contradict our analysis because those results pertain to a very different situation than that reported in the paper.

We are surprised to see Martin, et.al., dismiss the validity of our model and analysis at this early date on the basis of one limited parameter study. As these discussers are well aware, from the numerous presentations we have made to them, we have developed several models representing differing levels of detail in the representation of the vehicle dynamics. We are in the process of comparing results from these models and analyses with field test data recently provided us by the AAR Research Center. For those interested in details of this process, certain of these other analyses are described in a companion paper (reference [15] of original paper), and in a Federal Railroad Administration Report [2] now being printed. The field tests are described in [3] and our validation procedure is documented in [4]. Limited quantities of these documents are available from the authors.

In our opinion, the model presented in this paper quite adequately represents the dynamic behavior of typical freight cars with three piece trucks on tangent track. Improvement of the model for studying situations such as derailment mechanics should include representation of rail flexibility, nonlinear creep force laws and possibly three dimensional rail/wheel geometry effects. The first two of these effects are included in an analysis that we have developed for studying rail vehicle curving behavior. The wheel lift and wheelset yaw influence on wheel-geometry effects cited by Martin, et. al., are, in our opinion, relatively unimportant in accurately modeling rail vehicle dynamic behavior.

To our knowledge, ours is the most complete attempt yet to correlate theoretical and experimental results for rail vehicle dynamics. We hope that most people will withhold judgement on the validity of our models until we complete our validation study.

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Influence of Axial Load, Track Gage, and Wheel Profile on Rail Vehicle Hunting[1]

D. J. Reynolds.[2] This paper describes analyses conducted on simplified models of wheelsets. It is excellent that this work should be done, and it is evident that the authors' linearization of hunting conditions by the describing-function method gives us all new insights into this aspect of vehicle behavior. However, the title of the paper and the summary, mentioning and discussing "Rail-Vehicle Hunting," seem to extrapolate this computer analysis to the point of seeking to give immediate practical guidance. For instance, in the summary we read, "For freight car applications, coulomb friction in the suspension (e.g., constant contact side bearings) may act to increase the range of speeds over which hunting will not occur and may permit operation at higher speeds for extremely straight track."

Noting that by "extremely straight track" the authors mean "track extremely free from irregularities," one asks why the "mays" in this statement. It is common knowledge in the industry that dry friction so acts, and quite apart from the products on the market and in daily use, the action has been published many times [1, 2].[3]

Turning to the text to find the support for this statement about constant contact side bearers, Table 1 shows that the only breakout torsional friction, considered for a light car, is 1.916 × 0.00122, × 2, or 4675 ft.lbs. This is the appropriate figure for a moderately lubricated center plate alone [3]. Any effective side bearer would add at least twice as much resistance again. It would appear that the statement regarding constant contact side bearers may be a complete extrapolation.

In Fig. 12, it would appear that instability occurs at 125 ft./s. over a wide range of axle loads stated as 8E, 6E, 4E, and 2E. This is contrary to field experience where it almost is axiomatic that increased loading raises the critical speed, as is confirmed by the authors' Fig. 16(*b*), and this casts doubt on the usefulness of the linear-suspension model initially used. It also appears that for the first three loads, vibration will increase to an amplitude of 0.48-inch approximately, where it remains limited by flange action, but for the load of 2.00 E, there is no stability whatever. This is also contrary to field experience; unless a wheel is completely weightless or completely flangeless, it is bound to receive some restraint when the flange touches the rail. Possibly relevant to these points are the substitutions Kg^+ and $f_L{}^1$, given at the top of page 4. A check on the preceding equations 12 and 14 shows that values of $Kg^+ = Kg(1 + K_y/K_r)$ and $f_L{}^1 = f_L(1 + K_y/K_r)$ would be more rational than the values given.

I find the statement on page 3, "For the cases of interest here, the creep force is small compared to the contact force," rather puzzling. In the first place, for small amplitudes, without flange contact, the peak creep force calculates to be at least twice the peak lateral contact force, even for worn wheels, even for low values of the creep coefficient. Secondly, these forces are sinusoidal and approximately 90 deg out of phase with each other. Thirdly, the resultant approximation, $y_r = K_g/K_r \cdot y$ is not further used in the text.

For the more sophisticated model, a lateral suspension and a yaw suspension with a spring and a friction element in series was used. This contrasts with the car model of the companion paper, 76-WA/RT-2, where the yaw suspension was modeled by friction only, which would seem to be the condition obtained on any car with a center bowl, and I have submitted discussion on the very interesting results obtained there.

"Derailment" is mentioned several times, and it would be helpful if the authors could state what criterion is used to indicate this. All the graphs seem to contemplate side-to-side motion of up to two inches continuously maintainable provided the forward speed is below a critical velocity. For instance, in Fig. 12, the worst case is permitted

[1]By D. N. Hannebrink, H. S. H. Lee, H. Weinstock and J. K. Hedrick JOURNAL OF ENGINEERING FOR INDUSTRY, TRANS. ASME, Series B, Vol. 99, Feb. 1977, 99, 77, pp. 186–195.

[2] Asst. Mgr. Research and Tests, Southern Railway Company, Mem. ASME.

[3] Numbers in brackets designate References at end of discussion.

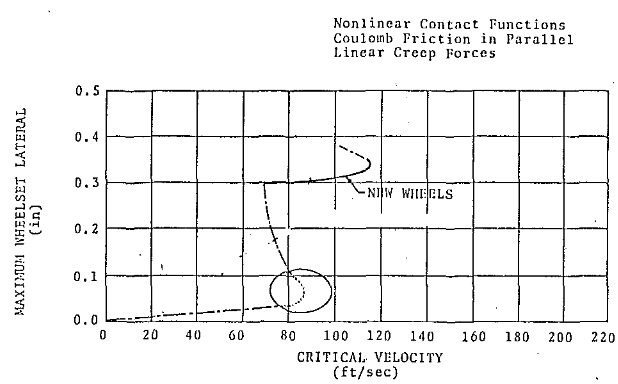

Fig. 1 Freight vehicle limit cycle amplitudes: nonlinear suspension

a half-amplitude of 1-in. if the velocity is below 55 ft./s., and in Fig. 16(b), the more sophisticated model, the loaded wheel is permitted the same at speeds up to 150 ft./s. A stable excursion of this magnitude, at any speed, does not appear realistic.

References

1 Cain, *Vibration of Rail & Road Vehicles* (1940), p. 171.

2 Matsudaira, *Proceedings of the Institution of Mechanical Engineers 1965–66*, p. 62.

3 TTD Harmonic Roll Series, Vol. 2, p. 69.

Author's Closure

The authors wish to thank Mr. Reynolds for his interest and comments on our paper.

His first comment is one that we feel is very interesting and deserves further investigation. He states it is common knowledge that dry friction acts to increase the critical speed. We are not sure that this is common knowledge, and in fact always true, for example in the companion paper, [1][1] the opposite conclusion is reached. Wickens [2] points out that as a rigid connection between axles for a two axle car is approached the critical speed goes to zero, thus too much friction can reduce the critical speed for a rigid car. The discussor's Fig. 1 *is a conjecture by the authors* concerning the effects of Coulomb friction. This figure is an extension to low amplitudes of numerical results presented in [3]. The interesting point here is that lateral limit cycle type oscillations exist even at very low speeds, however, the hunting amplitude is so low as to be barely detectable. For speeds greater than 75 ft/s, Fig. 1 shows multiple solutions. Thus between 75 ft/s and 85 ft/s the limit cycle amplitude depends upon the magnitude of the initial disturbance, if the initial disturbance is small the hunting amplitude is small; however, if the disturbance is large the wheelset will hunt at a dangerously large amplitude. It is this multiple solution possibility that we feel needs further investigation.

The reviewers' comment that increased axle-load always increases the critical speed is shown in Fig. 16(b) of our paper for freight car trucks. However, this is not necessarily true for passenger or locomotive trucks, where more linear type suspension elements are used. Fig. 12 applies to simple flexibly suspended wheelset and is more typical of passenger truck behavior.

The expressison for k_g* and f_L' are correct as is, for large displacements the rail flexibility limits the effective lateral stiffness with respect to inertial space produced by wheel rail contact to a value less than the track stiffness.

The lateral suspension was modeled as a parallel spring/coulomb friction combination while the yaw suspension was modeled, as he points out as, as a series combination. This was done since the wheelset model must include both the compliance of the truck, and the coulomb friction at the centerplate whereas the comparison papers' nine degree of freedom system [1] models these components separately.

Our discussion of "derailment" is qualitative and identifies those regions, where the analyses indicate growing and unbounded oscillations for the simple models, quantitative results on derailment would require a much more sophisticated model.

[1] Numbers in brackets refer to References at end of closure.

References

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3 Hedrick, J. K., Cooperrider, J. K., and Law, E. H., "The Application of Quasi-Linearization Techniques to Rail Vehicle Dynamic Analyses," Final Report, DOT-TSC-FRA, Jan. 1977.