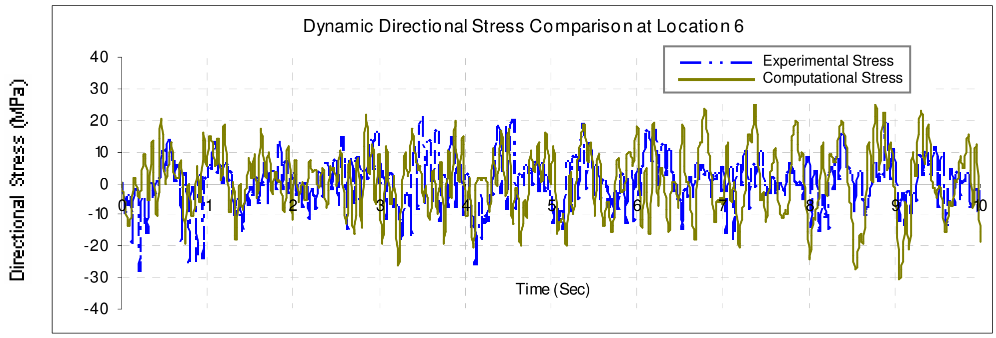

Figure 16. Experimental vs. Computational Dynamic Directional Stress at Location 6

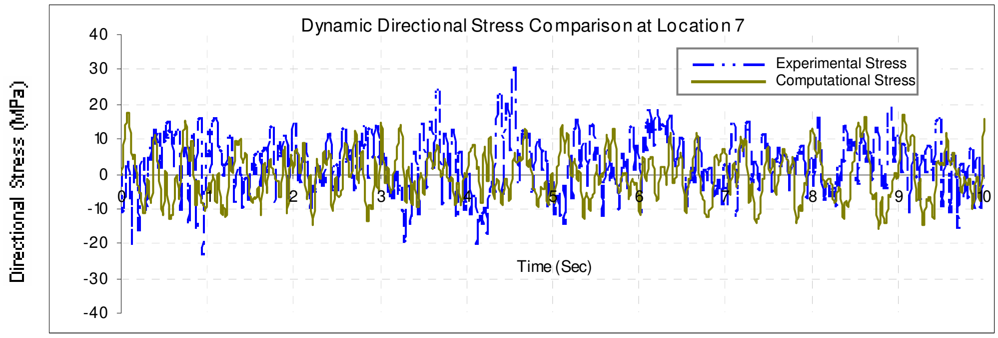

Figure 17. Experimental vs. Computational Dynamic Directional Stress at Location 7

From the comparative dynamic stress plots, it could be observed that the computational stress amplitudes are higher than the experimental stress amplitudes at locations 1 and 2 and lower at locations 5 and 7.

Among other locations, we could observe close correlation between computational and experimental stresses in locations 3 and 6, although there are slight deviations at the end.

The likely causes of the difference between simulated and measured stresses in locations 1, 2, 5 and 7 could be the residual stresses resulting from the fabrication and assembling of frame structure. These residual stress effects could not be accounted in the computational method employed in this work and currently research is under progress to validate the hypothesis.

6. Summary and Conclusions

This work presents an efficient methodology for the integrated dynamic response analysis of a heavy commercial vehicle consisting of accelerated road load measurement, road load simulation on a six poster, dynamic stress analysis on experimental as well as computational environment.

With the demonstrated methodology, dynamic response analysis of any prototype vehicle can be performed efficiently in simulated environment for any road load or extreme road operating conditions. As this method is faster and at the same time tests are performed with real time road inputs, it would be a preferred method for OEMs especially during initial stage of vehicle development.

This study also opens up an interesting hypothesis that the residual stresses resulting from the fabrication and assembling of frame structure may have significant effect on experimental strain measurement. These effects are being investigated separately as a future work.

Acknowledgments

The authors would like to thank M/s Ashok Leyland Ltd, India, especially Mr.R.R.G.Menon, Executive Director, Product Development, for his valuable support to carry out this exercise at their Technical Centre. The authors also thank Mr. P.Dayalan and Mr. S.Nandakumar of M/s Ashok Leyland Ltd for their help during experimental measurement.

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