

# Development of a Dynamic Vibration Absorber to Reduce Frame Beaming

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## ABSTRACT

This paper describes the development and testing of a Dynamic Vibration Absorber to reduce frame beaming vibration in a highway tractor.

Frame beaming occurs when the first vertical bending mode of the frame is excited by road or wheel-end inputs. It is primarily a problem for driver comfort. Up until now, few options were available to resolve this problem.

The paper will review the phenomenon, design factors affecting a vehicle's sensitivity to frame beaming, and the principles of Dynamic Vibration Absorbers (AKA Tuned Mass Dampers). Finally, the paper will describe simulation and testing that led to the development of an effective vibration absorber as a field fix.

**CITATION:** Anderson, J., "Development of a Dynamic Vibration Absorber to Reduce Frame Beaming," *SAE Int. J. Commer. Veh.* 7(2):2014, doi:10.4271/2014-01-2315.

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## INTRODUCTION

"Frame beaming" occurs when the first vertical bending mode of the frame is excited by road or wheel-end inputs. It is primarily a driver comfort problem. Frame Beaming is frequently a smooth-road problem that occurs at highway speeds (80 to 100 km/h) when the tire rotational frequency coincides with the frame beaming frequency. Frame beaming problems occur most often without a trailer, although they sometimes occur with one.

This is not a new problem. William LeFevre describes it in his "Truck Ride Guide" from 1967 [1] and Thomas Gillespie also mentions it in his SAE paper from 1985 [2]. Frame beaming problems are minimized in design by providing isolation of the cab from the frame, and by limiting the excitation at its source, wheel end irregularities, particularly radial run-out. Cab/sleeper suspensions were developed primarily to isolate the driver from the beaming mode of the frame. Wheel-end excitation can be reduced by limiting radial run-out of wheels and tires, and by match-mounting the rear drive tires. That is, by pairing dual tires of similar radial run-out and clocking them so that the high spots are 180 degrees apart.

Despite the best efforts of Engineering Design and Field Service, some tractors will still exhibit unacceptable vibration in the cab. Kenworth, with support from the PACCAR Technical Center, undertook a study to better understand this problem and to look for ways to reduce its severity and occurrence. This ultimately led to the development of the Dynamic Vibration Absorber described in this paper.

Since the tractor under study (and most encountered in the field) had the most pronounced beaming response without a trailer, the plots in this paper apply to that case. Simulation and testing was also performed with a laden semi-trailer, confirming that the vibration in that case was also reduced.

## THE FRAME BEAMING PHENOMENON

The flexible ladder frame that is common to most highway tractors will have many modes of vibration that will be excited by inputs from the road, wheel-ends, drivetrain, etc. What is more, the modes of the frame are typically lightly damped, and additional damping is not easy to add. The "frame beaming" mode is the first vertical bending mode of the frame. It will typically occur between 7 and 9 Hz in conventional U.S. market trucks. While this mode can be excited by road roughness, adversely affecting ride comfort, complaints most frequently

arise on smooth pavement when the natural frequency of this mode coincides with wheel-end rotation frequency. [Figure 1](#) shows an ODS (Operating Deflection Shape) of the frame beaming of a real bobtail tractor (without semi-trailer) at its problem speed.

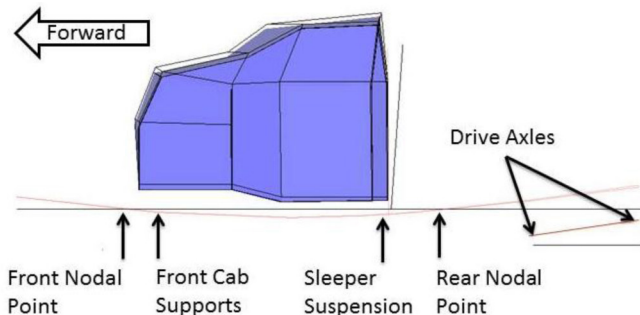


Figure 1. Operating Deflection Shape (ODS) of the tractor beaming mode. Un-deformed shape is shown in black. A line connects the rear drive axle ends for visualization. The approximate locations of the front cab supports and sleeper suspension are also indicated.

Several things become clear from looking at the ODS:

1. There are two nodal points for the mode (where there is no translational motion)
2. Front cab mounts would ideally located near the forward nodal point to minimize input to the cab.
3. The rear of the sleeper is far from the rear nodal point. A suspension will be required for isolation.
4. Rear drive axle is far from a nodal point therefore it will be very effective at exciting this mode. Wheel-end irregularities here must be minimized.
5. Drive axles move with the tail of frame, so suspension damping has little effect on this mode.

Since the frame beaming mode is symmetric (left and right frame rails deform in phase), the tire irregularities that excite it must also be symmetric. Frame beaming can come and go as the vehicle travels through curves. This is because the phase angle changes between inside and outside wheels. Run to run results can vary. For this reason, much of the experimental work described in this paper was performed on a road simulator, to allow consistent phasing between left and right wheels.

## SIMULATION AND TESTING APPROACH

To gain a thorough understanding of the frame beaming problem, a combined simulation and test approach was taken. A finite element analysis (FEA) model of a Kenworth T660 test tractor was created ([figure 2](#)). Most flexible components were included as Abaqus substructures (a.k.a. super-elements). The

cab/sleeper, engine/transmission, cooling module and axles were included as rigid bodies with the appropriate mass properties. Tires are a simple linear model used for ride comfort evaluation. More information about Kenworth's approach to full vehicle vibration simulation can be found in reference [3].

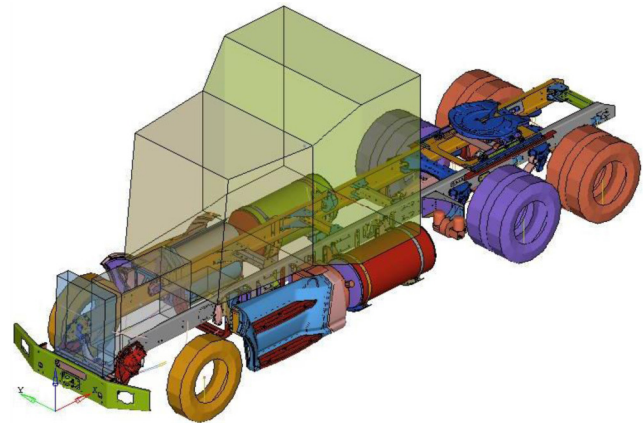


Figure 2. Finite element model of the test tractor. The hood and trailer are not shown for clarity.

Since the model was constructed almost entirely of previously modeled components, only vehicle level validation was considered necessary. Data were collected on the road simulator for comparison to the vehicle model and to demonstrate that the frame beaming response was reasonably well reproduced. Model validation consisted of comparing test and simulation ODS and transmissibility functions from individual axle inputs to response points on the chassis. An example transmissibility function between rear drive axle excitation and vertical response at B-pillar is shown in [figure 3](#). Note that "background" 2 Hz sinusoidal symmetric excitation was present at all other axle ends to ensure that the suspensions do not stick. These cause mismatch in transmissibility at low frequency between the test and simulation results.

The location of measurement points and some additional example measurement comparisons are given in the [appendix](#). Details of the validation work are outside of the scope of this paper.

With a validated vehicle model in hand, it was then possible to investigate the sensitivity of the vehicle to a number of design choices and parameters. The most influential of these were chosen for physical testing. Testing was performed on the Road Simulator and track at the PACCAR Technical Center, and on local roads in the area.

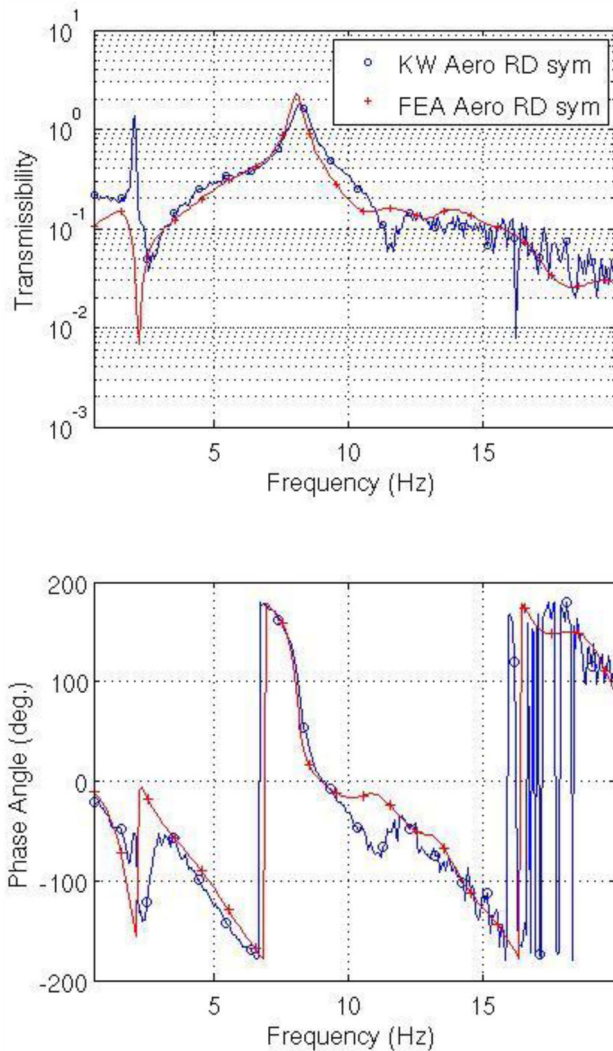


Figure 3. Transmissibility between rear drive axle symmetric excitation (at tire contact points) and B-pillar vertical response without semi-trailer. The peak at about 8 Hz corresponds to the frame beaming mode.

## EFFECTS OF DESIGN CHOICES ON VIBRATION IN CAB

A number of possible design choices and parameter values were investigated by simulation. The only one found to have a significant effect was front cab mount location. Several other parameters that have been “tweaked” in the past were found to have little or no effect in this investigation. They are also discussed here in hopes that no more time will be wasted on them.

### Front Cab Mount Location

As was seen in the operating deflection shape, a nodal point occurs near the front of the cab. Ideally, one would locate the front mount here to avoid vertical input, with the mount also close to the center of rotation to avoid fore/aft input. These are indeed the recommendations one finds in the literature [1] [2]. In practice, this is not always convenient to do. So how important is this? To answer this question, simulation work was performed with the mounts in two locations: One near the nodal point and one set back approximately 450 mm. The results are shown in figure 4. It is clear that front mount location is an important choice.

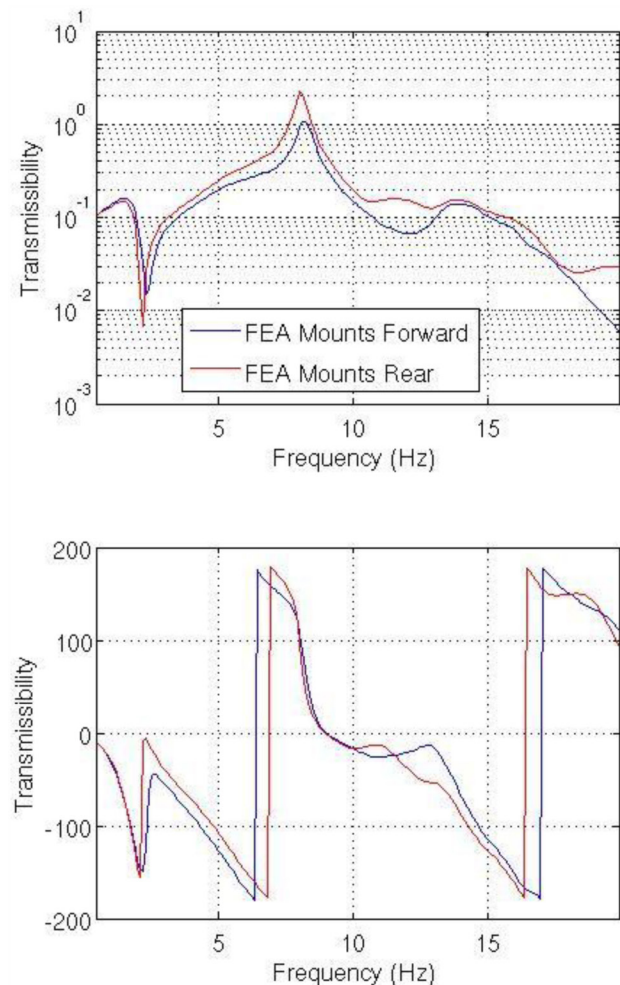


Figure 4. Effects of front mount fore/aft location on transmissibility from rear drive axle symmetric excitation to b-pillar vertical response, without trailer (simulation)

Testing on the road simulator showed less effect, but the effect was still clearly visible (figure 5). It is hypothesized that the discrepancy has to do with the flexibility of the structure used to cantilever the mounts off the front of the cab for the physical test.

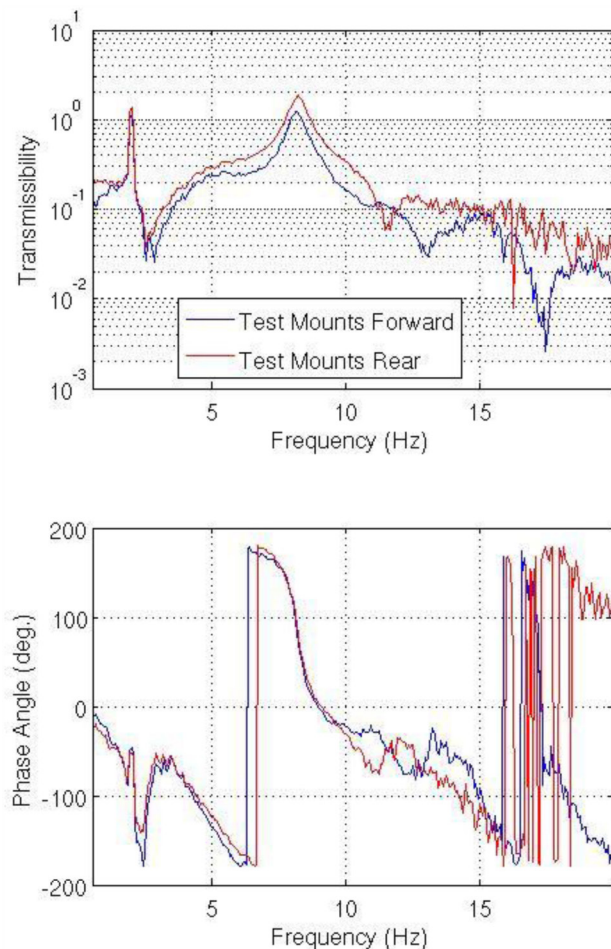


Figure 5. Effects of front mount fore/aft location on transmissibility from rear drive axle symmetric excitation to B-pillar vertical response, without trailer (road simulator test)

### Front Cab Mount Tuning

Field engineers sometimes attempt to reduce frame beaming vibration by installing lower durometer (less stiff) front cab mounts. Simulation showed that softening the mounts will do little, or possibly make the problem a bit worse.

### Sleeper Suspension Tuning

Simulation showed that tuning of the sleeper suspension could not significantly reduce frame beaming vibration in the cab. In fact, it was seen that changing damping from the base tuning could make it slightly worse. Less suspension damping reduces the damping in the mode while high damping transmits more force from the frame into the sleeper. Changes in stiffness within a reasonable range for air springs had almost no effect.

### Front Engine Mount Tuning

It was suggested that the engine might be used as a vibration absorber. In fact, PACCAR holds a patent on this concept [4]. Experimentally, this was not successful. It was thought at the time that perhaps the right combination of stiffness and damping

was not achieved with the sample mounts available. To look for the “sweet spot,” simulation was run with the front engine mount stiffness and damping varied over a wide range of values. Figure 8 below shows the result: The engine apparently does not work as a vibration absorber. Rather than have the “sweet spot” that was sought, there appears to be a “sour” one at about 1200 N/mm and 0.5 Ns/mm. It is hypothesized that this is so because the mass of the engine is much greater than that of the frame: the frame becomes the absorber and has the higher amplitude. As neither the test nor simulation yielded a useful solution, this was not investigated further.

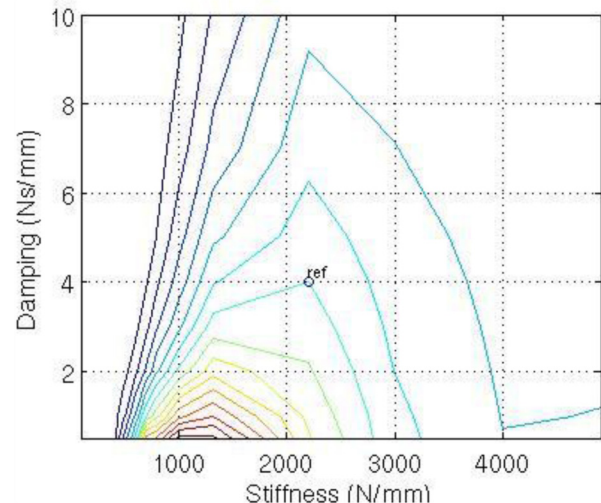


Figure 6. Effects of front engine mount tuning on B-pillar acceleration with drive axle symmetric run-out of 1 mm, without trailer (simulation). Blue = unity (base tuning), Red = ~1.7X acceleration.

### Rear Axle Suspension Damping

Another proposal to reduce frame beaming was to increase damping in the rear suspension. Unfortunately, simulation predicts that this will have little effect. This is not surprising when one looks at the ODS (Figure 1): There is little relative motion between frame and the rear axles, so the dampers will not be working to remove energy from the mode. Some simple experiments with dampers on the physical vehicle confirmed this. Damping effects were also evaluated experimentally by removing the rear axle dampers. This did provide a small reduction in frame beaming response but would not be practical or safe for a road going vehicle.

### Frame Inserts

Another common field fix is to use frame inserts to stiffen the chassis. This means doubling the mid-frame with C-channels nestled the side members. The main effect of this is simply to move the frame beaming frequency up a few Hz (or km/hr). In some cases, this is an acceptable fix if the problem occurred at the upper range of operating speed. Often however it merely shifts the complaint to a different speed while adding a few hundred kg to the vehicle.