An Investigation of Railroad Car Retarder Squeal

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1. Introduction

While the use of humpyards for the classification of railroad cars has been very effective in increasing the volume of cars handled in a yard, one unpleasant by-product of this system is the squeal which is often generated by the retarder systems. As the cars roll down an incline (the "hump") they are slowed by means of a retarder system which usually is two braking mechanisms. These clamp onto the wheels of the car as it passes. The initial retarder, called the master, begins to slow the cars while the second is one of the group retarders. Figure 1 shows three of the group retarders as well as the classification tracks at the CNR Calder Yard in Edmonton. For certain cars (usually the heavier ones) the retarder brake acting on the car wheels causes a high pitched squeal to be generated. The retarders must dissipate energy at rates of up to 400 kW (>500 HP) and even if a relatively small amount is converted to acoustic power high sound pressure levels can result.

Levels of approximately 120dB have been recorded at distances of 100 feet from the retarder. Since most of the acoustic energy is in the 2000-4000 Hz range the squeal tends to be very annoying. Because of the relatively high frequency of the squeal the use of barriers to control the noise seems an obvious answer. While barriers as low as six feet in height have been shown to be effective the practical limit to the insertion loss is about 25 dB. However, this is not sufficient as areas surrounding the retarders can still be severely impacted by the squeals. For this reason alternate techniques which would reduce the noise further have and are being sought.

2. Noise Generation Mechanisms

In order to consider the possible methods for reducing the noise the mechanism which causes the squeal should be understood. While the detailed mechanism for the retarder - wheel system is not known it is believed to be stick-slip vibration or friction induced vibration such as that studied for example by Brockley, Cameron, and Potter [1] and Remington, Rudd and Ver [2].

The screech noise is excited by friction forces that act between the retarder shoes and the car wheels. When these two surfaces have relative sliding motion, intermittent vibration which can be attributed to the variation of the friction forces is observed. For dry friction the motion is a self-excited phenomenon. This occurs because the static coefficient of friction is larger than kinetic coefficient. While the kinetic coefficient is a function of time or the relative velocity between the two surfaces it must in fact be lower than the static value if stick-slip vibrations are to occur. Once establishing the velocity and time dependence of friction, it is possible to obtain an analytical solution to the critical velocity which will cause this stick-slip oscillation.

The analysis also shows that increasing the damping of the system or altering the friction-velocity characteristic can reduce or eliminate the stick-slip action.

The above analysis would account only for radial type oscillations of the car wheel, however, because of the complex shape of the car wheel the transverse modes are also excited. It is these transverse modes which are held primarily responsible for the generation of the acoustic waves which propagates as the screech.

3. Methods of Controlling the Noise

The techniques used to control or eliminate the retarder shoe squeal include path-type controls such as barriers, which have been mentioned above, as well as modification of the source elements.

One of the easiest and quickest techniques to eliminate the screech was the use of various lubricants between the wheel and retarder shoe. These lubricants had the effect of reducing the differences between the static and kinetic coefficients of friction and proved quite effective in the elimination of the squeal. They did, however, reduce the effectiveness of the retarders and in many cases caused cars to "slip through" which could lead to disasterous consequences.

Another method of modifying the friction forces between the wheel and retarder shoe is to modify the metals in contact. The first metal used for the retarder shoes in place of steel was ASTM 60 ductile iron. While the incidence of squeal was reduced using the ductile iron the sound intensity was similar once the squeal began. As well these shoes wore at approximately four times the rate of steel shoes. Other types of shoes for example those containing nodules of graphite are being tested. These type of shoes have shown improvements as far as reducing the squeals, however, the life of these shoes is considerably less than those of the usual carbon steel ones.

As mentioned above the addition of damping to the stick-slip system could theoretically eliminate the stick-slip action. For this reason damping of both the retarder beams and the car wheels has been attempted. The retarder beams were damped by adding sand bags and loose sand to the top of the retarder beam. This produced no noticeable change in screech level. The car wheels have been damped by attaching a sheet of viscoelastic damping material to the wheels as well as by applying wooden damping shoes which were pressed against the wheels during the time in which the wheel is being braked. The wooden damping shoes had no effect whatever on the wheel screech while the viscoelastic damping did have some positive effect. While the addition of this damping material may be effective the idea of installing it on the thousands of railroad cars in North America makes it an unworkable solution.

More complicated techniques have also been tried. These include modulating the pressure applied to the retarder shoes so that the screech does not "build-up" and the modification of the retarder beam to alter its mass-stiffness characteristic. Neither of these ideas has shown much promise.

4. Current Investigation

One of the humpyards with squeal problems is the CN Calder Yards in Edmonton. This facility has an enclosure built over the master retarder to control the noise, however, the four group retarders have only partial barriers to reduce the transmission of the high frequency squeal from the retarder-car wheel interaction (see Figure 1). In considering possible solutions to the squeal preliminary investigations of the nature of the noise and vibration from this facility were undertaken. The vibration of the retarder beam as well as the squeal produced were monitored simultaneously. A closeup of the retarder beam is shown in Figure 2 while the system used to monitor the noise and vibration is shown in Figure 3.

The recordings were taken on May 13, 1976 while a group of ten cars were humped 5 consecutive times. These cars were all relatively heavy and all of them squealed excessively.

Recordings were taken for vibration in the longitudinal direction (i.e. parallel to the track) and in the vertical direction. Recordings were also taken in the horizontal but perpendicular to the track. As well the natural vibrations which occurred when the track was impacted in the longitudinal direction were also recorded.

The recordings were later analyzed on a Hewlett Packard 3721A Correlator and displayed on a 3720A Spectrum display and finally plotted on a Hewlett Packard 7044A X-Y Plotter.

Examples of the spectrum analysis are shown in Figure 4 where the results for 3 different cars during the first hump are shown. These results as well as similar ones for other cars and other hump numbers show a very strong correlation between the vibrations of the retarder (measured in the longitudinal and vertical directions) and the noise produced. They also indicate that in most cases a large amount of the energy lies in the 2600-2800 Hz frequency range and not in a wide band of frequencies. Coupled with the fact that one of the natural modes of vibration (in the longitudinal direction) is in the 3000 Hz range leads one to believe that the most critical frequency range is the 2600-2800 Hz band. While the magnitudes of the vibration both above and below this range were sometimes substantial the 2600-2800 Hz band was most often the major contributor to the overall acoustic energy.

5. Possible Solutions

While the investigation to date has not uncovered all apsects of the noise generation mechanism the results obtained point to some possible techniques for controlling the noise.

There are many reasons to believe that the wheel and not the retarder beam is the primary resonant system and therefore responsible for the majority of the noise created. For this reason noise control measures which reduce wheel vibration would appear to be most successful. However, the treating of all railroad car wheels appears to be economically unattractive and one is therefore led to consider the retarder. The fact that there is a strong correlation between the vibration of the retarder and the noise generated leads to the conclusion that there is a strong coupling between the wheel and the retarder. It follows that reducing the retarder vibration will in turn reduce the noise generated.

Previous attempts (mentioned above) at damping the retarder beam by such means as sand bags were unsuccessful. However, the possibility of absorbing the retarder vibration by means of a dynamic vibration absorber [3] could be more effective. This is especially true since the frequency which tends to be the most bothersome is in a fairly narrow band of frequencies (2600-2800 Hz). In this range a vibration absorber should be able to be quite effective in nearly eliminating the retarder vibration. Should the absorber vibration itself cause a squeal it could be readily housed and the noise easily absorbed. It is planned to design and test such a dynamic vibration absorber system in the future.

References

[1] C.A. Brockley, R. Cameron and A.F. Potter, "Friction Induced Vibration", Journal of Lubrication Technology, Vol. 89, pp. 101-108, (1967).

[2] P.J. Remington, M.J. Rudd and I.L. Ver, Wheel/Rail Noise and Vibration Volume I: Mechanics of Wheel/Rail Noise Generation, Urban Mass Transportation Administration Report No. UMTA-MA-06-0025-75-10 (1975).

[3] W.T. Thomson, Theory of Vibrations with Applications, Prentice Hall (1972).

Acknowledgements

The authors would like to express their appreciation to Canadian National Railways, Mountain Division for their help and cooperation in this project. In particular Mr. Ivan King and Mr. Tony Spencer were invaluable in providing the assistance necessary to carry out the testing as well as giving insight into railroad operations.

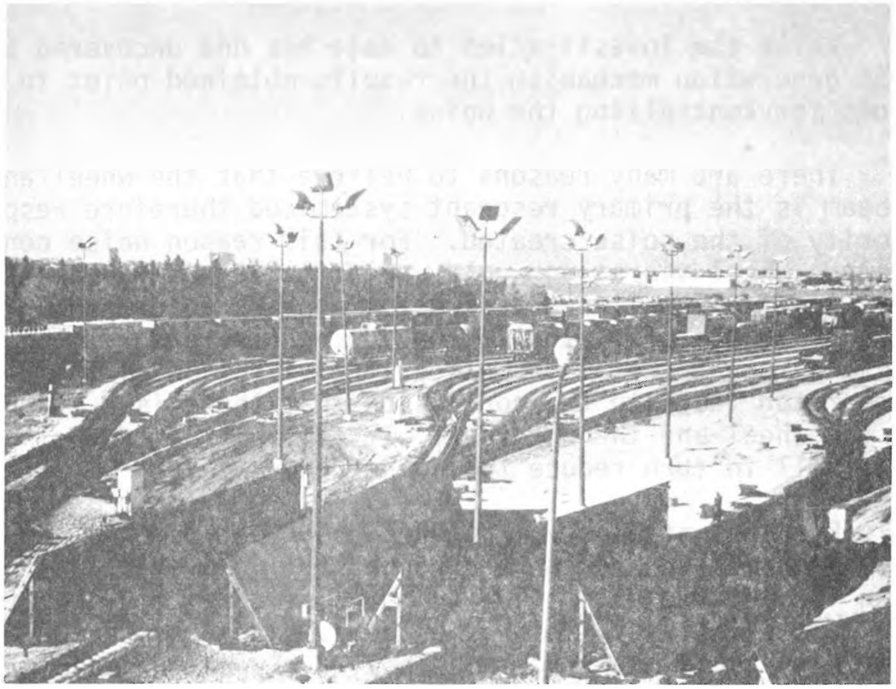


Figure 1

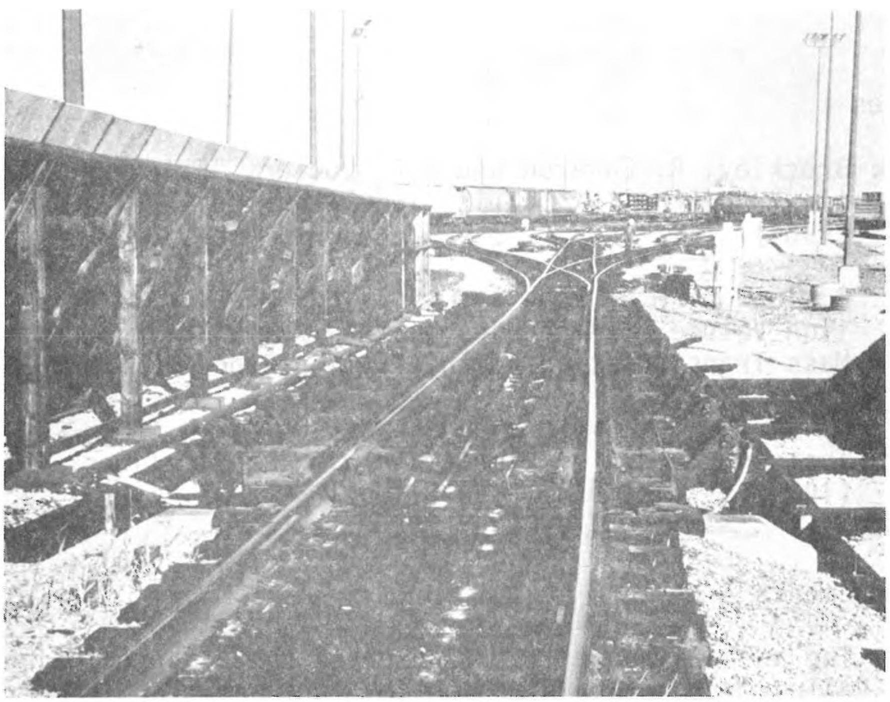
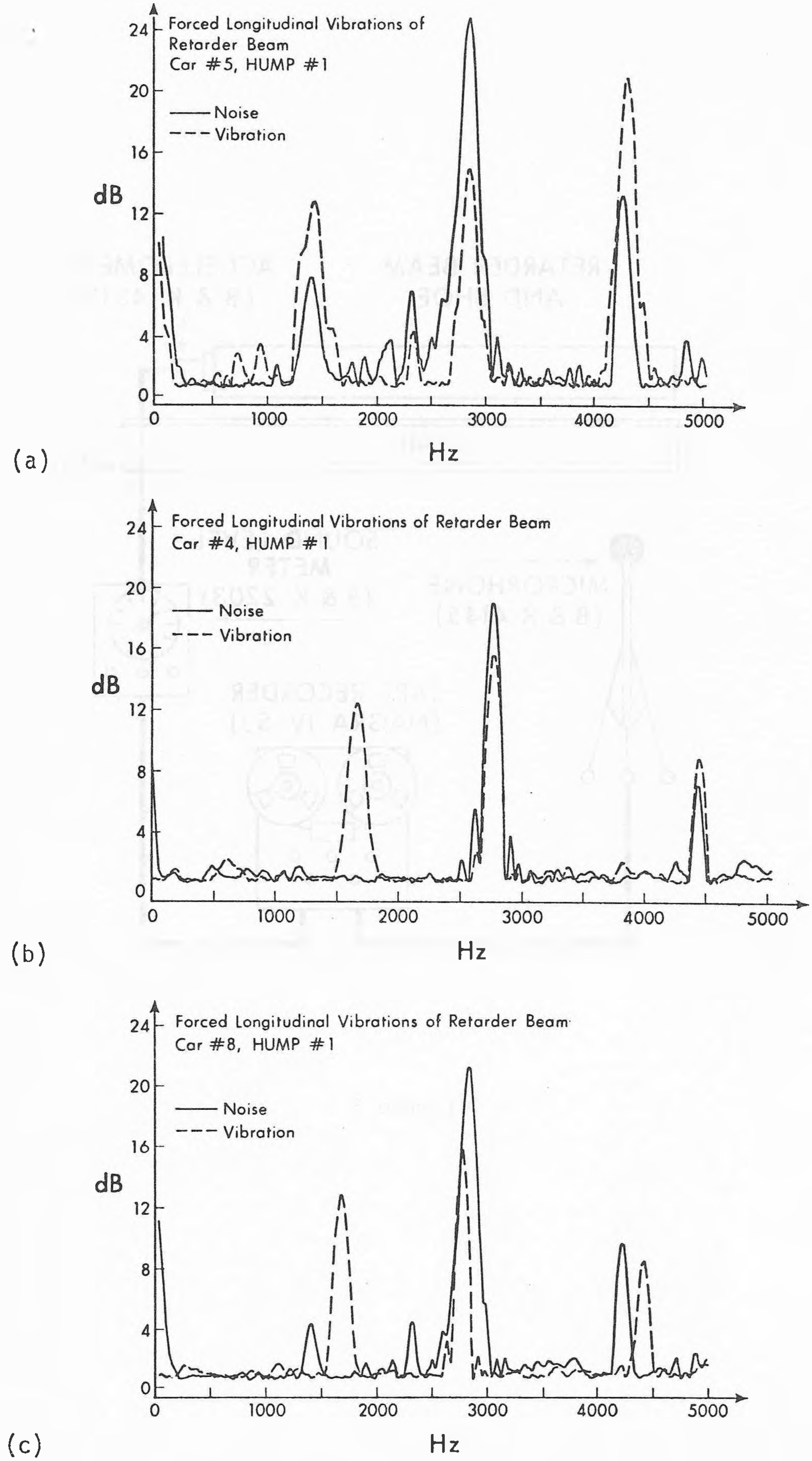


Figure 2



Forced Longitudinal Vibrations of Retarder Beam
Car #5, HUMP #1
Noise
Vibration
dB
Hz
(a)
Forced Longitudinal Vibrations of Retarder Beam
Car #4, HUMP #1
Noise
Vibration
dB
Hz
(b)
Forced Longitudinal Vibrations of Retarder Beam
Car #8, HUMP #1
Noise
Vibration
dB
Hz
(c)

Figure 4

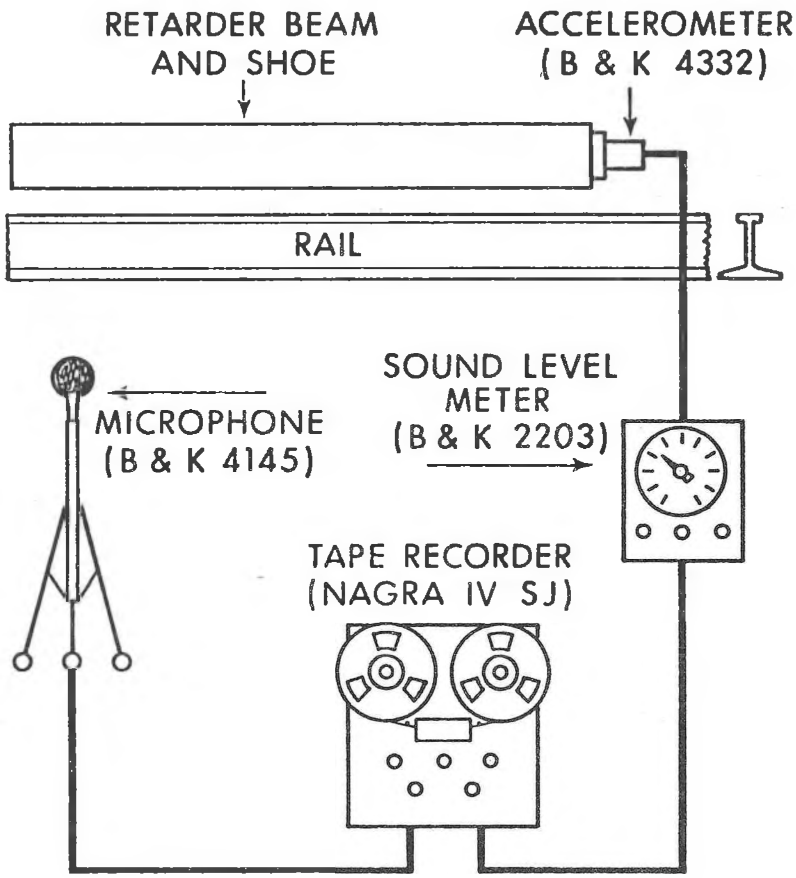

RETARDER BEAM
AND SHOE
ACCELEROMETER
(B & K 4332)
RAIL
SOUND LEVEL
METER
(B & K 2203)
MICROPHONE
(B & K 4145)
TAPE RECORDER
(NAGRA IV SJ)

Figure 3