TP 14075E

Performance Testing of the Resco Steered Frame Freight Car Truck at the AAR Test Track, and Comparisons with Standard and Premium Three-Piece Trucks

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The Transportation Development Centre does not endorse products or manufacturers. Trade or manufacturers names appear in this report only because they are essential to its objectives.

Since some of the accepted measures in the railway industry are imperial, metric measures are not always used in this report.

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Un sommaire français se trouve avant la table des matières

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16. Abstract

This report describes tests conducted at National Research Council Canada's test facilities in Ottawa, Ontario, at the Association of American Railroads' (AAR) Transportation Test Facility in Pueblo, Colorado, and in the laboratories of Queen's University in Kingston, Ontario. The tests at Pueblo were conducted as part of a larger, internal AAR program of research that also encompassed several other trucks. The Pueblo results reported here are primarily those for the Resco Steered Frame Freight Car Truck but also include, where applicable, the results from testing of a standard three-piece truck and some results from testing of a "premium" truck.

The tests showed that the Resco truck outperformed all other premium trucks in all of the test categories identified except one. The exception is for the rolling resistance on tangent track, where the results equaled those of the premium truck. The economic benefits of the Resco truck were shown to be, even at a very conservative estimate, able to justify a reasonable increase in cost over the standard truck, while producing a very attractive return on investment.

Some design deficiencies were identified, primarily in the realm of physical clearances within the truck, and these need to be addressed before continuing to the next phase of testing. All of the modifications to the prototypes, required to eliminate the clearance problems, can be made very simply. Once these have been addressed, the trucks should be operated in an endurance cycle on the AAR's FAST test track at Pueblo and then exercised under a rail car in a railway environment.

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16. Résumé

Des essais ont été effectués dans les installations du Conseil national de recherches du Canada, à Ottawa, en Ontario, au centre d'essais ferroviaires de l'Association of American Railroads (AAR), à Pueblo, au Colorado, et dans les laboratoires de l'Université Queen's, à Kingston, en Ontario. Les essais de Pueblo s'inscrivaient dans un programme interne de recherche plus considérable de l'ARR, auquel étaient soumis plusieurs types de bogies. Les résultats des essais menés à Pueblo sont principalement ceux du bogie orientable Resco pour wagon de marchandises; sont également présentés les résultats comparatifs d'un bogie classique et d'un bogie de qualité supérieure, à trois éléments.

Les essais ont démontré que le bogie Resco avait des performances supérieures à celles de tous les autres bogies, et ce, pour toutes les catégories de tests sauf ceux mesurant la résistance au roulement sur voie en alignement droit, où il a obtenu des résultats équivalents à ceux du bogie de qualité supérieure. Selon les résultats globaux, les avantages économiques du bogie Resco, même après évaluation très prudente, justifient son coût raisonnablement plus élevé que celui du bogie classique, et démontrent un rendement très intéressant de l'investissement requis.

Des défauts de conception ont été observés, particulièrement des interférences physiques des éléments constitutifs. Ces défauts doivent être corrigés avant que les essais du bogie Resco entrent dans la phase suivante. Toutes les modifications aux prototypes pour éliminer ces problèmes d'interférence sont très faciles à réaliser. Une fois ces problèmes corrigés, les bogies devront être soumis à des épreuves d'endurance sur le circuit FAST de l'AAR, au Colorado, puis installés sur un wagon pour essais dans un environnement d'exploitation ferroviaire.

17. Mots clés Bogie orientable pour wagon de marchandises, bogie à trois éléments, Association of American Railroads, centre d'essais de l'AAR, résistance au roulement, équilibrage des charges sur les essieux, efforts sur la voie en courbe, gauchissement du bogie, stabilité à haute vitesse, essais sur voie présentant des défauts	18. Diffusion Le Centre de développement des transports dispose d'un nombre limité d'exemplaires.

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Executive Summary

Tests were conducted on a pair of Resco Engineering Steered Frame Freight Car Trucks mounted under a lightweight, aluminum, coal gondola. The tests were conducted under both empty and loaded conditions. The loaded condition provided for nominally 286,000 lb. (130,000 kg) at the head of the rail. The tests measured truck weight, rolling resistance, wheel load equalization, curving forces, truck warping, high-speed stability, and behaviour on perturbed track. Similar tests were also done using the same car, but with standard three-piece freight car trucks, for comparison purposes, in most of these test regimes.

The Association of American Railroads (AAR) standard for 36 in. (914 mm) wheels has been for the axle load not to exceed 65,750 lb. (29,886 kg), or a total car weight of 263,000 lb. (119,545 kg). Typically this meant that the suspension supported 44,000 lb. (20,000 kg) in the empty condition and 246,000 lb. (111,818 kg) in the loaded condition, a ratio of 5.6:1 between loaded and empty. There has been a move afoot recently to increase the axle load to 71,500 lb. (32,500 kg) for a total car weight of 286,000 lb. (130,000 kg). There has also been a move towards lighter car structures. The car utilized in this test has a sprung mass (i.e. without trucks and wheelsets) of 25,000 lb. (11,364 kg) when empty and 269,000 lb. (122,272 kg) when loaded, a ratio of 10.76:1, almost twice the challenge that the earlier cars presented.

The test car was loaded from a previous test in which it had utilized standard three-piece trucks. In that configuration it had a total weight of 286,509 lb. (130,231 kg). After the Resco trucks, and carbody adapters, were mounted under the car it was weighed again and found to be 282,939 lb. (128,609 kg) This shows that a pair of Resco trucks, including carbody adapters, weighs 3,570 lb. (1,623 kg) less than a pair of standard three-piece trucks. The carbody adapters weigh a total of 1,300 lb. (591 kg). These were only necessary to provide the required interface between the truck and the carbody. If the carbody were designed with this interface integral to it then the weight savings could be 4,870 lb. (2,214 kg) per car. Even taking into account that some weight might have to be added to the car frame to make it suitable (not shown to be required at this time), it is reasonable to estimate that at least 4,500 lb. (2,045 kg) per car will be saved. In most instances of bulk commodity movements this would mean that 4,500 lb. (2,045 kg) more of the commodity could be loaded into each car.

Rolling resistance measurements showed that the Resco truck has virtually the same resistance on a curve as it does on tangent track. This is a very favourable result when compared with standard trucks and with other premium performance trucks. A standard truck, in good condition, will produce a rolling resistance of 1.8 lb./ton (7.3 N/tonne) on tangent track and this will increase by 0.8 lb./ton (3.24 N/tonne) for each degree of the curve that it operates on. The tangent track resistance is often found to be as high as 2.3 lb./ton (9.32 N/tonne), with the same increase in the curves. For the premium trucks the typical value found is approximately 1.6 lb./ton (6.49 N/tonne) on tangent track and an increase of 0.4 lb./ton (1.62 N/tonne) for every degree of curve. The Resco truck gave 1.62 lb./ton (6.57 N/tonne) on tangent track and an increase of 0.06 lb./ton (0.24 N/tonne) for every degree of curvature. For a 12 curve, therefore, the standard truck under a 286,000 lb. (130,000 kg) car will produce 1,630.2 lb. (7,269 N) of rolling resistance. On the same curve, a good premium truck will produce 915.2 lb. (4,081 N) of resistance (56.1% of the standard) and the Resco truck will produce 331.8 lb. (1,479.5 N) (20.35% of the standard). Five cars equipped with Resco trucks could be hauled through the 12 curve by virtually the same force as one car with standard trucks.

Wheel load equalization was measured by lifting one wheel out of the plane of the other three and measuring the loads on all four wheels. The standard truck started out with two wheels carrying approximately 85% of the average load and two wheels carrying approximately 115%, on level track. As one wheel was lifted these tended first to equalize and then to reverse the unbalance until, at 3 in. of lift, the wheels supported 155%, 115%, 75% and 55% of the average load. Each wheel changed load by 60 to 70%. As the wheel was then lowered there was a large amount of 'hysteresis' evident in the paths that the load/deflection curves took, but at level track the original condition was almost exactly restored. By comparison, the Resco truck started out at between 96% and 103.4% of the average load and changed to 128.1%, 101.9%, 98.1% and 71.4% of the average load at 3 in. of wheel lift. Each wheel changed load by approximately 25%. As the wheel was lowered again the loads followed virtually the same paths down again, with no discernible hysteresis. Clearly the Resco truck behaved better than a standard truck on this criterion. No data was available for other premium trucks. As their structure and suspensions are very similar to the standard truck's, it was felt that they would probably show similar results and no testing was done on them.

Curving forces were measured on various curves in the Wheel/Rail Mechanism (WRM) loop at the test centre. The table below shows the results of the lateral force measurements for the Resco truck compared with those for a typical premium truck and the baseline truck.

Curvature	4	7.5	10	12
Baseline	6,500 lb.	10,000 lb.	14,000 lb.	17,500 lb.
Typ. Premium	1,000 lb.	4,000 lb.	6,500 lb.	11,000 lb.
Resco	-500 lb.	2,000 lb.	3,000 lb.	4,000 lb.

Table 1. Comparison of Lateral Forces on Curves

Again, the substantial performance improvement of the Resco truck over both the standard truck and the premium truck is quite evident. The negative lateral force at the mild curvature is an interesting phenomenon. It occurs because the wheel alignment is so good that there is virtually no lateral force exerted as a result of that, and the residual measured is primarily the result of the spin creep term at the wheel/rail interface, due to the conicity at the point of contact. The negative sign indicates that this force is in the direction to bring the rail heads in, rather than to spread them apart.

Truck warping is the cause of much of the deficiency in performance of the standard three-piece truck. Because the Resco truck differs from a standard truck in so many ways, the level of truck warping was measured. On a 12 curve it was found that one sideframe had warped less than 0.12 in. relative to the other, for a shear distortion of less than 0.1 . This is an extremely low value, demonstrating that the truck has more than adequate shear stiffness between the sideframes, to prevent undesirable warping. Typically, values of 1 in. or more can be measured on some standard trucks in steep curves, indicating very high values of truck warping.

The high-speed stability of the Resco truck surpassed any that has been witnessed on a freight car truck before. In the loaded condition the car was tested as far as 105 mph (168 km/h) with absolutely no sign of truck instability. The lateral ride quality on the car was surprisingly good. A subway car with a ride index of 2.5 hrs ISO (considered to be good), produced wideband accelerations of 0.03g rms at 50 mph (80 km/h) on well maintained subway track. The freight car with the Resco steered truck produced 0.027g rms at 50 mph (80 km/h) and 0.055g at 105 mph (168 km/h), on the test centre track. In the empty condition the car achieved 90 mph (144 km/h) without visible signs of truck hunting, but at speeds above 80 mph (128 km/h) the lateral accelerations were rising more rapidly than at speeds below that – usually taken to be a sign of impending or established hunting. It is unclear at this time what the cause of this was, but it is hypothesized that it may have been due to insufficient lateral clearance in the suspension. Alternatively, it could have been caused by the orientation of the dampers, which encouraged lower centre roll motions. In any event, although the empty condition high-speed performance might very well be improvable, it has already surpassed the industry requirements, and the capabilities of the standard truck and all other trucks tested under this very challenging car. The standard truck showed instability at around 45 to 50 mph (72 to 80 km/h) in the empty condition. Some of the premium trucks did not achieve 70 mph (112 km/h).

The perturbed track tests included perturbations of vertical alignment, lateral alignment and a combination of curve, vertical and lateral alignment. These tests were really a test of the suspension system's capability. After some initial difficulties with ineffective dampers the Resco truck's suspension system showed the potential to handle these "worst case" perturbations but was not able to complete the entire testing program. The vertical perturbation test, known as the "pitch and bounce" test, was concluded at 50 mph (80 km/h), without passing completely through the resonance velocity zone. The vehicle motions in this direction were quite well controlled and it appeared that the test would be completed successfully. At 50 mph (80 km/h), however, it was discovered that there was metal-to-metal contact being made at the bottom of each vertical cycle. Subsequent investigation revealed that an interference condition existed, preventing the suspension from using all of its intended travel. Similarly, in the lateral perturbation test, known as the "yaw and sway" test, the motions at first appeared to be well controlled, but a roll began to occur about a point somewhere below the top of rail. This condition had inadvertently been created when the dampers were inclined in order to provide sufficient lateral control for the yaw motions. As a result of this, large lateral motions were created at the truck bolster and it was found that a physical interference existed, preventing the suspension from using all of its intended travel in that direction. The unintended lateral contact exerted an impact moment on the truck frame, which caused brief but severe reductions in the wheel vertical load. It was decided that it would be wise to discontinue this type of testing until these interference and configuration conditions had been eliminated.

Based on the rolling resistance measurements made in these tests, it is a relatively simple matter to calculate the fuel savings that accrue from the use of the steered truck in place of standard trucks in a particular service. This calculation was done for a train of 286,000 lb. (130,000 kg) cars operating in two quite different scenarios. The scenarios were real ones, taken from route profiles for two different railroads. In one case the track was extremely curvy and with steep grades for about 420 mi. (672 km), whereas in the other case, it was very mildly curved and less steeply graded for about 620 mi. (992 km). For the steeply curved route the saving was 21.7% of the total fuel used, or approximately 3,129.5 US gal. (11,892 L) of fuel per car per 100,000 mi. (160,000 km) traveled. For the less curved route the saving was 20.7% or approximately 1,152.4 US gal. (4,379 L) of fuel per car per 100,000 mi. (160,000 km) traveled.

The increased load-carrying capacity of the car, when using the Resco truck, can easily be translated into increased revenue for the railway. If the railway charges US$15 per 1,000 ton-mi. (1,455 tonne-km) to transport a commodity, then the increased revenue accruing from the reduced truck mass will be between US$1,350 and US$1,800 for every 100,000 mi. (160,000 km) traveled. This increase in revenue accrues with no increase in operating expenses. In fact, as we have seen, the overall operating expense for movements in cars fitted with these trucks is less than on standard trucks. A 2% increase in productivity, therefore, creates an increase in revenue of US$1,800 for every 100,000 mi. (160,000 km) traveled.

It was shown that the reduced drawbar pull for a train of cars could increase productivity by 12% on steep, curvy routes and 4.85% on flat, tangent routes. This translates to revenue increases of between US$4,365 and US$10,800 for every 100,000 mi. (160,000 km) traveled.

Although no wheel wear measurements were possible within the scope of the testing conducted here, the AAR has software that can predict the savings in wheel and rail wear costs due to the improvement in wheel/rail alignment achieved. The values estimated from this software, based on the test results reported here, were US$250 in wheel wear and US$1,000 in rail wear for a curvy track and US$120 in wheel wear and US$480 in rail wear for a less curved track. These values are based on a total distance traveled of 100,000 mi. (160,000 km). Obviously they can be prorated for different travel distances.

The net present value (NPV) of the truck can readily be calculated if there is a required return on investment (ROI) and the revenue increase (or expense savings) is known. If we estimate that the truck produces a net saving and/or revenue increase of US$1,800/year, after account is taken of any change in costs of maintenance relative to a standard truck, and that a capital amortization period of 10 years is required, with an ROI of 30%, the NPV is US$5,690. This means that the railway could purchase the trucks at a premium of US$5,690 over the price of standard trucks, and the additional investment would be retired in 10 years while producing an ROI of 30%. A slightly less ambitious requirement for the ROI could produce a higher value for the NPV. Following the amortization period the ROI would increase slightly, under the assumption that the trucks remained in service, which they would be expected to do, the normal service life of a freight car truck being in excess of 30 years. The data presented suggests that US$1,800/year is a very conservative estimate for the overall annual revenue/expense savings produced by the Resco truck, depending on the routes upon which it is operated. The premium price, of course, will depend very much upon the quantities of trucks sold, the manufacturing techniques used, and other factors. Given the reduced weight of the truck it is felt likely that, despite the added machining required by its nature, the total increase of cost of US$5,000 to US$6,000 is quite attainable, within a reasonable period from its introduction.

This summary would not be complete without a short discussion of the flexibility of the design concept to various techniques of manufacture. The prototype trucks were built using a hybrid "cast-fabricated" structure. Wherever there are complex shapes and/or difficult stress concentrations, the structure was cast and then the cast pieces were welded into fabricated sections of simple geometry, the welds being located in areas of relatively low stress. Because of this design technique the trucks were able to be built quite readily in an extremely small, low-tech machine and fabricating shop. This demonstrated the feasibility and attractiveness of this type of construction, and was essential in keeping the costs of prototype construction within reasonable bounds. It also showed that the technique creates the situation where a multitude of manufacturing facilities can potentially provide serviceable railway trucks to the North American industry. This can greatly improve the potential for competitive pricing of the trucks. On the other hand, if a one-piece cast sideframe is required, due to a customer policy or specification concerning such things, nothing prevents such a structure being utilized and no weight penalty would be accrued.

The final subject of this summary must be a discussion of the path forward from here. The potential of the truck to improve railway operations and "bottom line" performance is quite clear. A number of relatively minor design issues were identified and these should be addressed as soon as possible, to show that they can be overcome without compromise to the very attractive performance benefits achieved so far. Once this has been done, the prototypes should be subject to "endurance" type testing. It is suggested that an appropriate first step in this direction would be to install the trucks under a freight car in the train conducting trials around the FAST loop at Pueblo and to monitor them for wear and damage. Following this, if all indications are positive, a small number should be introduced onto a railway service, preferably one having sharp curvatures, and similar monitoring should be done.

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Sommaire

Une paire de bogies orientables de la société Resco Engineering, pour wagon de marchandises, ont été mis à l'essai sous un wagon tombereau léger en aluminium destiné au transport du charbon. Les essais ont été menés en charge et à vide, dans le premier cas avec une charge nominale de 286 000 lb (130 000 kg) sur le champignon du rail. Les paramètres mesurés étaient la masse du bogie, sa résistance au roulement, l'équilibrage des essieux, les efforts sur la voie en courbe, le gauchissement, la stabilité à haute vitesse et le comportement sur voie présentant des défauts. Aux fins de comparaison, des essais similaires ont été effectués avec des bogies classiques à trois éléments, dans des conditions correspondant à la plupart des régimes d'essais.

Pour des roues de 36 po (914 mm), la norme de l'Association of American Railroads (AAR) exigeait que la charge sur l'essieu ne dépasse pas 65 750 lb (29 886 kg), ou que la masse totale du wagon ne dépasse pas 263 000 lb (119 545 kg). Habituellement, pour ces valeurs, la charge imposée à la suspension est de 44 000 lb (20 000 kg) à vide et de 246 000 lb (111 818 kg) en charge, soit un ratio de 5,6:1 entre ces deux conditions. On a récemment songé à augmenter la charge à l'essieu à 71 500 lb (32 500 kg), pour une masse totale du wagon de 286 000 lb (130 000 kg). Les chercheurs ont également considéré l'allègement de la structure du wagon. Celui utilisé pour les essais avait une masse suspendue (sans bogies ni essieux instrumentés) de 25 000 lb (11 364 kg) à vide et de 269 000 lb (122 272 kg) en charge, soit un ratio de 10,76:1, près du double de celui des wagons plus lourds.

Le wagon d'essai était déjà en charge par suite d'un essai précédent avec bogies classiques à trois éléments. Dans cette configuration, la masse totale du wagon était de 286 509 lb (130 231 kg). Le pesage du wagon après installation des bogies Resco et des adaptateurs nécessaires a donné une masse de 282 939 lb (128 609 kg). Cela permet d'établir que les deux bogies Resco, y compris les adaptateurs, pèsent 3 570 lb (1 623 kg) de moins que les bogies classiques à trois éléments. Les adaptateurs seuls comptent pour 1 300 lb (591 kg). Ils servent uniquement d'interface, nécessaire entre le bogie et la caisse du wagon. Une interface intégrée donnerait une économie de masse de 4 870 lb (2 214 kg) par wagon. Même en ajoutant une masse supplémentaire au châssis pour le rendre conforme aux besoins (solution inutile à ce stade-ci), on peut raisonnablement prévoir, par wagon, un allégement d'au moins 4 500 lb (2 045 kg) environ. En transport de marchandises en vrac, cela signifie, dans la plupart des cas, une charge utile additionnelle possible de 4 500 lb (2 045 kg) pour chaque wagon.

Les mesures de la résistance au roulement ont démontré que le bogie Resco, à ce chapitre, obtient pratiquement les mêmes résultats en voie courbe qu'en alignement droit. Cette performance est très intéressante si on la compare avec celle des bogies classiques et d'autres bogies de qualité supérieure. Un bogie classique en bon état aura une résistance au roulement de 1,8 lb/tn (7,3 N/t) sur voie en alignement droit, valeur qui augmentera de 0,8 lb/tn (3,24 N/t) pour chaque degré de courbure. La résistance au roulement sur voie en alignement droit atteint souvent 2,3 lb/tn (9,32 N/t), l'augmentation en courbe étant celle précédemment mentionnée. Dans le cas des bogies de qualité

supérieure, la résistance au roulement est généralement d'environ 1,6 lb/tn (6,49 N/t) sur voie droite, avec augmentation de 0,4 lb/tn (1,62 N/t) pour chaque degré de courbure. Le bogie Resco a donné une résistance au roulement de 1,62 lb/tn (6,57 N/t) sur voie droite et une augmentation de 0,06 lb/tn (0,24 N/t) par degré de courbure. Dans le cas d'une courbe de 12 , la résistance au roulement d'un bogie classique installé sous un wagon de 286 000 lb (130 000 kg) est de 1 630,2 lb (7 269 N). Sur la même voie courbe, un bogie de qualité supérieure offrira une résistance de 915,2 lb (4 081 N) (56,1% de la résistance mesurée avec bogie classique) et le bogie Resco donnera une résistance de 331,8 lb (1 479,5 N) (soit 20,35 % de la résistance avec bogie classique). Tracter dans une courbe de 12 cinq wagons équipés de bogies Resco a nécessité la même force que pour un seul wagon sur bogies classiques.

Pour évaluer la répartition des charges, les chercheurs ont soulevé une roue hors du plan des trois autres roues, puis mesuré la charge sur les quatre roues. Sur voie en terrain plat, au début, deux roues portaient environ 85 % de la charge moyenne et les deux autres, environ 115 % de cette charge. En soulevant une roue, la charge supportée par les trois autres avait d'abord tendance à s'équilibrer puis à inverser le déséquilibre jusqu'à ce que, pour un soulèvement de 3 po, la répartition soit de 155 %, 115 %, 75 % et 55 % de la charge moyenne. Pour chaque roue, la charge variait de 60 à 70 %. À l'abaissement de la roue, les courbes charge-déformation ont montré beaucoup d'hystérésis, mais la condition initiale était presque entièrement rétablie lorsque la roue était revenue au même niveau que les autres. En comparaison, le bogie Resco a commencé par indiquer 96 % et 103,4 % de la charge moyenne pour passer ensuite à 128,1 %, 101,9 %, 98,1 % et 71,4 % de la charge moyenne pour un soulèvement de 3 po. La charge sur chaque roue variait d'environ 25 %. À mesure que la roue redescendait au niveau des autres roues, les charges présentaient pratiquement le même comportement, sans hystérésis perceptible. Sur ce critère, le bogie Resco avait une performance nettement clairement supérieure à celle du bogie classique. Aucune donnée n'était disponible pour les autres bogies, de qualité supérieure. Comme ces derniers présentent une construction et une suspension similaires à celles des bogies classiques, on a estimé qu'ils produiraient probablement des résultats semblables, et ils n'ont par conséquent été soumis à aucun essai.

Les efforts exercés sur le rail, en courbe, ont été mesurés dans diverses courbes de la boucle WRM (Wheel/Rail Mechanism) du centre d'essais ferroviaires de l'AAR. Le tableau ci-après présente les valeurs des efforts transversaux mesurés pour divers degrés de courbure, avec un bogie classique, un bogie de qualité supérieure et un bogie Resco.

Bogie	4	7,5	10	12
Classique	6 500 lb	10 000 lb	14 000 lb	17 500 lb
Qualité sup.	1 000 lb	4 000 lb	6 500 lb	11 000 lb
Resco	-500 lb	2 000 lb	3 000 lb	4 000 lb

Tableau 1. Comparaison des efforts transversaux en courbe

De nouveau, l'amélioration substantielle de performance du Bogie Resco par rapport au bogie classique et au bogie de qualité supérieure est très évidente. L'effort transversal négatif en courbe douce représente un phénomène intéressant. La valeur négative est due à la qualité satisfaisante de la position de la roue par rapport au rail : pratiquement aucun effort latéral n’est exercé. La valeur résiduelle résulte principalement du moment de cheminement du rail à l’interface roue-rail, causé par la conicité du point de contact. Le signe négatif indique que cette force agit de manière à réduire l’écartement des rails plutôt qu’à l’élargir.

Le gauchissement explique en grande partie la performance inférieure du bogie classique à trois éléments. Les nombreuses différences entre le bogie Resco et le bogie classique ont incité à mesurer la valeur de ce gauchissement. En courbe de 12 , un des longerons avait une différence de gauchissement inférieure à 0,12 po par rapport à l’autre, ce qui donnait une déformation de cisaillement inférieure à 0,1 . Cette valeur très faible démontre donc, entre les deux longerons de bogie, une rigidité plus que suffisante pour empêcher qu’un gauchissement indésirable se produise. Habituellement, la différence peut être de 1 po ou plus avec certains bogies classiques en courbe prononcée, ce qui indique des valeurs très élevées de gauchissement.

En stabilité à haute vitesse, le bogie Resco a surpassé tous les bogies pour wagons de marchandises. L’essai en charge a été poussé jusqu’à 105 mi/h (168 km/h) sans aucun signe d’instabilité. La qualité transversale du roulement était étonnamment bonne. Une voiture de métro présentant un indice de roulement ISO de 2,5 h ISO (jugé bon) a produit des accélérations efficaces en large bande de 0,03 g à 50 mi/h (80 km/h) lors de circulations sur voie bien entretenue. Le wagon de marchandises monté sur bogies Resco orientables a donné des accélérations de 0,027 g à 50 mi/h (80 km/h) et de 0,055 g à 105 mi/h (168 km/h), sur la voie du centre d’essais. Le wagon vide a atteint des vitesses de 90 mi/h (144 km/h) sans signe visible d’oscillations transversales du bogie, mais, au-dessus de 80 mi/h (128 km/h), les accélérations transversales augmentaient plus rapidement qu’aux vitesses inférieures, ce qui est habituellement interprété comme un signe de mouvement oscillant établi ou imminent. La cause de ce phénomène n’est pas encore claire, mais on a émis l’hypothèse selon laquelle il serait causé par un débattement latéral insuffisant de la suspension. L’orientation des amortisseurs, qui favorisait le roulis autour d’un axe plus bas est une autre cause probable. Quoi qu’il en soit, même si la performance à vide à haute vitesse peut très bien être améliorée, elle a déjà surpassé les exigences de l’industrie, ainsi que les capacités du bogie standard et de tous les autres bogies essayés sur ce wagon très intéressant du point de vue de ses caractéristiques. Le bogie classique a démontré de l’instabilité autour de 45 à 50 mi/h (72 à 80 km/h) lorsque le wagon n’est pas chargé. Certains bogies de qualité supérieure n’ont pu atteindre 70 mi/h (112 km/h).

Des essais ont été menés sur voie comportant des défauts d’alignement vertical et d’alignement transversal, de même que des défauts combinés de courbure et d’alignements vertical et transversal, qui ont mis à l’épreuve les qualités du système de suspension. Après des difficultés initiales reliées à l’inefficacité des amortisseurs, le bogie Resco a démontré qu’il avait le potentiel de supporter les pires défauts de voie mais qu’il ne pourrait pas franchir le programme complet des essais. L’épreuve des défauts dans l’axe vertical, pour vérifier la maîtrise du tangage et des rebondissements, a été effectuée à 50 mi/h (80 km/h), sans toutefois entrer complètement en régime de résonance. Les mouvements du véhicule dans cet axe étaient très bien maîtrisés et l'essai semblait se diriger vers une réussite. Or, à 50 mi/h (80 km/h), on a constaté un contact métal-métal au point bas de chaque cycle vertical. Un examen subséquent a révélé l'existence d'une interférence qui limitait le débattement normal de la suspension. De manière similaire, durant l'essai aux défauts transversaux destiné à vérifier la maîtrise des mouvements de lacet et de roulis, les oscillations semblaient bien atténuées au début, mais un roulis est apparu lorsque l'axe de rotation est passé légèrement au-dessous du sommet du rail, condition causée accidentellement par l'inclinaison donnée aux amortisseurs pour une meilleure stabilité des mouvements en lacet. Il en est résulté des mouvements latéraux importants à la traverse danseuse et on a constaté une interférence physique qui empêchait la suspension de jouir de la totalité de son débattement dans cet axe. Le contact latéral indésirable causait un impact sur la structure du bogie, entraînant de courtes mais fortes réductions de la charge verticale sur les roues. Il a été décidé d'interrompre ce type d'essai jusqu'à ce que soient éliminées les conditions d'interférence causées par la configuration du bogie.

Grâce aux mesures de la résistance au roulement, il est relativement simple de calculer les économies de carburant découlant de l'utilisation d'un bogie orientable par rapport à un bogie classique pour des conditions données de service. Le calcul a été effectué pour un train tractant des wagons de 286 000 lb (130 000 kg) selon deux scénarios très distincts, réalistes, correspondant à des profils de parcours de deux sociétés ferroviaires différentes. Le premier parcours, d'une longueur d'environ 420 mi (672 km), comportait beaucoup de courbes et de fortes dénivellations; l'autre avait environ 620 mi (992 km), était moyennement courbé, avec dénivellations moins fortes. Le premier scénario a donné une économie de carburant de 21,7 %, soit environ 3 129,5 gallons US (11 892 L) par wagon, par 100 000 mi (160 000 km). Pour le deuxième scénario, soit le parcours ayant moins de courbes, on avait réalisé une économie de 20,7 % ou d'approximativement 1 152,4 gallons US (4 379 L) par wagon, par 100 000 mi (160 000 km).

Pour un transporteur ferroviaire, la capacité de chargement accrue du wagon équipé de bogies Resco, peut facilement se traduire par un bénéfice plus élevé. Avec un tarif de 15 $ par 1 000 tn-mi (1 455 t-km), on aura, du fait de l'allégement du bogie, un bénéfice supérieur de 1 350 $ à 1 800 $ pour chaque 100 000 mi (160 000 km) parcourus. Donc une hausse qui n'aura coûté aucune augmentation de frais d'exploitation. De fait, comme nous avons pu le constater, les dépenses d'exploitation globales associées aux déplacements des wagons montés sur ces bogies sont moins importantes que pour les wagons sur bogies classiques. Par conséquent, un gain de productivité de 2 % produit une augmentation de 1 800 $ pour chaque 100 000 mi (160 000 km).

Des recherches ont démontré que dans le cas d'un convoi, une réduction de l'effort au crochet d'attelage pourrait entraîner une augmentation de productivité de 12 % sur les parcours en voie courbe et en pente forte, et de 4,85 % sur les parcours droits en terrain plat. Ce gain correspond à une augmentation de bénéfice entre 4 365 $ et 10 800 $ pour chaque 100 000 mi (160 000 km).

Même si mesurer l'usure des roues et des rails n'entre pas dans le cadre de cette recherche, il est utile de signaler que l'AAR possède des logiciels qui peuvent prédire les économies apportées par l'usure réduite due à une meilleure interaction de ce couple. Selon une estimation logicielle d'après les résultats des essais, ces économies seraient, pour une voie plutôt courbe, de 250 $ et 1 000 $ respectivement pour l'usure de la roue et du rail, et, sur voie plutôt droite, de 120 $ et 480 $. Ces valeurs sont fondées sur un parcours de 100 000 mi (160 000 km). Évidemment, les économies sur parcours de longueurs différentes seront calculées au prorata.

La valeur actualisée nette (VAN) du bogie se calcule aisément si le rendement du capital investi (RCI) est un critère à prendre en compte et si on connaît l'augmentation de bénéfice (ou la réduction des dépenses). En évaluant à 1 800 $ par année l'économie nette ou l'accroissement de bénéfice dû au bogie, après avoir tenu compte des variations des coûts d'entretien par rapport à l'utilisation d'un bogie classique et de l'amortissement sur 10 ans pour un RCI de 30 %, on obtient une VAN de 5 690 $. Cela signifie que pour des bogies coûtant 5 690 $ de plus que les bogies classiques, l'investissement requis serait récupéré après 10 ans, pour un RCI de 30 %. En supposant un objectif de RCI légèrement inférieur, on pourrait s'attendre à une VAN plus forte. À la fin de la période d'amortissement, le RCI augmenterait légèrement si les bogies restaient en service, hypothèse raisonnable, la durée de vie normale d'un bogie de wagon de marchandises dépassant les 30 ans. Compte tenu des données utilisées, l'estimation de bénéfice ou d'économie de coûts de 1 800 $ par année due au bogie Resco est très prudente pour les parcours d'exploitation utilisés. La différence de prix en sus pour le bogie Resco dépendra beaucoup du nombre de bogies vendus, des techniques de fabrication et d'autres facteurs. Vu le poids réduit du bogie, on estime que malgré l'usinage plus important que nécessite sa fabrication, l'objectif de différence de coût entre 5 000 $ à 6 000 $ est très réalisable, et ce, dans un délai raisonnable après son implantation.

Le présent sommaire ne serait pas complet sans une brève discussion de la flexibilité de l'étude conceptuelle des diverses techniques de fabrication. Les prototypes de bogies ont été construits à partir d'une structure hybride coulée. Pour les formes inusitées et/ou les zones critiques de concentration des efforts, on a coulé les structures, puis les pièces ainsi obtenues ont été assemblées par soudage en sections de géométrie plus simple. Les soudures ont été réalisées aux points soumis à des efforts relativement faibles. Grâce à cette technique, il a été possible de fabriquer les bogies très facilement, dans un très petit atelier de construction-fabrication mécanique à faible technologie. Ce mode de fabrication a démontré la faisabilité et l'attrait de ce type de concept, et a joué un rôle primordial dans le respect des objectifs de maintien des coûts de construction du prototype à l'intérieur de limites raisonnables. Il a également été démontré que la technique créée un environnement dans lequel une multitude d'installations de fabrication ont le potentiel de fournir des bogies utilisables à l'industrie ferroviaire nord-américaine. C'est un avantage qui peut améliorer de beaucoup la capacité de production de bogies à prix compétitif. D'autre part, si un client ou une spécification technique requièrent des longerons coulés monopièce, rien n'empêche de satisfaire à cette exigence, et cela n'entraînerait aucune surcharge.

Enfin, le sommaire de cette recherche ne pouvait que déboucher sur une discussion de l'orientation à lui donner. Le potentiel que possède ce bogie d'améliorer les opérations ferroviaires et la performance nette ne fait aucun doute. Les chercheurs ont décelé certains problèmes mineurs de conception qui devront être traités le plus tôt possible pour démontrer qu'ils ne risquent pas de compromettre les avantages de performance intéressants obtenus jusqu'à maintenant. Une fois les correctifs apportés, les prototypes devront être soumis à des essais d'endurance. Dans un premier temps, il est recommandé d'installer les bogies sur un wagon de marchandises intégré au train faisant des essais sur la boucle FAST du centre de tests ferroviaires de Pueblo et de contrôler l'usure et les dommages. Après, si toutes les indications sont positives, quelques bogies devraient être affectés à un service ferroviaire opérationnel, de préférence sur voie comportant des courbes prononcées, et être surveillés de manière similaire.

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1. Background

The Resco Engineering Steered Frame Freight Car Truck is a concept that provides direct steering to the axles of a freight car in a package that is at the same time simple, inexpensive, and lightweight. The tests described herein were conducted on the first prototype pair of such trucks. Previous testing was performed on the BC Rail tracks near Lillooet, British Columbia, under a wood chip car and was reported in Transport Canada Report TP 13334E [1]. For that application the suspension was designed for a car with a maximum mass of 263,000 lb. (119,545 kg).

The series of tests reported on here was conducted primarily at the Association of American Railroad's (AAR) test facility in Pueblo, Colorado. The tests were conducted under the auspices of a wider program of truck testing instigated by the AAR to identify potential improved performance trucks for bulk commodity service in North America. This program is known as Strategic Research Initiative #2 or The Advanced Freight Car Truck Program of the AAR and was funded entirely by the AAR. The data gathered by the AAR in this test program has been reported at the AAR Annual Research Review.

Some preliminary testing was also carried out at National Research Council Canada's (NRC) test facility in Ottawa.

The total car mass for a typical heavy haul rail application in North America is currently 263,000 lb. (119,545 kg) for a nominal 100 ton (91 tonne) car. For many years the AAR standard has been that 263,000 lb. (119,545 kg) is the maximum load that eight 36 in. wheels can support. There is a movement afoot to increase the loaded mass to 286,000 lb. (130,000 kg) on 36 in. wheels – a nominally 110 ton (100 tonne) car – and then to 315,000 lb. (143,182 kg) on 38 in. wheels, nominally a 125 ton (114 tonne) car. To qualify for this series of tests, trucks could support either a 286,000 lb. (130,000 kg) car or a 315,000 lb. (143,182 kg) car. As the Resco prototype trucks were designed for 36 in. wheels, it was decided to design the suspension for 286,000 lb. (130,000 kg) total mass. Wheels of 38 in. diameter would have been required for the 315,000 lb. (143,182 kg) car, necessitating substantial changes to the prototype truck structures.

In order to accommodate the 286,000 lb. (130,000 kg) requirement, and also to rectify some design deficiencies that had been discovered in the earlier testing, several design changes were incorporated in the trucks compared with those tested under the wood chip cars at BC Rail. These changes are described in section 1.3.

In addition to the increased loaded weight of the aluminum coal car used in these tests, over the maximum for the wood chip car used in the earlier tests, the empty weight was also approximately 16,000 lb. (7,273 kg) less, because of its aluminum construction, providing for 39,000 lb. (17,727 kg) more load capacity than a typical steel 100 ton car. This meant that the suspension system had to support a carbody with a mass of 25,000 lb. (11,364 kg) empty, and 269,000 lb. (122,273 kg) loaded, a ratio of 10.76:1. By comparison, for the 100 ton (91 tonne) wood chip car (or a conventional 100 ton (91 tonne) steel coal hopper), the suspension supports approximately 44,000 lb. (20,000 kg) empty and 246,000 lb. (111,818 kg) loaded, a ratio of 5.6:1. The challenges for the suspension design in the case of the aluminum coal car are clearly more difficult than for the steel wood chip car.

For the testing at Pueblo most of the data was gathered by staff from the Transportation Technology Centre Inc. (TTCI), a wholly-owned subsidiary of the AAR, which also provided, installed and operated the instrumentation and recording equipment. For the testing at NRC, data was gathered by Resco Engineering staff.

Substantial funding to support Resco's participation in the tests at NRC and Pueblo was provided by Transport Canada, through the federal Program for Energy Research and Development (PERD).

1.1 Test Description

Prior to the testing at Pueblo a rollout test was conducted at NRC's test facilities in Ottawa, Ontario. The purpose of this test was merely to confirm that the design modifications made following the tests at BC Rail had overcome the problems of excessive deflection in a rubber shear pad, which had brought those tests to a slightly premature end. The modified truck (one only) was installed under a loaded freight car and operated at low speed through some curves in the yard track. The operation of the steering mechanism was observed to see that it moved correctly, angles of attack were measured to confirm the steering function, and the deflections in the rubber components were observed to see that they were no longer excessive.

Partway through the truck testing at Pueblo, some component (damper) testing was also performed at the laboratories of the Mechanical Engineering Department, Queen's University, Kingston, Ontario.

At Pueblo a number of tests were performed on the trucks to examine various aspects of their performance and to compare them with those of a standard three-piece freight truck and a typical premium freight car truck, which were also tested as part of the program. Figure 1 shows the consist of locomotive, instrument car, buffer car and test car with test trucks, ready for the start of testing.

Figure 1. The Test Consist at Pueblo

The categories of performance examined were truck mass, dynamic stability, rolling resistance, curving behaviour, dynamic curving, perturbed tangent track, and wheel load equalization.

1.1.1 Truck Mass

The mass of the truck was determined by weighing the loaded car. The car was taken to a weigh-scale and the load at each axle was determined. The same exercise had been performed shortly before on the same loaded car with standard three-piece trucks under it, so by subtracting one total from the other, it was possible to determine the difference in mass compared to a pair of standard trucks. The absolute mass of the truck was not determined but the change in mass is what is most important because this determines how much more load can be carried in the car without exceeding the axle load limits. Increasing the payload of a car without increasing axle load brings substantial economic benefit to the railways. It improves profitability by moving more payload without increasing operating costs, and it increases the overall capacity of the existing infrastructure.

1.1.2 Dynamic Stability

The purpose of this aspect of the testing was to show that the trucks maintain stable dynamic behaviour throughout the operating speed range for the car, in both empty and loaded conditions.

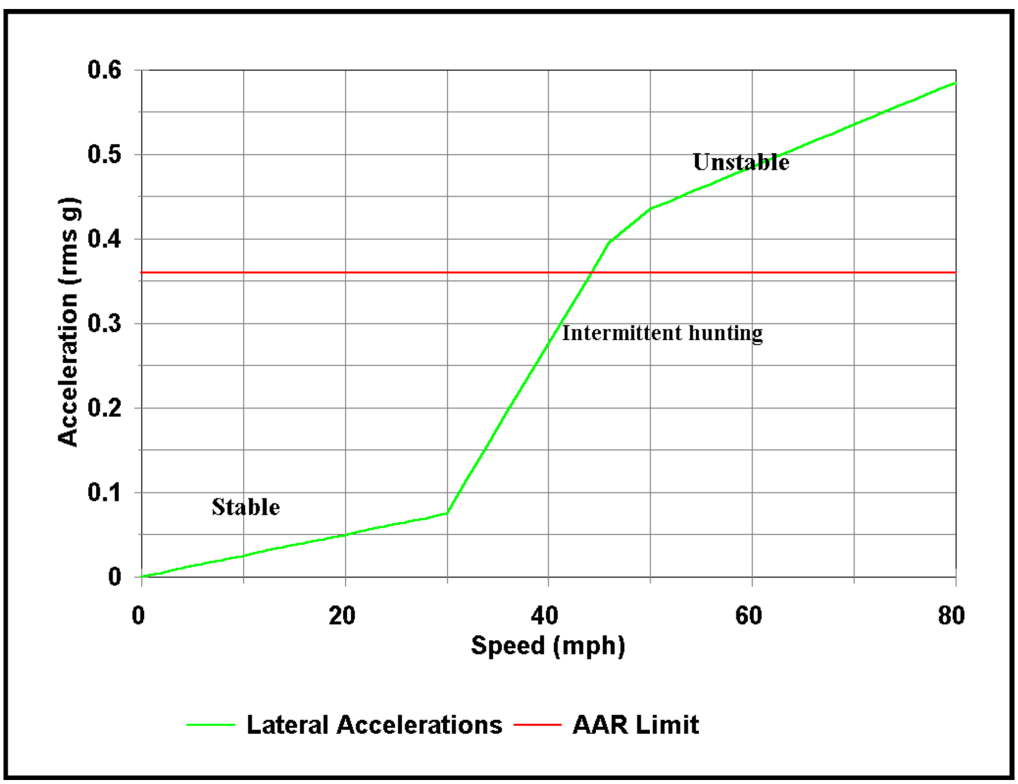

Figure 2. Typical Characteristic of Unstable Car

In fully unstable behaviour (also known as truck hunting), the wheelsets within a truck will oscillate from side to side in regular periodic fashion, the wheel flanges contacting the rails on each side alternately. At the onset of this condition the behaviour may be observed to occur for a few cycles and then stop, only to begin again some time later. As speed is increased the periods of oscillation increase until the oscillation is non-stop, known as fully established hunting. Carbody lateral accelerations increase as a result of this behaviour and so the hunting condition can often be identified by measuring carbody lateral accelerations and observing the speed at which these begin to increase rapidly. For a stable car the lateral accelerations are found to increase steadily with increasing speed, simply due to the increased energy input into the suspension system, by track irregularities, as speed increases. This is shown in Figure 2.

The initial part of the graph, from 0 to 30 mph (0 to 48 km/h) shows the steady increase simply due to track irregularities acting on the car at increasing speed. At 30 mph (48 km/h) the slope of the graph changes abruptly. Hunting has begun at this point but the conicity is only high enough to create it in a few sections of track. As the speed increases from 30 mph (48 km/h) to 45 mph (72 km/h), the proportion of track creating hunting behaviour increases until, at 45 mph (72 km/h), the car is hunting on the entire track. Beyond this point the acceleration levels start to “plateau” because the motions are limited by the flange clearance. They will typically increase steadily, however, because the frequency of oscillations increases with speed and thus the accelerations do also.

There can be other reasons why the carbody lateral accelerations might increase rather in the manner shown in Figure 2, so it is essential, if truck hunting is to be identified as the cause, to observe the wheelset motions to determine whether large periodic lateral oscillations are occurring. Genuine hunting always results in large, periodic, lateral wheelset excursions. Among the other non-hunting sources for large lateral carbody accelerations, and a sudden change in their rate of increase, is the effect of hitting lateral bump stops once the lateral motions have grown to a certain point.

Tests were conducted on specific areas of the test tracks at speeds that were increased from 40 mph (64 km/h) in steps of 5 mph (8 km/h) to the point at which lateral accelerations increased above the predetermined maximum allowable, or until a safety criterion (such as minimum wheel load) was met, or until the predetermined maximum speed for the test was achieved. Wheelset lateral motions were also observed visually from the instrumentation car to confirm whether any untoward carbody accelerations were the result of truck hunting or from some other source. For the bulk commodity test program the target speed specified was 70 mph (112 km/h). For the tests of its trucks, Resco Engineering requested a target speed of 105 mph (168 km/h) in order to attempt to identify where the onset of hunting occurred, because it was anticipated to be significantly higher than 70 mph (112 km/h).

The bulk commodity test program also only required high-speed stability testing to be done under empty conditions because conventional wisdom is that this is the least stable condition for a rail car. Resco requested that its trucks be tested for high-speed stability both empty and loaded because of the potential that a fully steered truck might have different characteristics in this regard than a conventional non-steered truck.

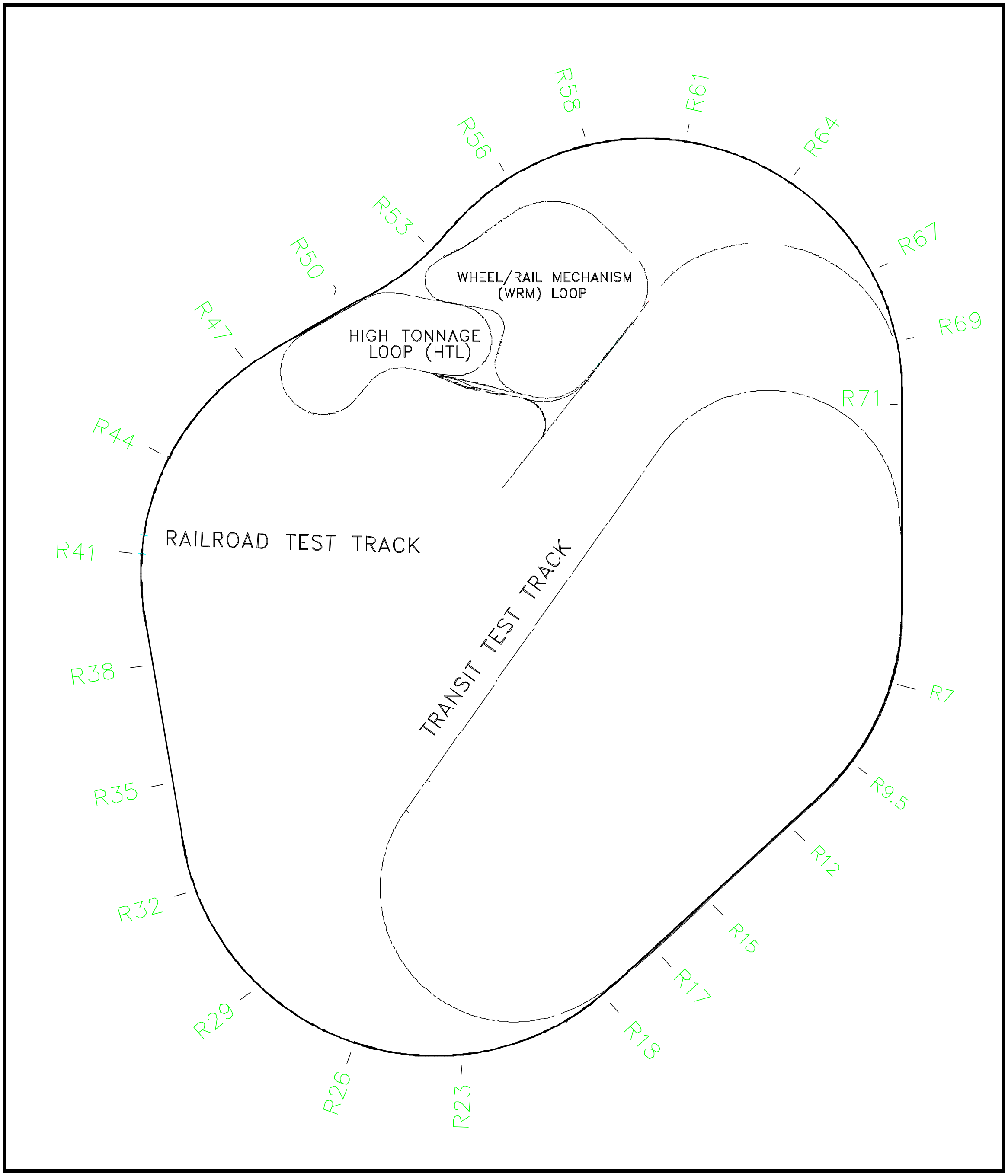

Figure 3. Railroad Test Track (RTT) and Other Test Loops at AAR Facility, Pueblo, Colorado

An adequate level of stability in a freight car is necessary if current operational practices are to be maintained or improved. A reduction in operating speeds to accommodate poor high-speed performance of a truck that has otherwise attractive characteristics would reduce the capacity of the railway and thus bring large economic disadvantage to offset any advantages it may have. By the same token, increased stability in its rail vehicles has the potential for economic benefit to the railway by increasing the throughput capacity of its lines (more trains per day over the same lines). This would depend, of course, upon other operational considerations allowing the increased speed.

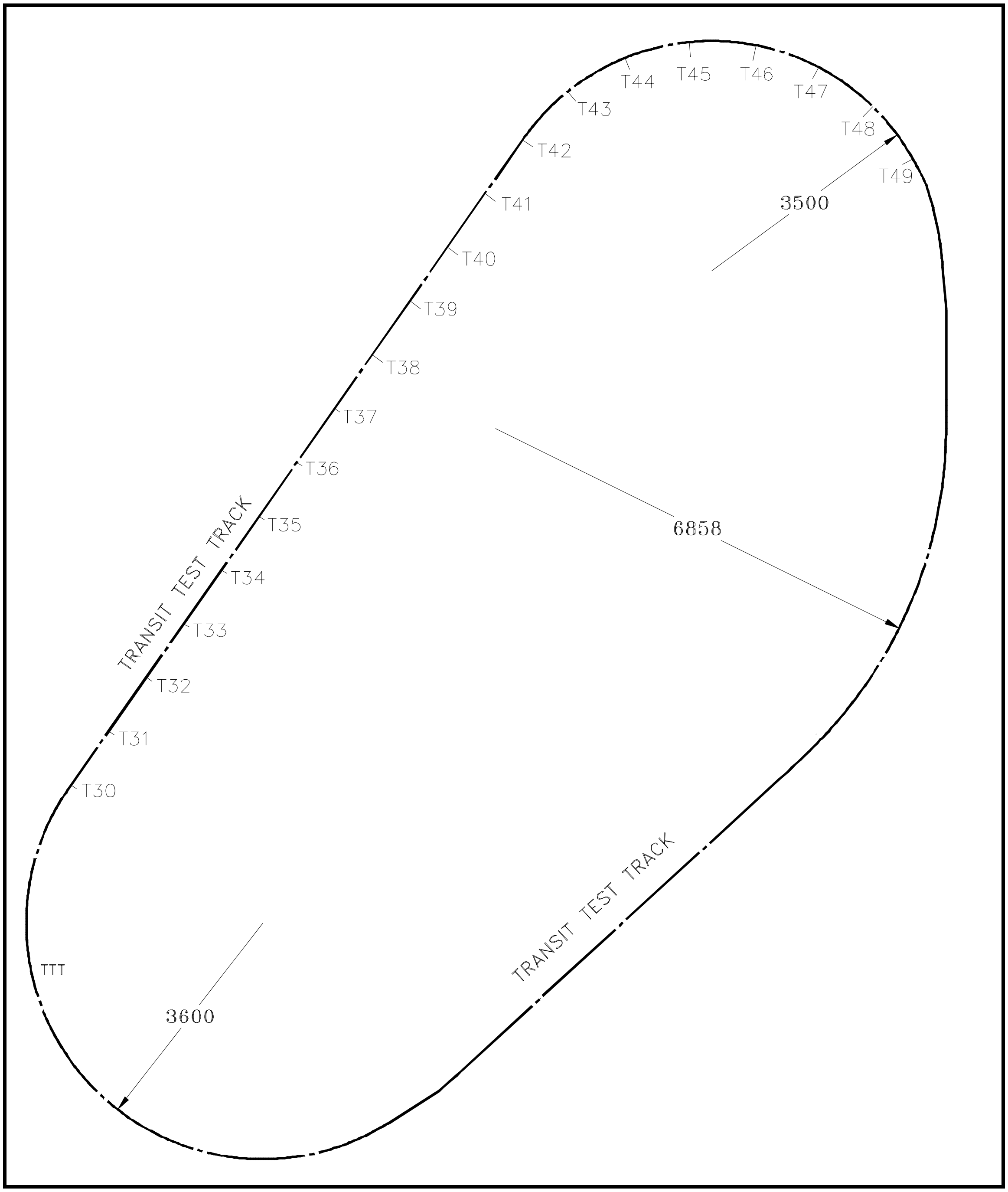

Figure 4. Transit Test Track (TTT)

Two sections of track at the test centre are used for the establishment of high-speed dynamic behaviour: the Transit Test Track (TTT) and the Railroad Test Track (RTT). These sections are shown in Figure 3. The TTT is limited to speeds up to 80 mph (128 km/h) but the RTT has been recently upgraded to allow speeds in excess of 125 mph (200 km/h).

On the RTT the dynamic test section is usually in the tangent section between mileposts R41 and R32, a distance of 9,000 ft. (2,744 m). On the TTT, shown in more detail in Figure 4, the dynamic test section is usually somewhere between T43 and T30. This is a distance of 13,000 ft. (3,963 m). The whole loop has three major curves, 3,500 ft. (1,067 m) radius, 3,600 ft. (1,098 m) radius and 6,858 ft. (2,091 m) radius, and three tangent sections. The test section is the longest tangent section.

For high-speed (stability) testing of freight cars, specially identified wheelsets are used, which are known to produce the desired level of conicity on the specific test track sections. This conicity has been chosen so as to normally cause instability in a standard freight car truck at a speed below 70 mph (112 km/h) in the empty condition.

1.1.3 Rolling Resistance

The rates of fuel consumption, and the rates of wheel and rail wear, are contributed to significantly by the action of the wheel against the rail. This is particularly true of the action of the wheel flange against the side of the rail, but there is also a contribution from the rolling of the wheel tread against the rail head. Measurement of the effort required to haul a car of known weight on level track at low speed provides an indication of the performance of the truck in this regard. Because perfectly level track is not available at the test centre in all of the required tangent and curve conditions required for the test, the resistance is determined by having the car travel in both directions on each test section and the average is taken. This averaging removes the effect of the grade of the track from the result and partially removes the effect of wind resistance. By operating the car at low speed and only in low wind conditions, the effect of the wind resistance is also largely eliminated, but not altogether. The wind resistance of the car is not equal when operated in opposite directions, even when the wind speed and the train speed do not vary. Firstly, of course, even if there is no change in ambient wind speed throughout the test, its effect on the car is to be added to the train speed for operations in one direction and subtracted from it in the other direction. Averaging leaves in the wind resistance due to train speed. The wind resistance at train speed should therefore be subtracted from the total resistance obtained in this measurement in order to derive the actual net rolling resistance. Even this, alas, does not quite compensate for the differences occurring in the two directions, and neither does it completely compensate for the effect of ambient wind direction on the results. Because there is a railcar connected to the test unit on one side but not on the other, there is a distinct difference in the wind resistance acting on the test car, depending upon whether the effective wind is blowing from the test car end of the train towards the locomotive, or from the locomotive end of the train towards the test car. In one direction the effective wind will produce a greater resistance than in the other, due to the sheltering effect of the car next to the test car. These effects are ignored in the test data processing, and the average from the two directions is taken as providing the rolling resistance value. It is anticipated that the errors in wind compensation are small but they will, of course, be larger relative to the force measurement when those measurements are themselves small. In other words, the results from tangent track measurements are more prone to errors from this source than those on steeply curved track, where the total forces measured are usually much larger and thus the errors are, relatively, smaller.

1.1.4 Curving Behaviour

Besides the rolling resistance in curves, mentioned above, other significant factors in assessing the performance of a truck in curves are the forces exerted by the truck against the rail. These forces, and the ratios between them, significantly affect the tendency of the wheels to climb the rails (wheel climb derailment) and the damage to the track (in alignment, in ballast damage and in actual rail fractures) produced by the car. Tests were conducted on the Wheel/Rail Mechanism (WRM) loop at the test centre to measure these forces. A drawing of the loop can be seen in Figure 5.

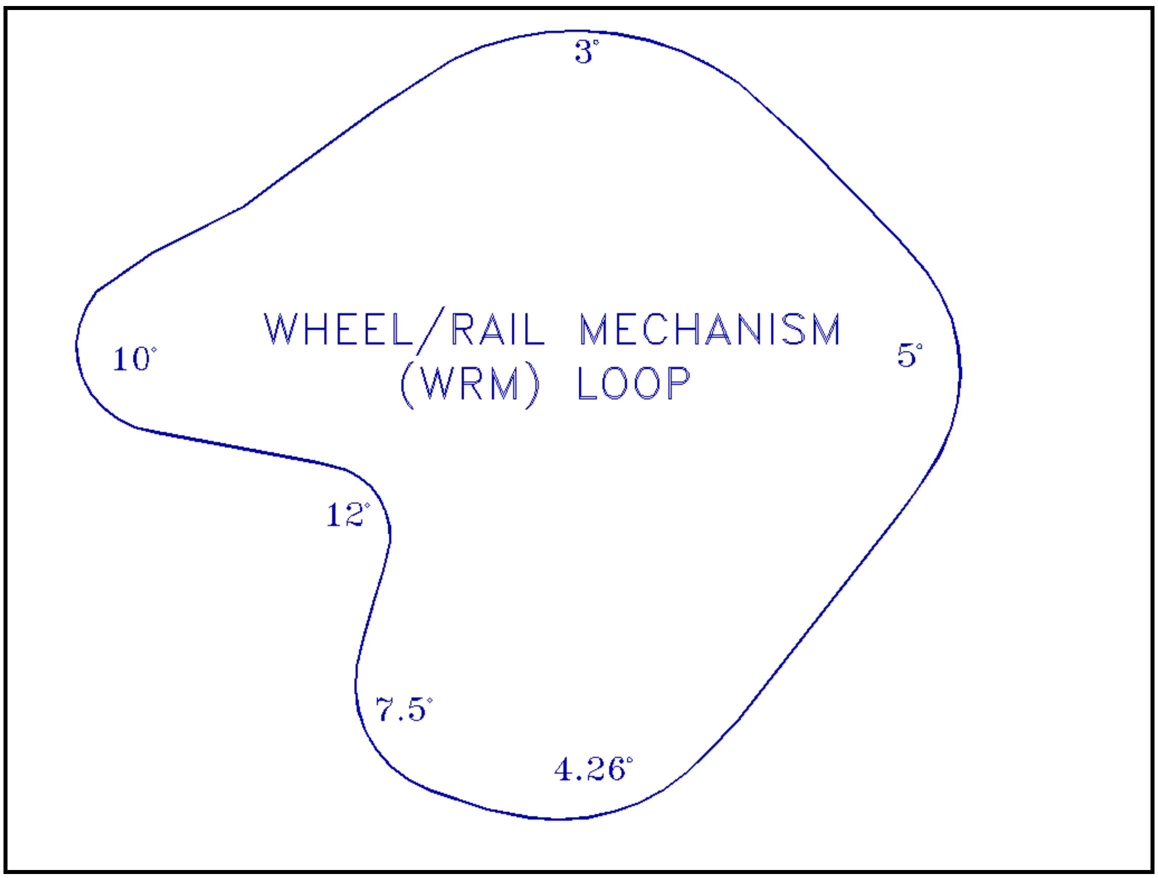

Figure 5. Layout of WRM Loop

The lateral wheel/rail forces produced by a truck in a curve are expected to be dominated by the lateral creep due to wheelset misalignment. They may also be affected by the quasi-static lateral forces between the car and the track due to the centrifugal effects of the speed of the car on the curve, especially when the creep forces are small.

Four curves in the WRM loop were utilized for this test, a 4.26º, a 7.5º, a 10º and a 12º. For each of these curves the "balanced" speed is 24 mph (38.4 km/h). [Note: The "balanced" speed is the speed at which the superelevation produces a lateral component of the acceleration due to gravity, across the rail heads, exactly balancing the centripetal acceleration.]

The test speeds selected were 12 mph (19.2 km/h), 24 mph (38.4 km/h) and 32 mph (51.2 km/h) for the underbalanced, balanced and overbalanced conditions, respectively, except for the 12º where the underbalanced speed was 15 mph (24 km/h).

Table 1 shows the degree of overbalance or underbalance for each of these curves at the respective operating speeds.

Curve (º)	Speed (mph)	Acceleration (g)	Overbalance (g)	Superelevation Deficiency (in.)
4.26	12	0.007	-0.02	-1.29
	24	0.027	0	0
	32	0.048	0.023	1.34
7.5	12	0.012	-.038	-2.25
	24	0.050	0	0
	32	0.089	-.039	2.34
10	12	0.017	0.050	-3
	24	0.067	0	0
	32	0.119	0.052	3.12
12	15	0.031	-.049	-2.94
	24	0.080	0	0
	32	0.143	0.063	3.78

Table 1. Curving Test Conditions

1.1.5 Dynamic Curving

In addition to the curving behaviour tests described above, there is an additional curving test, on a 10º curve, in which lateral and vertical alignment perturbations have been intentionally installed. The vertical perturbation has dips at each of the joints in the rail, 0.5 in. (12.7 mm) below the high point of the rail, the joints spaced 39 ft. (11.89 m) apart, and those on the low rail exactly halfway between those on the high rail. The lateral perturbation consists of cusp-shaped gauge widening of the outer rail 1 in. (25.4 mm) above standard gauge, the widest part coinciding with the lowest point at the dipped joints. This section of track is parallel to the 10º curve in the main WRM loop and is accessed through a switch from the main loop (not shown in Figure 5). This section tests the ability of the suspension and truck to accommodate poor alignment in curved track. The test vehicle is operated through this section at speeds between 10 and 32 mph

(16 and 51.2 km/h), in 2 mph (3.2 km/h) increments, and the track forces are measured and compared with a pre-determined criterion. The perturbation is illustrated in Figure 6.

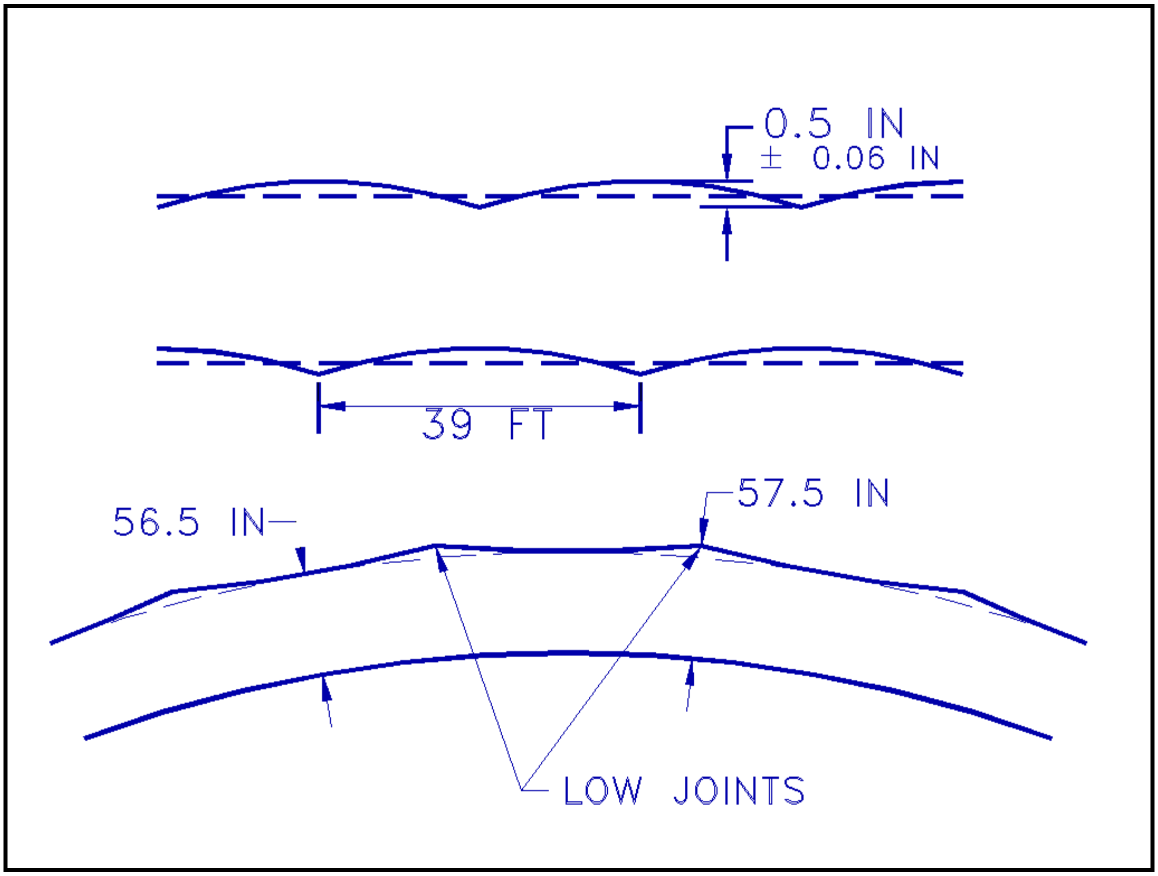

Figure 6. Dynamic Curve Lateral and Vertical Perturbation

1.1.6 Perturbed Tangent Track

There is also a tangent section of track at the test centre that has been installed with deliberate and well-known vertical and lateral perturbations in it. These sections are known as the Pitch and Bounce section, the Twist and Roll section, and the Yaw and Sway section. In the Pitch and Bounce section there are ¾ in. (19 mm) dipped joints (vertical) in both rails, placed opposite one another at 39 ft. (11.89 m) intervals, as illustrated in Figure 7.

In the Twist and Roll section the dipped joints are the same but they are placed exactly out of phase with one another on the opposite rails (i.e. the dips on one side are exactly halfway between the dips on the other side). This is illustrated in Figure 8.

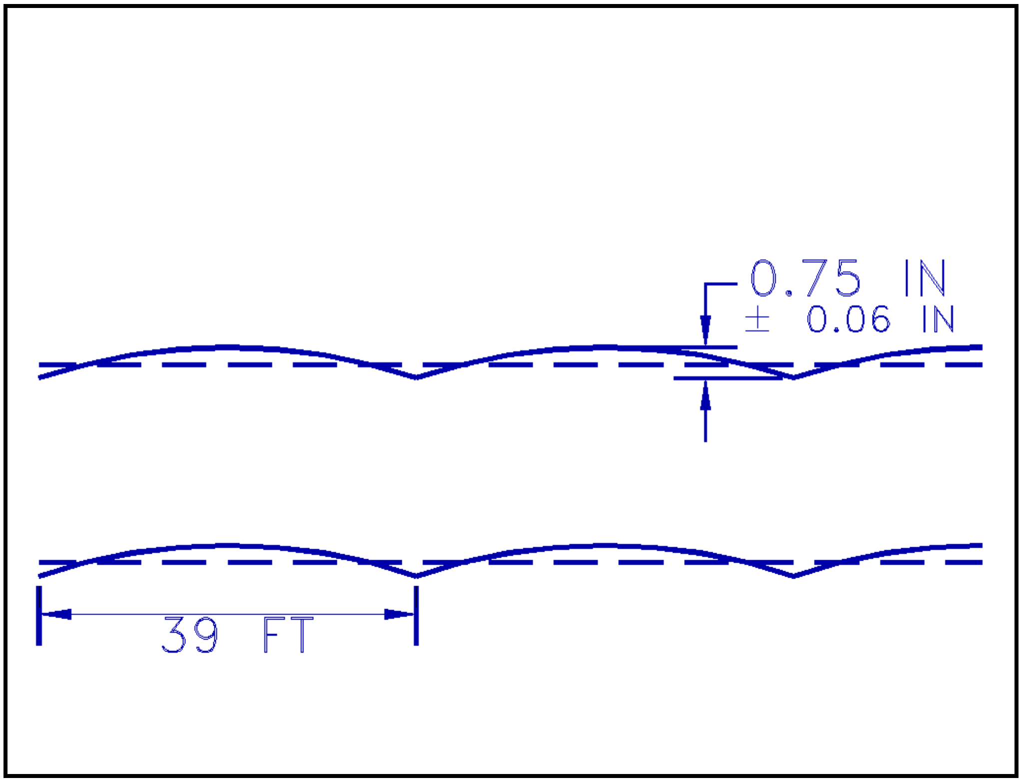

Figure 7. Pitch and Bounce Vertical Perturbation

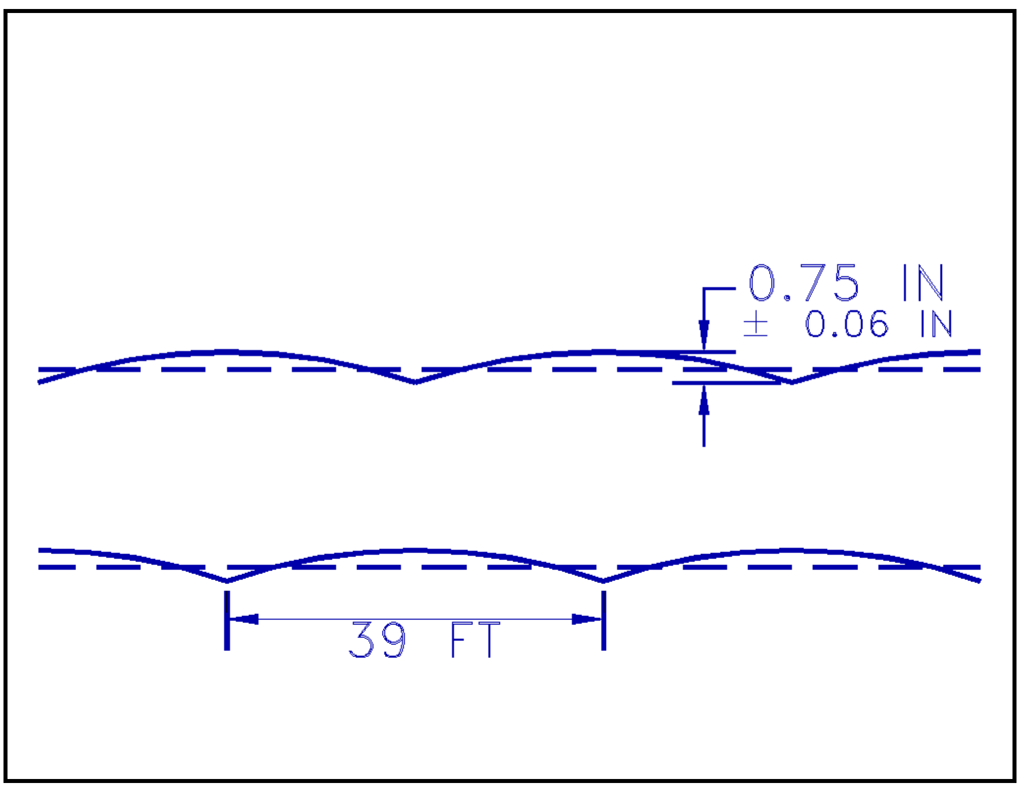

Figure 8. Twist and Roll Vertical Perturbation

In the Yaw and Sway section the track perturbations are lateral. The two rails are disturbed from a tangent alignment by a sinusoidal lateral displacement of 1.5 in. (38.1 mm) total amplitude and a wavelength of 39 ft. (11.89 m) as shown in Figure 9.

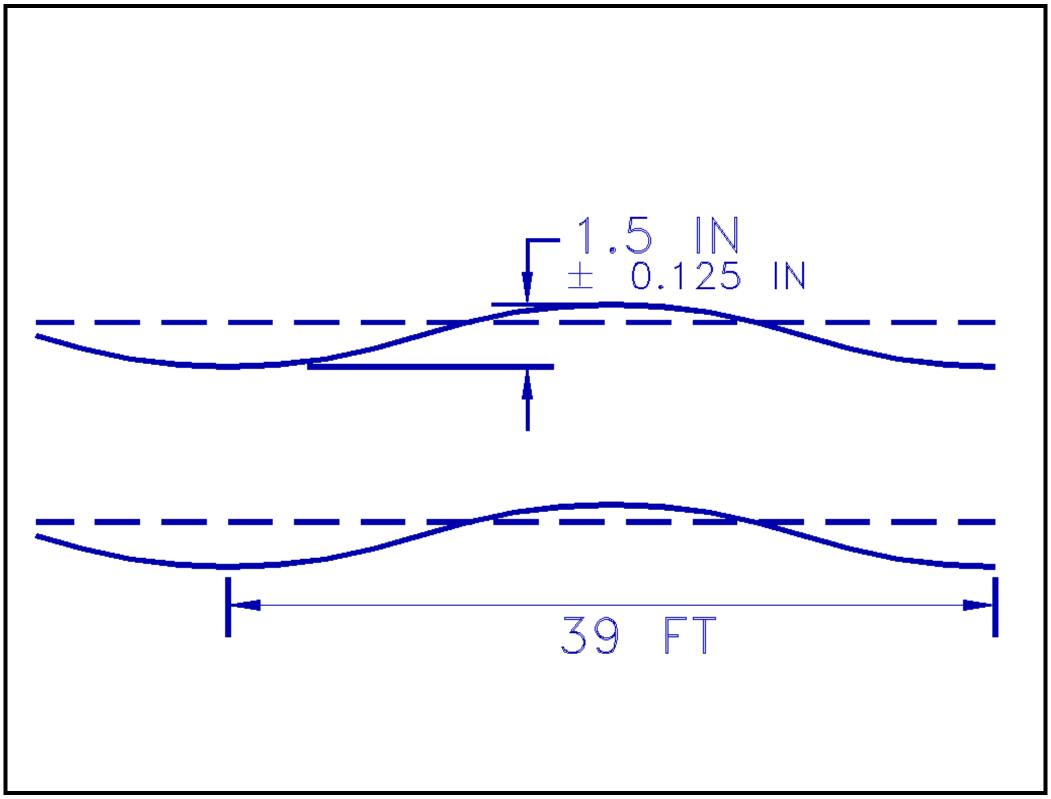

Figure 9. Yaw and Sway Lateral Perturbation

The test car is operated through the test sections at speeds starting at 12 mph (19.2 km/h) and increasing in 2 mph (3.2 km/h) increments up to 50 mph (80 km/h), and in 5 mph (8 km/h) increments thereafter up to 70 mph (112 km/h), or until the wheel/rail forces or body motions exceed pre-determined criteria.

1.1.7 Wheel Load Equalization

An important criterion, which provides an indication of the ability of the truck to accommodate twisted track, is to measure the wheel load equalization when one wheel in one truck is lifted out of the plane of the other three through known amounts. For this test the wheel was lifted to 3 in. (76.2 mm) out of the plane in increments of ½ in. (12.7 mm). In the case of the Resco truck the loads were determined from the instrumented wheelsets. One wheel was jacked and then rested on a shim block of known height. For the standard truck test a slightly different technique was used. Four bar-type load cells were utilized, one under each wheel, in the static condition. One wheel was jacked the required amount and the loads in the three undisturbed wheels were measured. The load at the jacked wheel was deduced from the known total load at the wheels in the level condition minus the three measured loads.

1.2 Comparison of the RESCO Steered Frame Truck, a Standard Three-Piece Truck and a Premium Truck

The truck that was the subject of these tests is a significant departure from a conventional freight car truck. Apart from the fact that the Resco truck contains a steering mechanism that creates an angle between the two axles that exactly matches the change in orientation of the track over the axle centre distance, the truck also has no bolster. The structural frame of a conventional three-piece truck consists of two sideframes and a transverse bolster. The bolster is connected to the sideframes at their centres through springs and friction wedges. The centre sill of the carbody has a centre plate that rests on a centre bowl in the middle of the bolster, to transfer the weight of the car to the truck. The bolster slides up and down in an opening in the sideframes, supported by the springs and wedges. It is the connection of the bolster to each sideframe, primarily through the friction wedges, that maintains the truck frame "square" and controls the orientation of the wheelsets to each other and to the track. As there is very little ability of the wheelsets to move longitudinally relative to the sideframes, their ends must remain a fixed distance apart longitudinally, and so they must remain essentially parallel to each other.

The main deficiencies of the three-piece truck design, relative to the demands now placed upon it, come about principally because of the reliance on the friction wedges to provide the essential performance characteristics. When the forces acting on the truck in service are sufficient to overcome the friction at the wedges, the truck goes out of "square". This can happen in an unstable oscillatory manner, on tangent track or mild curves, in which case it is known as "hunting". Hunting most often occurs at higher speeds and low axle loads (e.g. empty car). The other manner in which the "out of square" configuration causes deficient behaviour of the truck is at high axle loads in curves. The curving forces can overcome the friction at the wedges and hold the truck in a steady-state deformed shape, known as "lozenging" or "warping". This creates very rapid wear of wheels and rails, promotes wheel climb and/or rail rollover derailments, and increases the rolling resistance of the truck. The propensity for the truck to exhibit these deficiencies increases with increasing truck wear. [It should be noted that, while the hunting condition was observed during the standard three-piece truck tests reported here, there was never an instance of warping during the curving tests. This means that the results for the standard truck are those for a well-behaved truck, not the worst case.]

The premium trucks are all based upon the standard three-piece truck design, with added features aimed at improving the performance. Two principal features are added into premium trucks, differentiating them from standard three-piece trucks. First, the warp resistance is usually increased by some means. This can either be simply a matter of changing the wedge size, the wedge spring force, and/or the wedge angle or the wedge configuration, or a device can be added to the truck so that the warp stiffness is increased independently of the wedges. Second, provision is often made at the connection between the wheelsets and the sideframes to allow a small amount of longitudinal motion so that the wheelsets have the ability to self-steer on some curves. Provision of that freedom can negatively affect the hunting stability of the truck so it is important to reach a suitable compromise in order to achieve some increase in curving performance without allowing too much of a decrease in high-speed stability.

The Resco truck contains two sideframes connected to each other by a shear frame. The shear frame is configured so as to maintain the sideframes "square" with each other under very large shear loads but is torsionally flexible so as to allow the sideframe ends the ability to move independently in the vertical direction. This feature enables the truck to traverse uneven track with very little change in load at the wheels. There is no bolster in the Resco truck. Two pairs of load springs are carried on projections on the inside and outside of each sideframe, one pair of springs on each carrier. The weight of the car is supported directly on the springs, through sliders acting on horizontal wear plates attached to the underside of the carbody. None of the car's weight is carried through the centreplate. At the top of each pair of springs a single casting spans the two and this supports the sliders. During the testing in this program it was found to be desirable to replace the outer sliders with low friction rollers; the inner sliders were retained. The castings at the tops of the inner pairs of springs are attached to a transverse traction beam. The traction beam has a hole at its centre into which the carbody centreplate fits. The traction beam is also attached to both sideframes through drag links at its ends; these allow the desired vertical and lateral motions but restrain the beam longitudinally. This arrangement maintains the truck in position under the car.

The second substantial way in which the Resco truck differs from other North American freight car trucks is that it contains a steering mechanism. As configured for the tests reported here, the outboard axle is fixed to the structure consisting of the two sideframes and the interconnecting shear frame. The inboard axle is connected to the truck structure through a linkage that is connected to the carbody. The geometry of the linkage is arranged so that the rotation of the carbody relative to the truck frame, as the vehicle enters a curve, causes rotation of the inboard axle relative to both the truck frame and the outboard axle (which are fixed together). The amount of axle rotation is exactly sufficient to provide alignment on the curvature indicated by the rotation of the carbody. At the same time as the inboard axle is rotated, the sideframe is shifted laterally at that point, causing the outboard axle to also rotate through the desired steering angle. This steered position is illustrated in Figure 10.

Whereas other premium trucks seek to improve the trade-off of curving and dynamic stability available through the conventional truck arrangement, the Resco truck is intended to simultaneously achieve maximum stability and virtually eliminate the wear and power absorption of conventional trucks through curving action.

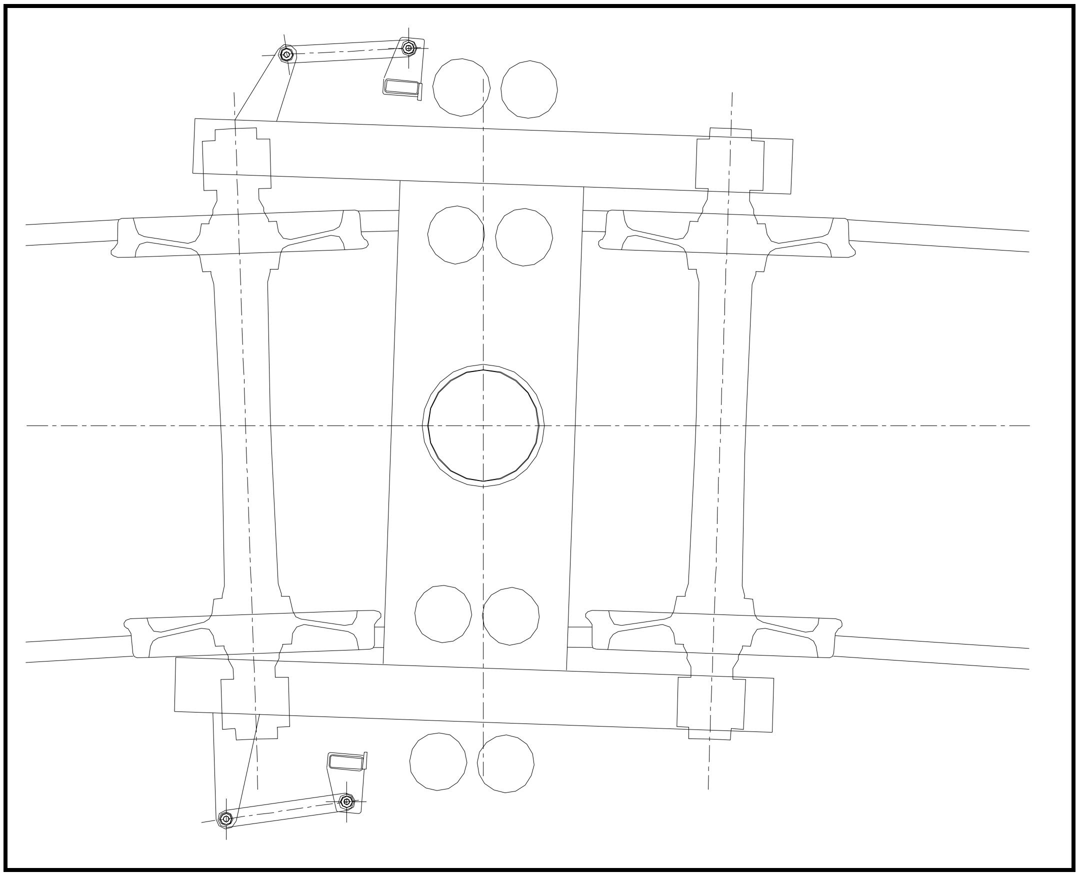

Figure 10. The Resco Truck in Steered Configuration

1.3 Design Modifications

The previous testing at Lillooet, British Columbia [1] identified a number of areas where design changes were necessary. The following changes were made prior to introducing the Resco truck into the bulk commodity program tests:

- Modified the spring design to accommodate the change in carbody mass, empty and loaded.
- Modified the steering geometry to accommodate the different truck centre distance between the wood chip car and the aluminum coal car.
- Modified the carbody adapter to accommodate the different carbody bolster design for the coal car.
- Modified the clearance between the traction beam castings and the shear frame (previously found to be inadequate).
- Modified the retention of the rubber pads at the steered end to prevent the excessive deflections that occurred in previous testing.

- Modified the attachment of the shear frame to the sideframes to eliminate cracking of the retention wedges used previously.

The truck remains a fully steered design, able to utilize standard AAR wheelsets and brake beams and, with the addition of adapters, able to be mounted under a car suited for standard three-piece freight trucks. Figure 11 shows the truck installed under the test car.

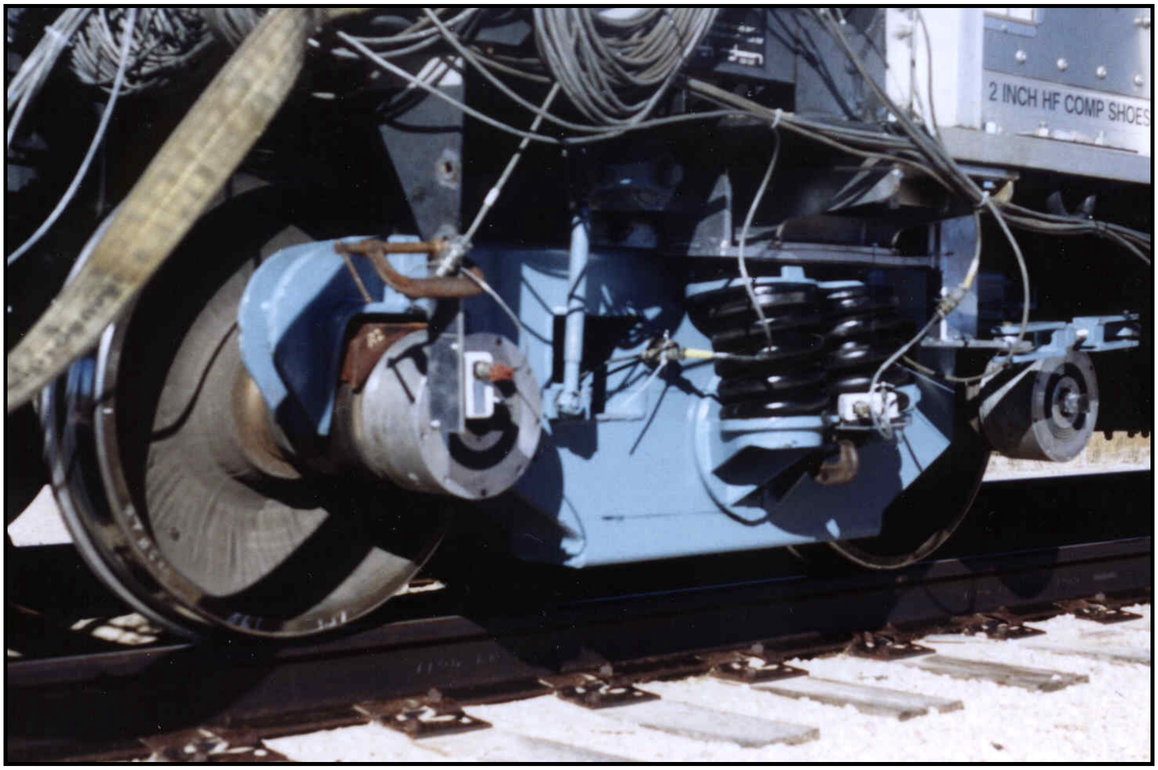

Figure 11. Instrumented Truck Ready for Test

1.4 Test Locations

Dynamic stability tests were conducted at two locations at the Pueblo test facility: the Transit Test Track (TTT), where nearly all of the bulk commodity program high-speed testing has been carried out and which is limited to 80 mph (128 km/h), and the Railroad Test Track (RTT), which is used for speeds higher than 80 mph (128 km/h).

It was decided to use the TTT for speeds below 80 mph (128 km/h) and the RTT for speeds above that, rather than using the RTT for all speeds, so that the comparisons with other trucks, which had only been tested on the TTT, could be made fairly.

Tangent track rolling resistance was measured on a smooth section of the Precision Test Track (PTT). Curved track rolling resistance was determined from low-speed tests on the WRM loop.

Response to track irregularities was determined on the PTT in areas where known irregularities had been created.

Curving behaviour was measured at underbalance, balance and overbalance speeds on the WRM loop, as mentioned previously.

2. Test Results

2.1 Initial Rollout

In the initial rollout tests, conducted at the NRC laboratories in Ottawa, one modified Resco steered frame truck was installed under one end of a loaded hopper car. A standard three-piece truck was mounted under the other end.

The car was moved into various curves in the NRC yard system, including a particularly difficult transition and switch at the entrance to the yard from the main line, which has been the cause of several derailments of freight cars entering or leaving the facility.

Visual observation of the truck showed that all motions of the steering mechanism were acting as expected and that there was no excessive deflection of the rubber shear pad at the steered end, as had occurred in the previous testing. Smooth steering operation was apparent.

The car was then taken back into the various curves in the NRC system and stopped at locations in the beginning and the body of the curves. The angles of attack were measured, with results as shown in Table 2.

	RH Spiral	RH Curve	LH Curve
Steered Leading	-1	-0.28	+0.19
Steered Trailing	-1.25	+0.19	+0.25
Unsteered Leading	+2.56	+2.13	+0.69
Unsteered Trailing	+1.13	+1.13	+0.44

Table 2. Angles of Attack – NRC

It can be seen from this table that the steered truck performed much better than the unsteered one. It is also clear that the unsteered truck has gone into the condition known as “warp collapse” (also known as “lozenging”) in the spiral leading into the RH curve and that it did not recover from this once fully into the curve. The evidence for this is that both axles have a very large, positive angle of attack. When a three-piece truck is behaving properly there will be a large, positive angle of attack at the leading axle and the trailing axle will either be nearly perfectly aligned or will have a smaller negative angle of attack, depending upon how sharp the curve is.

In this move the unsteered truck was leading. On the spiral the steered truck is showing a strong bias towards the low rail because, being the trailing truck in this instance, it is prematurely steered by the carbody motions. Once fully into the curve it recovers and shows a small, negative angle of attack at the leading axle and a similar positive one at the trailing axle. This indicates a slightly oversteered condition, probably brought about by the fact that the length of the test car used in this instance is somewhat greater than that for which the truck was designed.

When the direction of rolling was reversed, and the curve became a left-hand curve rather than right-hand, it can be seen that the unsteered truck fared rather better, as a trailing truck, but still produced large angles of attack. The steered truck, now the leading one, produced substantially better results than the unsteered in this direction as well as the other.

When operated through the switch at the entrance to the yard the steered truck operated smoothly and without tendency to climb. It was judged to be far more capable of negotiating the poor track than a standard truck.

The purpose of this testing was simply to provide assurance that the previously identified design issues had been addressed, and it was felt that this had been shown to be the case. The truck was now deemed to be ready for testing at Pueblo.

2.2 Truck Mass

The first move of the loaded test car at Pueblo, after the trucks had been installed under a loaded aluminum coal car, was to a weigh-scale where the individual axle loads were recorded. The results of this weighing, and the weighing of the same car previously with standard three-piece trucks under it, are shown in Table 3.

	Standard (lb.)	Resco (lb.)
Axle 1		70,127
Axle 2		71,163
Axle 3		71,141
Axle 4		70,420
Total	286,509	282,939

Table 3. Axle Load Comparison

It can be seen that the Resco trucks have saved 3,570 lb. (1,623 kg). In this assembly the Resco trucks were fitted to the carbody with special adapters that provide the interface between the trucks and the carbody bolster, which was designed to suit a standard truck. If the carbody bolster were redesigned to provide the mounting surfaces required by the Resco truck, it would weigh no more than it currently does, possibly less. In that case the mass of the adapters, 1,300 lb. (591 kg), would also be saved, bringing the total savings to 4,870 lb. (2,214 kg) per car. Even if some carbody weight increase is necessary to accommodate the truck interface, it seems reasonable to estimate at this time that the total savings will be at least 4,500 lb. (2,045 kg) per car for a production design.

2.3 Angle of Attack

The test consist was brought to a stop on the 12 curve and the angles of attack were measured. A second car was included in the consist, borne on standard three-piece freight trucks, so the axles of this were measured for comparison. The results are seen in Table 4.

Truck	Axle	RH Curve	Efficiency
Steered Leading	Lead Axle	0.0	107.1%
	Trail Axle	0.05	
Steered Trailing	Lead Axle	0.0	85.8%
	Trail Axle	-0.1	
Unsteered Leading	Lead Axle	0.68	3.3%
	Trail Axle	0.00	
Unsteered Trailing	Lead Axle	0.79	-5.3%
	Trail Axle	0.05	

Table 4. Angles of Attack – 12 Curve

On a 12 curve the included angle between the axles should be 0.7 . If a truck is completely rigid, then the difference of the angles of attack for both axles should be equal to this value. If a truck is perfectly steered, then the difference of the angles of attack for both axles should be zero. [Note: Skewing or warping (lozenging) of the truck can produce equal positive or negative angles of attack at both axles, but the difference will always be zero if the steering angle between the axles is correct.] The actual steering angle between the axles is, therefore, the theoretical angle minus the difference in the angles of attack. We define the steering efficiency as the actual angle divided by the theoretical angle (x 100 to express it as a percentage). This value is included in Table 4.

It is clear that the steered truck has achieved a high level of steering efficiency (slightly oversteered in the leading position and understeered in the trailing position) and that the standard truck, not surprisingly, achieves nearly none. The standard truck, in fact, can produce anti-steering actions, as seen here in the trailing truck, because of the effects of flanging.

A similar test was done on the 7.5 curve in the balloon track, where the theoretical steering angle is 0.437 , and the results are seen in Table 5.

Truck	Axle	RH Curve	Efficiency
Steered Leading	Lead Axle	0.0	102%
	Trail Axle	0.009	
Steered Trailing	Lead Axle	0.0	93%
	Trail Axle	-0.03	

Table 5. Angles of Attack – 7.5 Curve

The steering efficiencies are again very high and the angles of attack are extremely low.

2.4 Truck Warping

Warping is a serious problem with many existing three-piece trucks, especially when they become worn. This phenomenon occurs when the two sideframes of the truck shear longitudinally relative to one another, producing a “lozenge” shape of the frame instead of the intended rectangular form. The tests at NRC (see section 2.1, Table 2) revealed that the three-piece truck under that car was in a seriously “lozenged” or “warped” condition as a leading truck in the curve. In a standard three-piece truck the only resistance to this condition is from the friction wedges between the bolster and the sideframes.

The Resco Steered Frame Truck contains a shear connection between the two sideframes, designed to resist lozenging deformation. As this is different from the standard freight car truck layout, instrumentation was added to the truck in order to determine the effectiveness of the design.

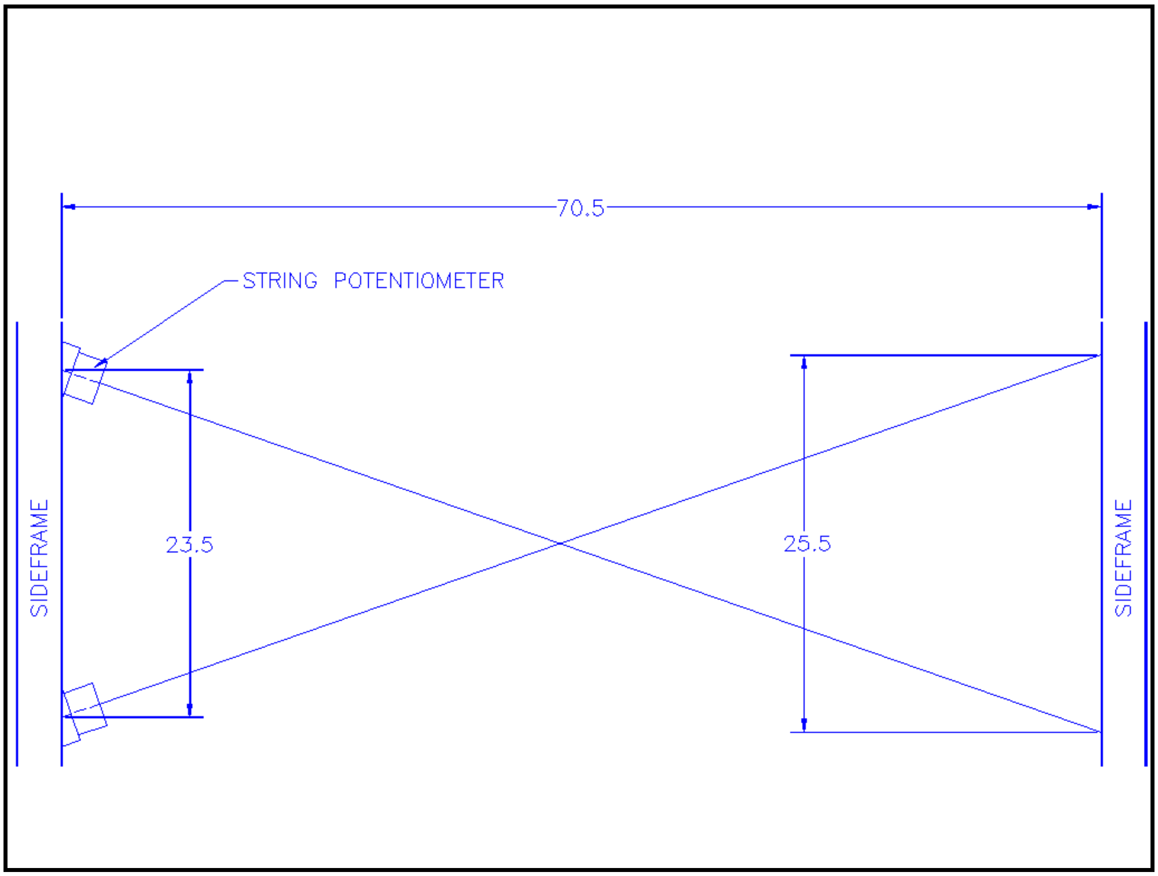

Figure 12. Stringpot Arrangement – Warp Measurement

Two string potentiometers were mounted on brackets on one sideframe with their strings directed at angles across the frame to attachment points on the other sideframe. The arrangement is shown in Figure 12.

In a 12 curve it was found that one stringpot had extended by 0.040 in. (1 mm) and the other had contracted by 0.037 in. (0.94 mm). This represents a total shear motion between the sideframes of less than 0.12 in. (3.05 mm), or less than 0.1 . This compares very well with the extremely minor angle-of-attack errors found in the curves, and indicates that the truck does not warp significantly during curving. In tangent track and shallower curves the warping was even less.

2.5 High-Speed Behaviour and Dynamic Stability

Initial high-speed testing was conducted in November 2000. The loaded car achieved only 50 mph (80 km/h) on the TTT before testing had to be terminated. Termination occurred, not because the truck was unstable, but because greater than expected vertical bouncing was observed, and it was concluded from this that the dampers were not functioning properly.

2.5.1 Lateral and Vertical Body Accelerations

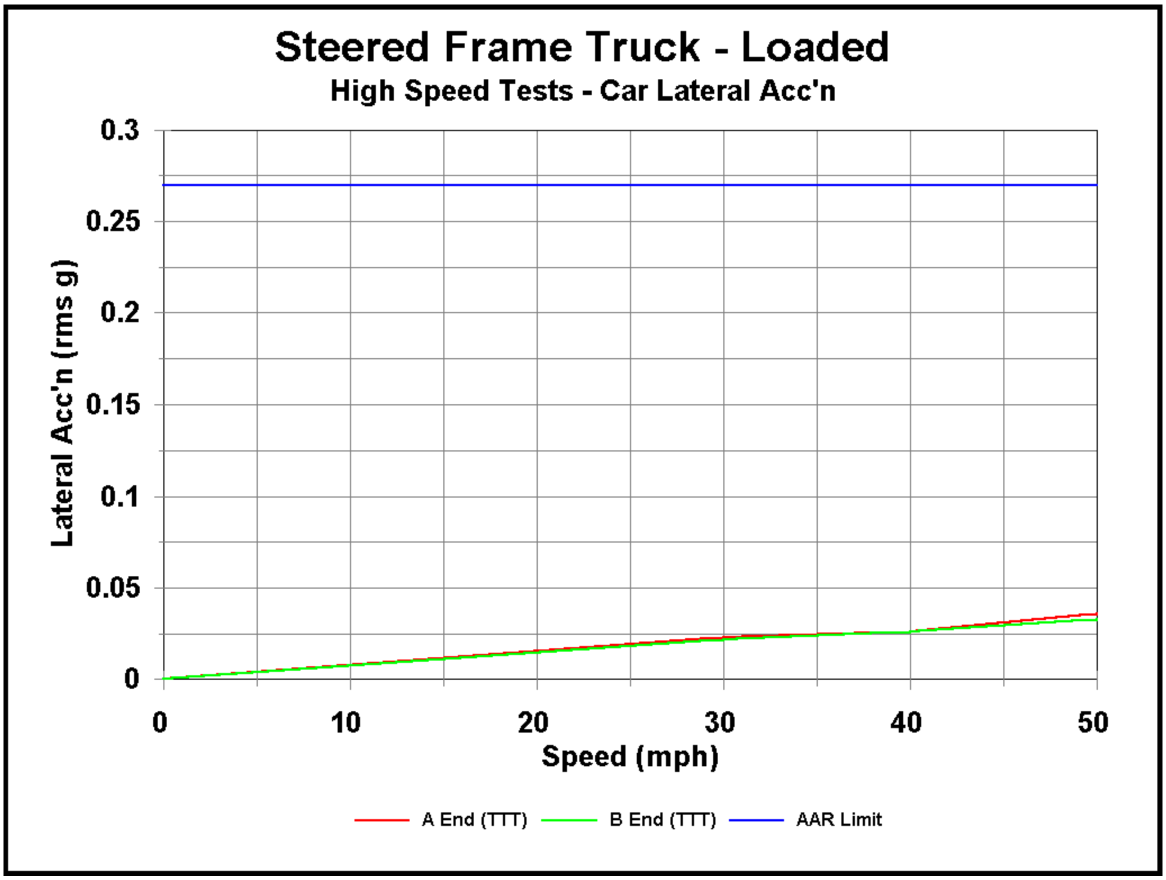

Figure 13. Lateral Accelerations – High-Speed Testing – Resco Truck – Loaded Car, No Damping

The lateral accelerations can be seen in Figure 13. Clearly, from a comparison with the criteria shown in Figure 2, and also with respect to the AAR limit of 0.27g (used to indicate the presence of hunting), no truck instability was present, which agrees with the visual observations made at the time. The vertical accelerations, however, were approaching 1g at 50 mph (80 km/h) and the car was visibly bouncing up and down.

The trucks were installed under an empty car, with the same (ineffective) dampers, and dynamic stability testing was completed up to 80 mph (128 km/h) on the TTT. The carbody lateral acceleration results are shown in Figure 14. It can be seen that the stability is extremely good, the criterion for Chapter XI testing, for instance (shown as a blue line at 0.27g), having been met with a very substantial margin to spare. Visual observation confirmed that truck hunting was not taking place.

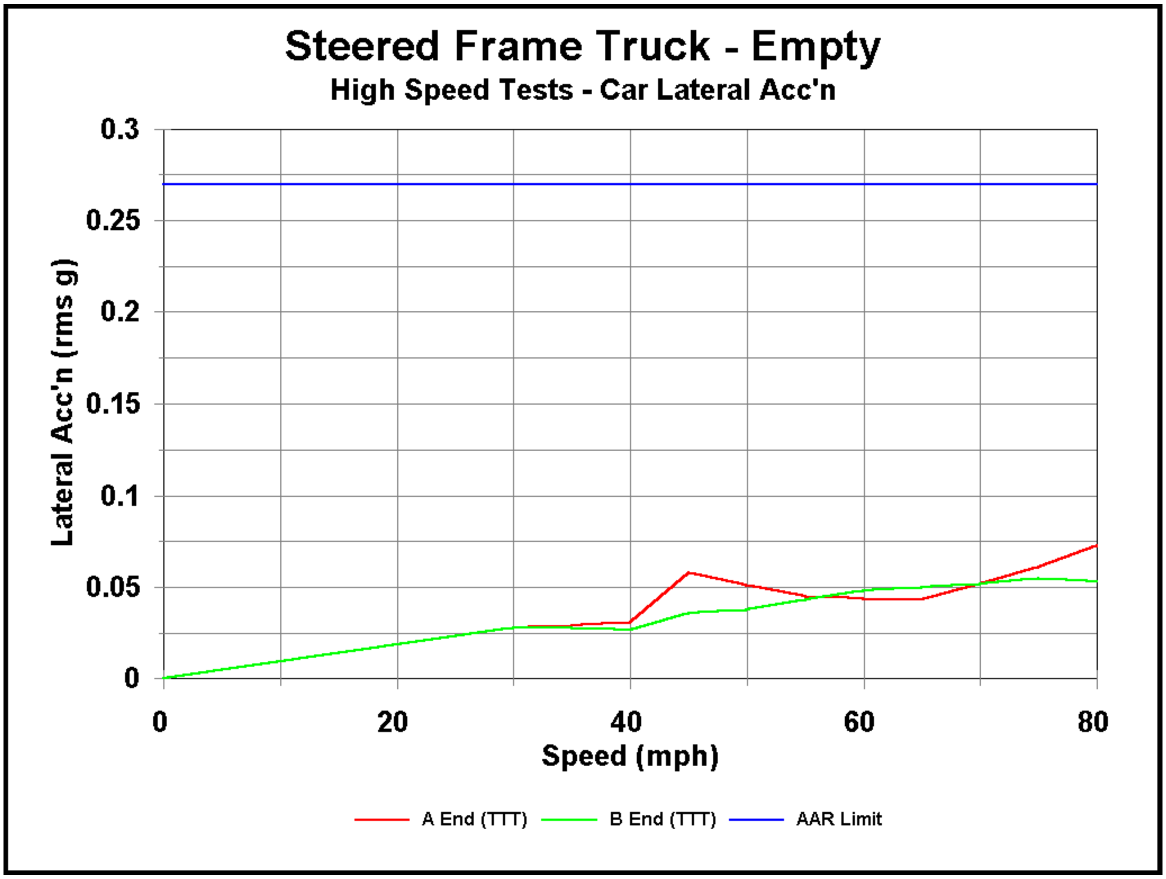

Figure 14. Lateral Accelerations – High-Speed Testing – Resco Truck – Empty Car, No Damping

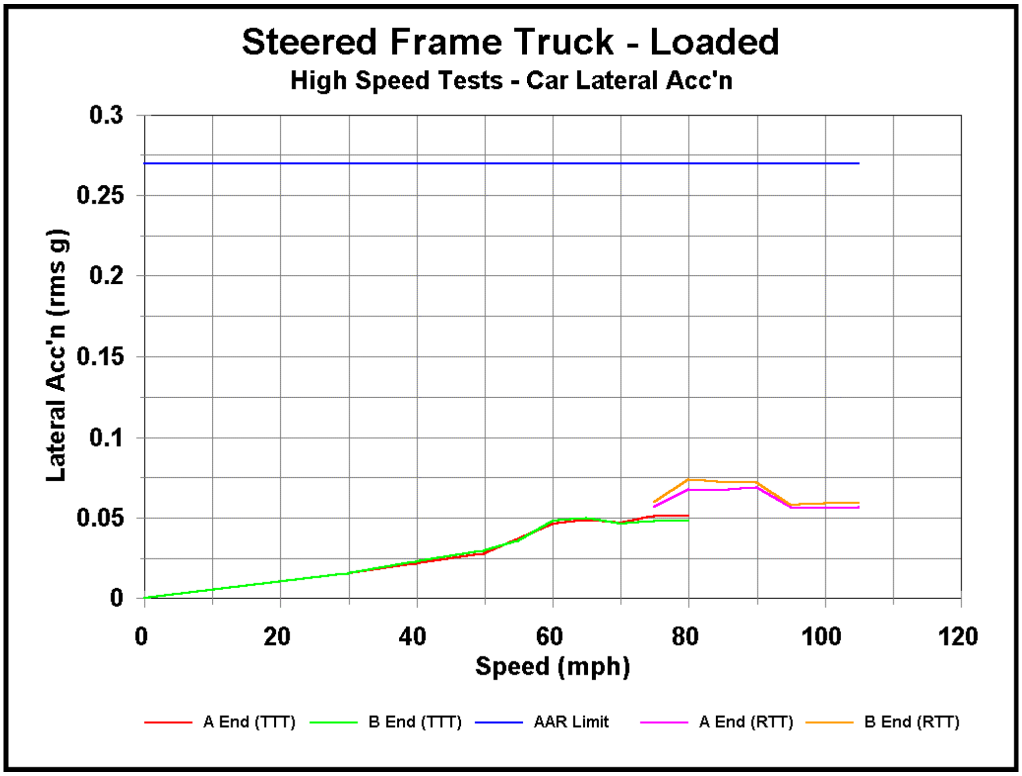

Figure 15. Lateral Accelerations – High-Speed Testing – Resco Truck – Loaded Car

The dampers were tested (see section 2.11) and found to be ineffective. They were rebuilt and retested and, once they had been re-installed in the trucks, high-speed testing for the loaded car was repeated in July 2001. The results for the TTT up to 80 mph (128 km/h), and for the RTT up to 105 mph (168 km/h), can be seen in Figure 15. The lateral acceleration criterion has again been met with a large margin and no sign of truck instability was visually observed.

Some interesting observations can be made concerning these results. The 75 and 80 mph (120 and 128 km/h) tests were repeated on the RTT and it can be seen that the values were slightly higher than for the TTT. This was somewhat unexpected as it is generally considered that the RTT has smoother track than the TTT. It can also be seen that the largest lateral accelerations occurred at about 60 to 90 mph (96 to 144 km/h) and that they reduced progressively after that, showing no tendency to increase even at 105 mph (168 km/h). No sign of lateral instability was observed. This is an indication that a resonance of the suspension was passed through in this speed range, resulting in the small peak, which disappeared at higher speeds. Looking at the results separately, it appears that the peak occurred at about 65 mph (104 km/h) on the TTT and at about 85 mph (136 km/h) on the RTT. No reason for this difference has been determined but it is surmised that there must be some difference in the track vertical inputs to account for it.

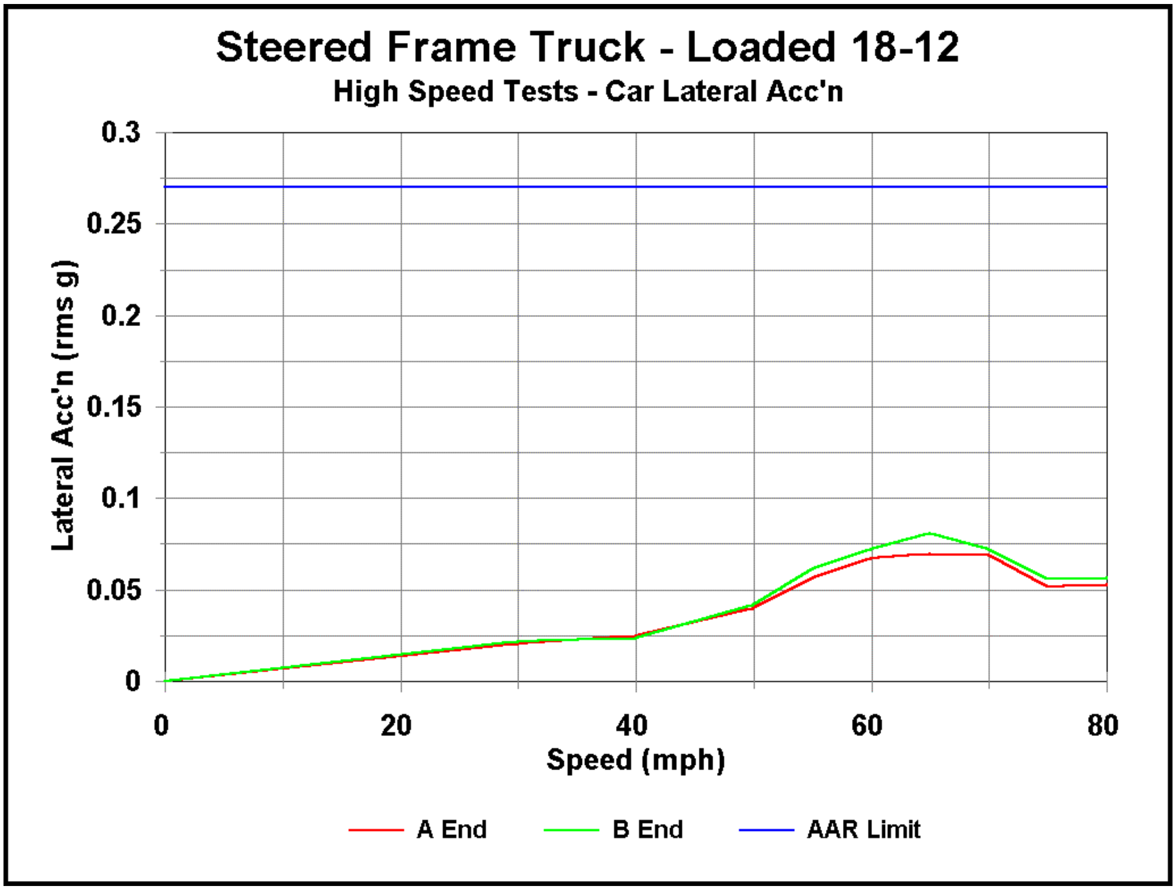

Figure 16. Lateral Accelerations – High-Speed Test – Resco Truck – Tangent TTT, Loaded Car

Figure 16 shows the results from the Resco truck in the loaded condition on a section of tangent track outside the official hunting test zone, also on the TTT. Again the small peak can be seen, also at about 65 mph (104 km/h), and the overall results are very similar to those from the official zone.

Figure 17 shows the results of high-speed testing up to 80 mph (128 km/h) in the empty condition on the TTT, and beyond 80 mph (128 km/h) on the RTT, with the rebuilt dampers installed. It can be seen that the lateral acceleration criterion is easily met up to 80 mph (128 km/h), although the A end of the car does have slightly greater accelerations than when it was tested earlier (see Figure 13). To that point there was no sign of impending instability. Beyond 80 mph (128 km/h) testing on the RTT, the lateral accelerations increased rapidly at both ends, usually a sign that truck hunting has begun. However, no sign of lateral instability was observed visually; the truck that could be seen (B end) appeared to be totally stable. The carbody lateral motions could be observed and it was clear that these were becoming rapidly larger. It was also clear, from the wheelset force measurements, that the vertical force variations were becoming quite large. The test was stopped when the minimum vertical wheel force approached 10% of the static load, which occurred at about the same time as the lateral accelerations approached the AAR limit. Those results are examined in section 2.5.2.

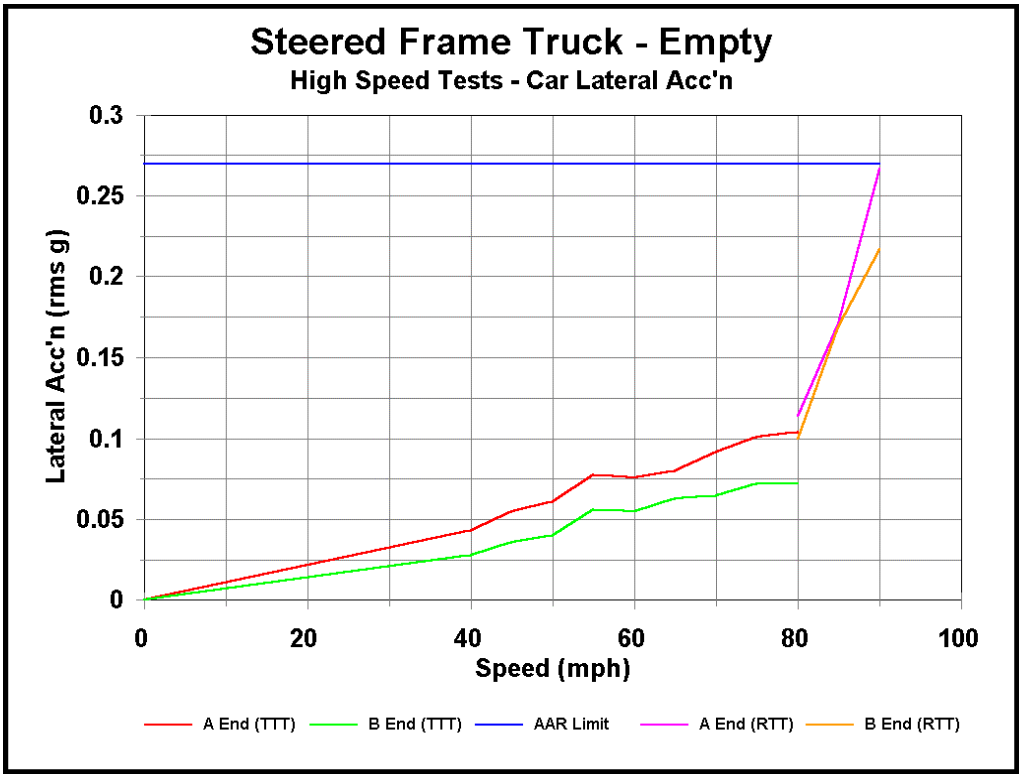

Figure 17. Lateral Accelerations – High-Speed Testing – Resco Truck – Empty Car

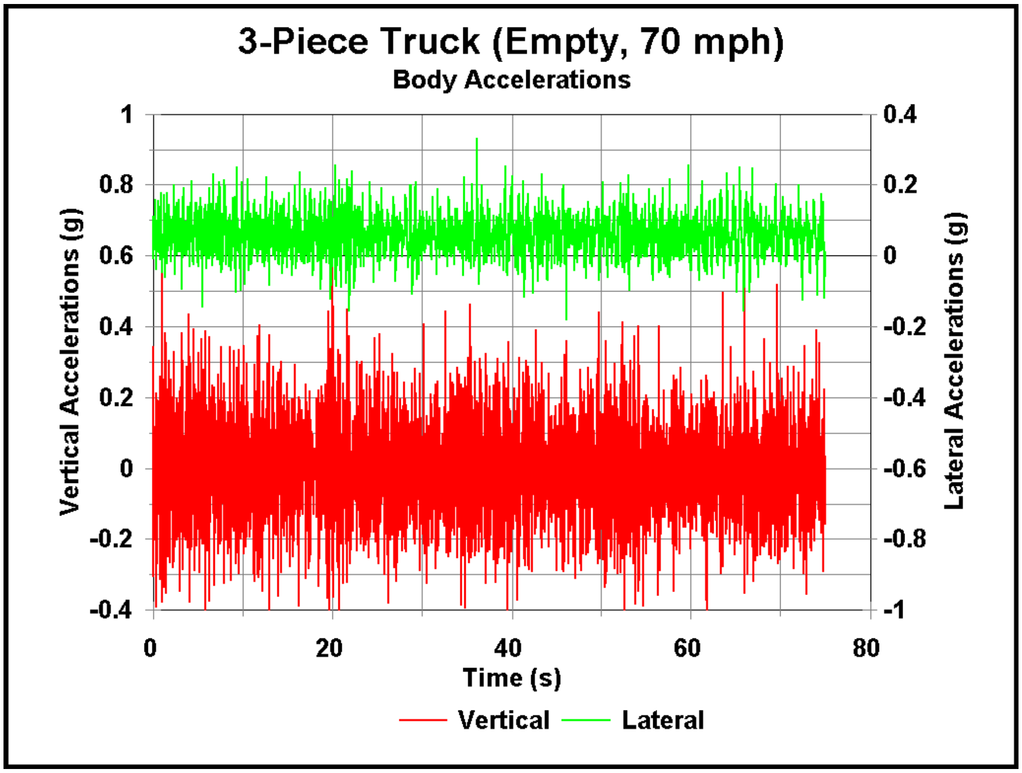

Figure 18. Body Accelerations – Three-Piece Truck, Empty

Figure 18 shows the actual acceleration values for a period of 75 seconds through the high-speed test section of the TTT. These results are from the baseline three-piece truck test. The upper graph is for the lateral accelerations and is plotted against the scale on the right ordinate axis, whereas the lower graph is for the vertical accelerations and is plotted against the left ordinate axis. It can be seen that there is a small offset of the zero for the lateral values.

Figure 19 shows corresponding plots for the Resco truck in the empty case and Figure 20 shows the results for the Resco truck in the loaded case. Comparing Figure 19 with Figure 18 we see that there is only a small difference in the overall lateral behaviour, the Resco truck exhibiting slightly lower overall vibration. In the vertical direction, however, there is a substantial improvement of the Resco truck over the baseline truck.

Figure 20 shows somewhat similar results to those of Figure 19, the lateral values appearing to be slightly worse and the vertical values somewhat better.

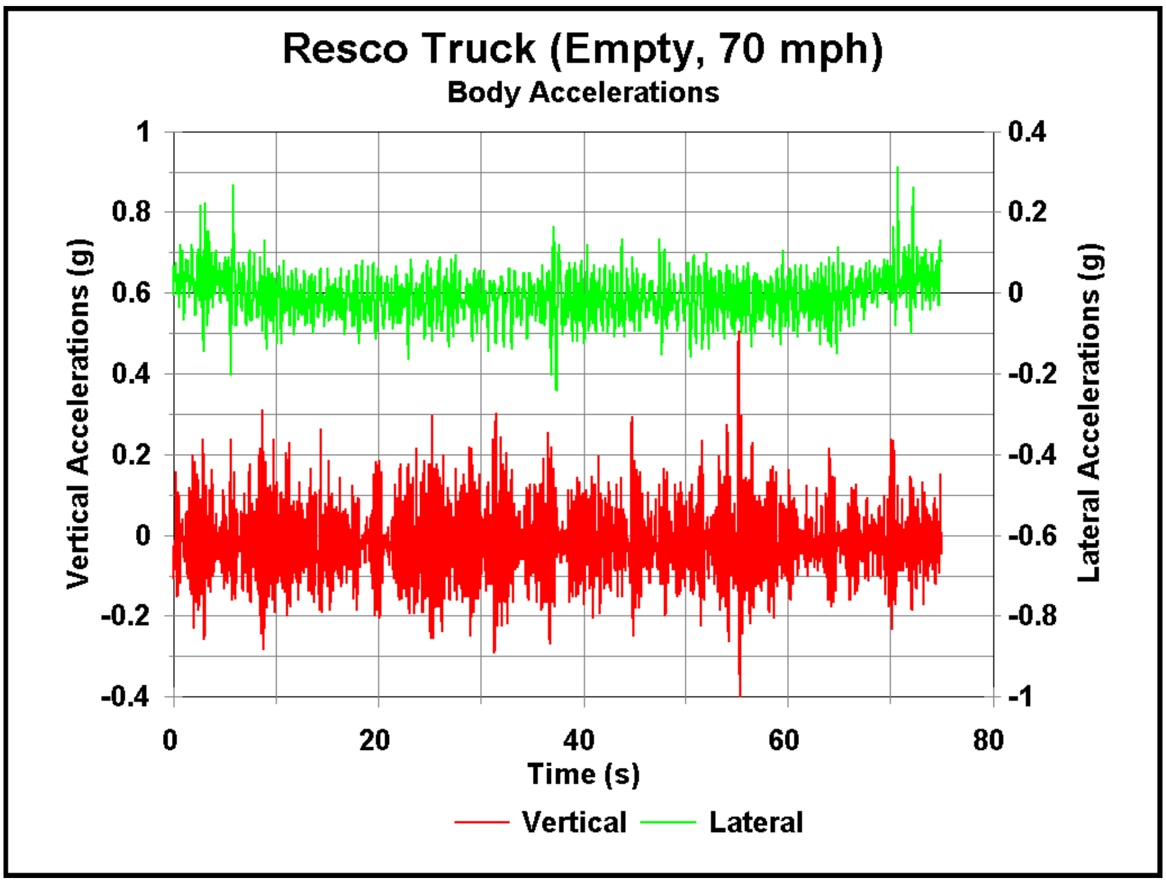

Figure 19. Body Accelerations – Resco Truck, Empty

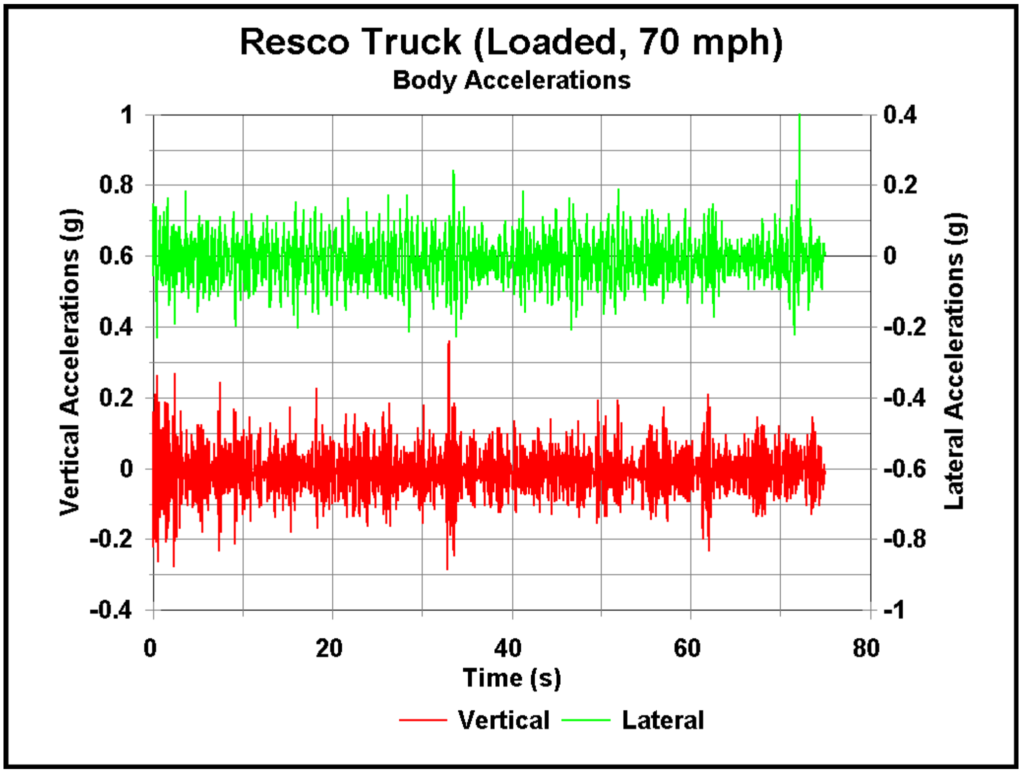

Figure 20. Body Accelerations – Resco Truck, Loaded

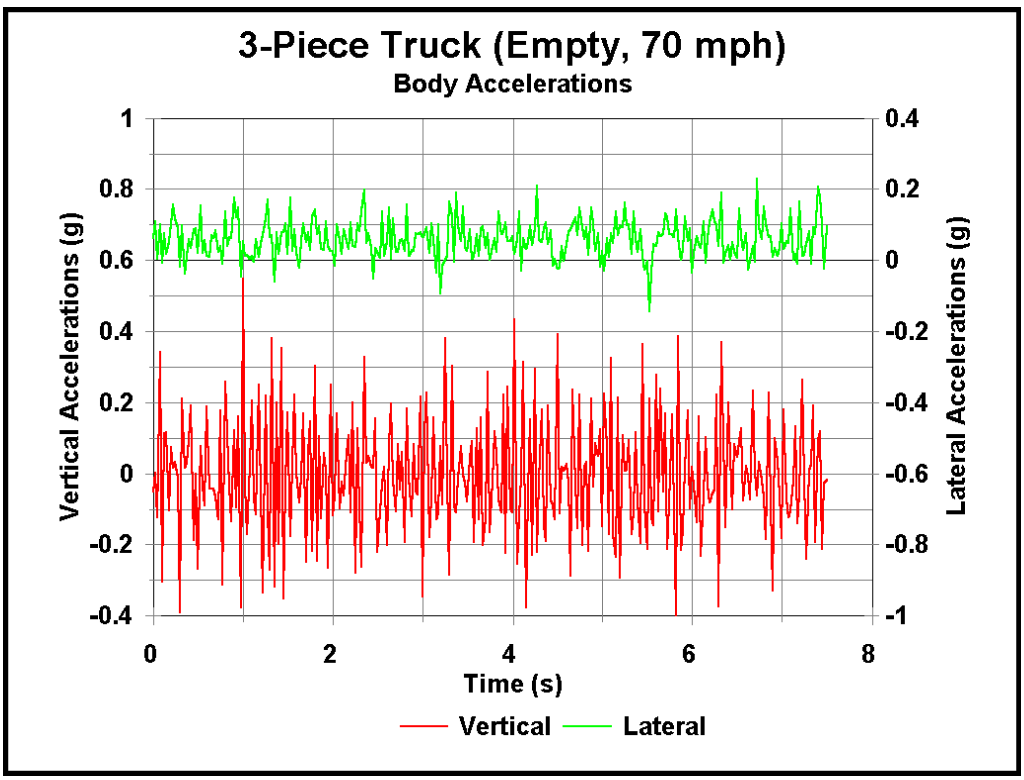

Figure 21. Body Accelerations – Three-Piece Truck, Empty – Expanded Scale

Figure 21 shows the same results as Figure 18 but on an expanded abscissa of only 7.5 seconds. This allows the dynamic behaviour to be seen more clearly.

It can be seen here that the vertical behaviour has a very high content of high-frequency motions, whereas the lateral behaviour shows a significant level of lower frequency motions with a smaller amount of higher frequency superimposed on it.

Figures 22 and 23 are for the Resco truck in the empty and loaded cases, respectively. In these figures it can be seen that the motions, especially the lateral motions, are dominated by a low-frequency (about 4 Hz) and show only a small amount of high frequency content.

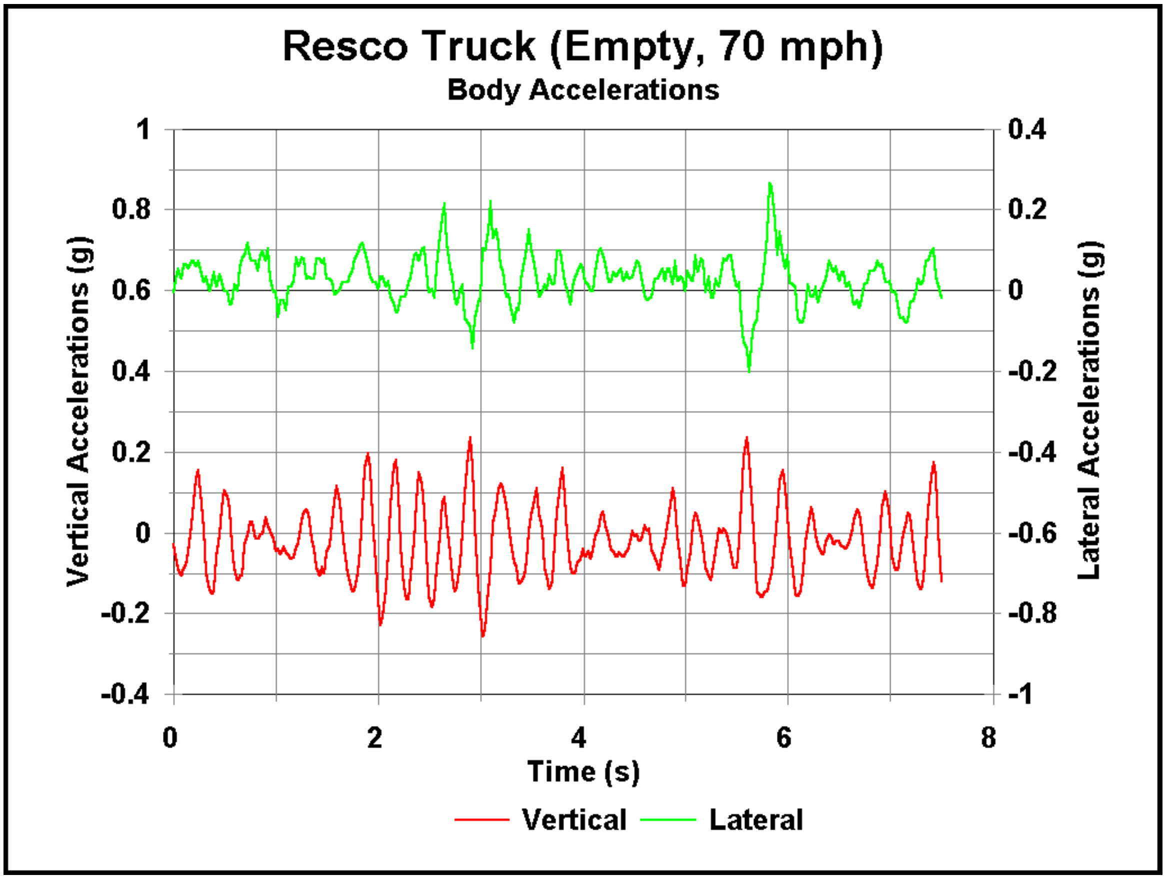

Figure 22. Body Accelerations – Resco Truck, Empty – Expanded Scale

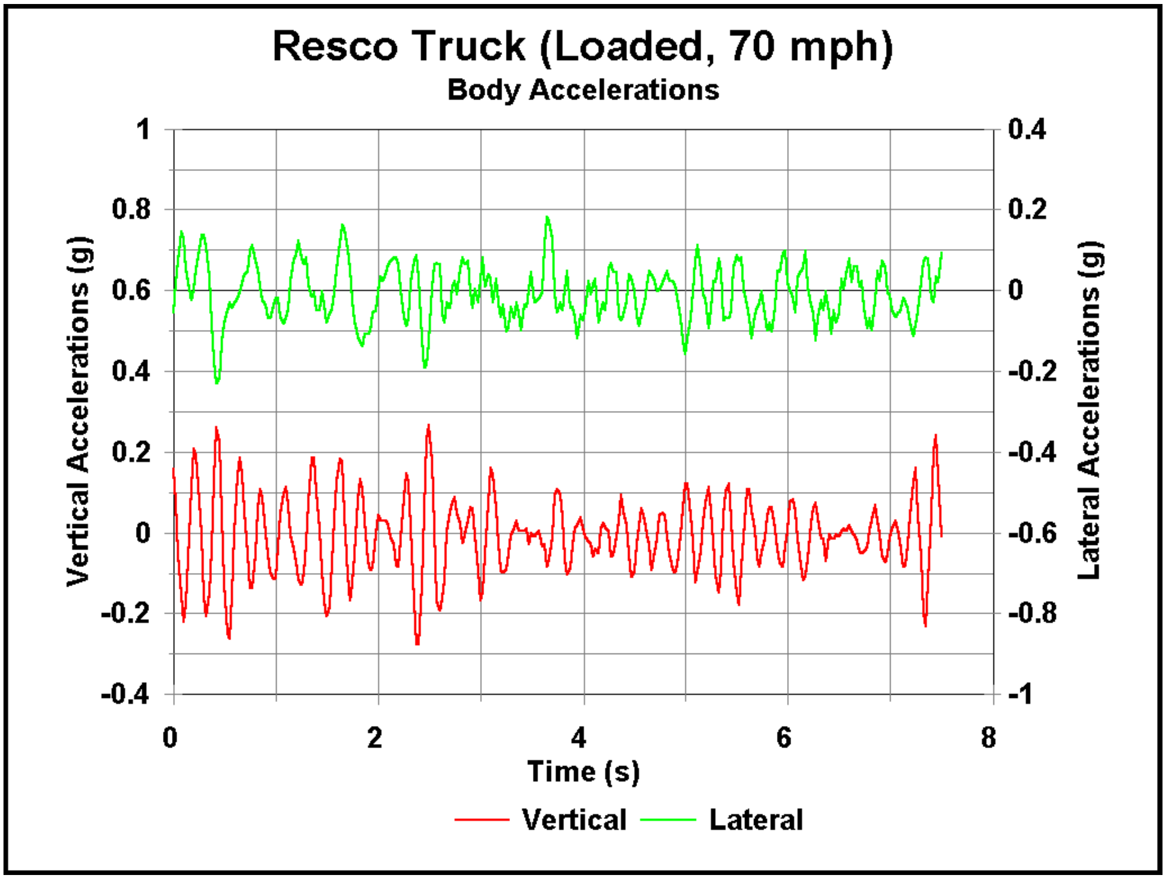

Figure 23. Body Accelerations – Resco Truck, Loaded – Expanded Scale

The data plotted in Figures 18 through 23 was further analyzed using the ISORIDE program from the A'GEM suite of dynamics analysis programs. This program performs a Fast Fourier analysis on the data and produces a spectrum of the accelerations broken down into 1/3 octave bandwidths. Figure 24 shows the comparison of the results from this analysis for the three cases shown above – the baseline truck (empty), the Resco truck (empty) and the Resco truck (loaded).

It can be seen here that the three cases are fairly similar at the lower frequencies, the Resco truck (empty) exhibiting slightly more than the baseline (empty) case at 1.25, 2 and 2.5 Hz and less at all other frequencies. At the higher frequencies (above 10 Hz) the Resco truck is substantially better than the baseline truck, except at 16 Hz. At 16 Hz the difference between the two trucks is less, although the Resco is still better by a margin of about 1.86:1. 16 Hz corresponds to a wavelength roughly equal to the axle centre distance in the trucks. The loaded case for the Resco truck is worse than the empty case for virtually all frequencies.

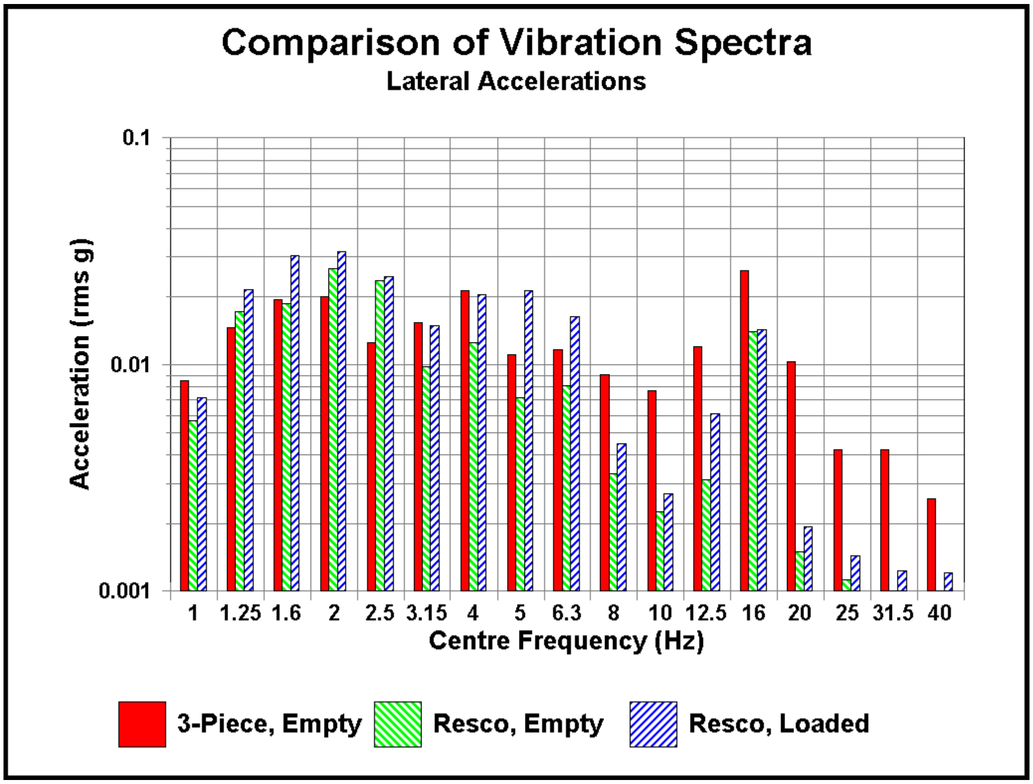

Figure 24. Vibration Spectra – Lateral Accelerations

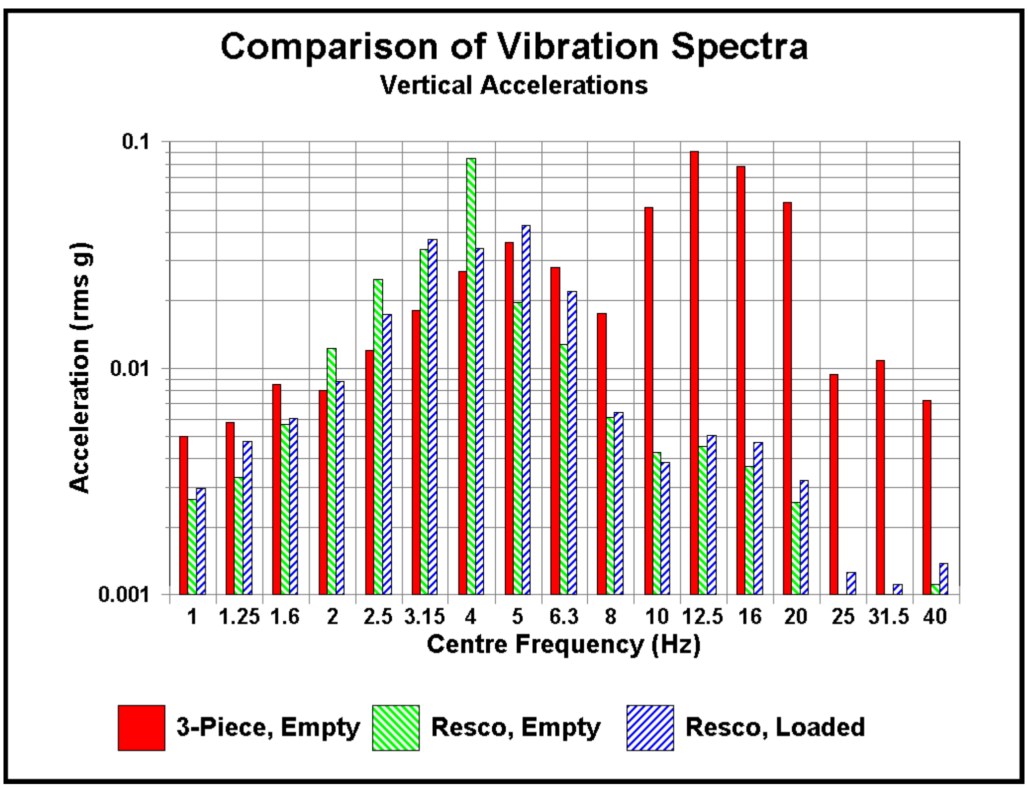

Figure 25. Vibration Spectra – Vertical Accelerations

Figure 25 shows the spectra for the vertical accelerations. Greater differences between the baseline (empty) and the Resco (empty) cases are seen here. The Resco truck is worse than the baseline truck for the frequencies 2, 2.5, 3.15 and 4 Hz, and the baseline truck is worse at all other frequencies. The first peak for both trucks occurs at about 4 to 5 Hz. There is a second peak at about 12.5 to 16 Hz, which is very pronounced for the baseline truck and barely noticeable for the Resco truck. At 12.5 Hz the baseline truck is 20:1 worse than the Resco truck, and a similar difference is maintained over a broad range of frequencies, whereas at 4 Hz the Resco truck is 3.2:1 worse than the baseline and this only exists over a quite narrow range of frequencies.

The analysis also produces a value for the broadband acceleration – a measure of the average over the whole frequency range. The results are shown in Table 6.

	Three-Piece, Empty rms g	Resco, Empty rms g	Resco, Loaded rms g
Lateral	0.0573	0.0514	0.0680
Vertical	0.1543	0.0990	0.0733

Table 6. Lateral and Vertical Broadband Accelerations

The spectra and the broadband values produced by the analysis agree very well with the observations made from the plots of the raw data. This type of analysis is very useful in pinpointing the dynamic behaviour of a truck or suspension, allowing the designer to make informed decisions about the nature of the inputs and how to effect any changes that might be required of the suspension.

2.5.2 Vertical Wheel Forces

A second measure of the dynamic performance of the truck at high speed is in the vertical forces of the wheels on the rails. It is important that the wheels maintain a good level of contact and also that they do not produce excessive impact forces on the rails.

Figure 26 shows the minimum vertical force at each of the wheels through the test section on the TTT for the empty car during the first tests (ineffective dampers). It can be seen that there is very little variation from the static loads and that the limit of 10% of the static loads, shown in the figure as a straight line, is not approached. The small decline with speed is to be expected, as there will be an increasing dynamic component of the wheel load that will result in slightly lower minimums being recorded as speed increases.

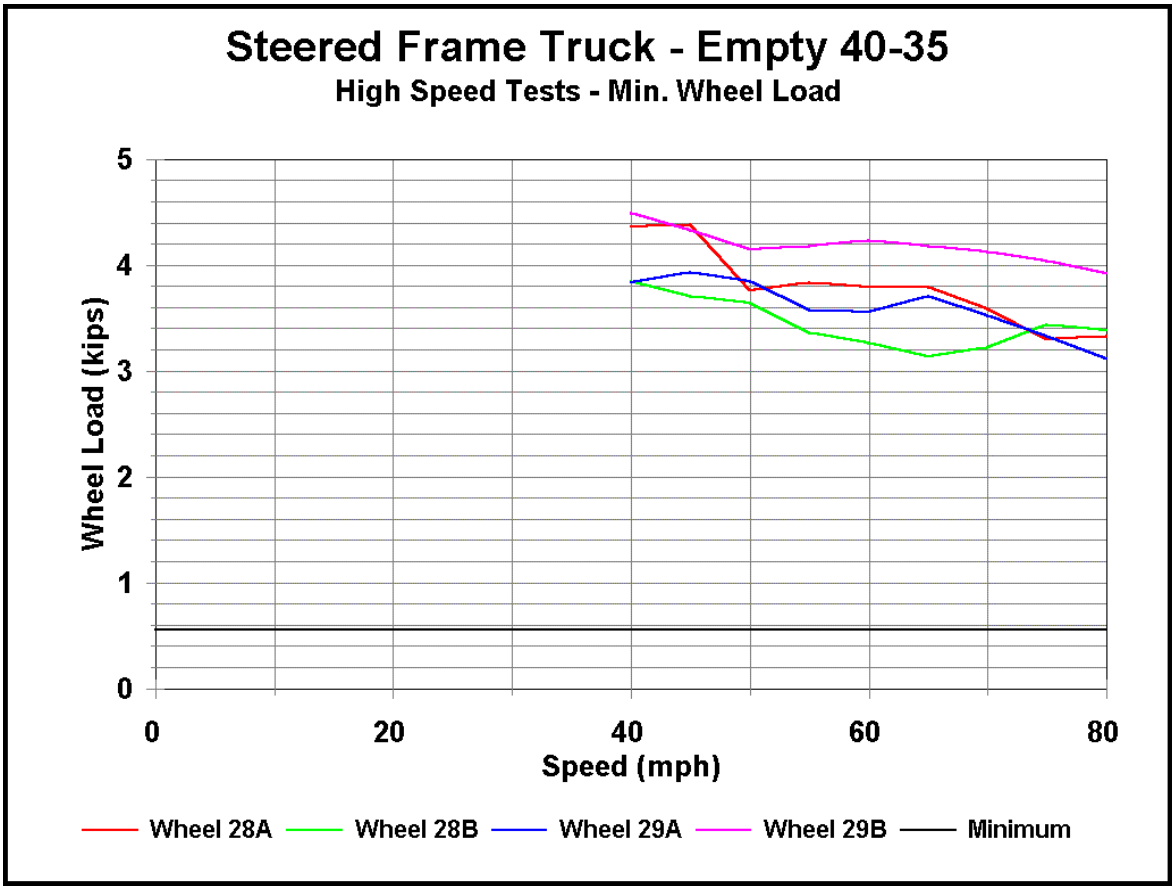

Figure 26. Minimum Wheel Vertical Force – High-Speed Tests (Empty) – Resco Truck, No Damping

The maximum vertical forces at the wheels are shown in Figure 27. Again, it can be seen that only very slight variations due to the increasing vertical dynamics are experienced. The car and truck are very well behaved, despite the ineffective damping.

Figure 28 shows the minimum force between each of the wheels and the rail through the test sections on the TTT and the RTT for the loaded car. In this case the dampers were rebuilt and working according to specification. The minimum force allowable is 10% of the static load and it can be seen that this value was never approached in the loaded condition.

The values measured have a certain randomness in them because the measurement is of the absolute minimum force observed in a dynamic test run. Even runs at exactly the same speed will show significant variation in the minimum value recorded from one run to the next. With this in mind it can be seen that there is no significant change in minimum wheel load over the whole speed range.

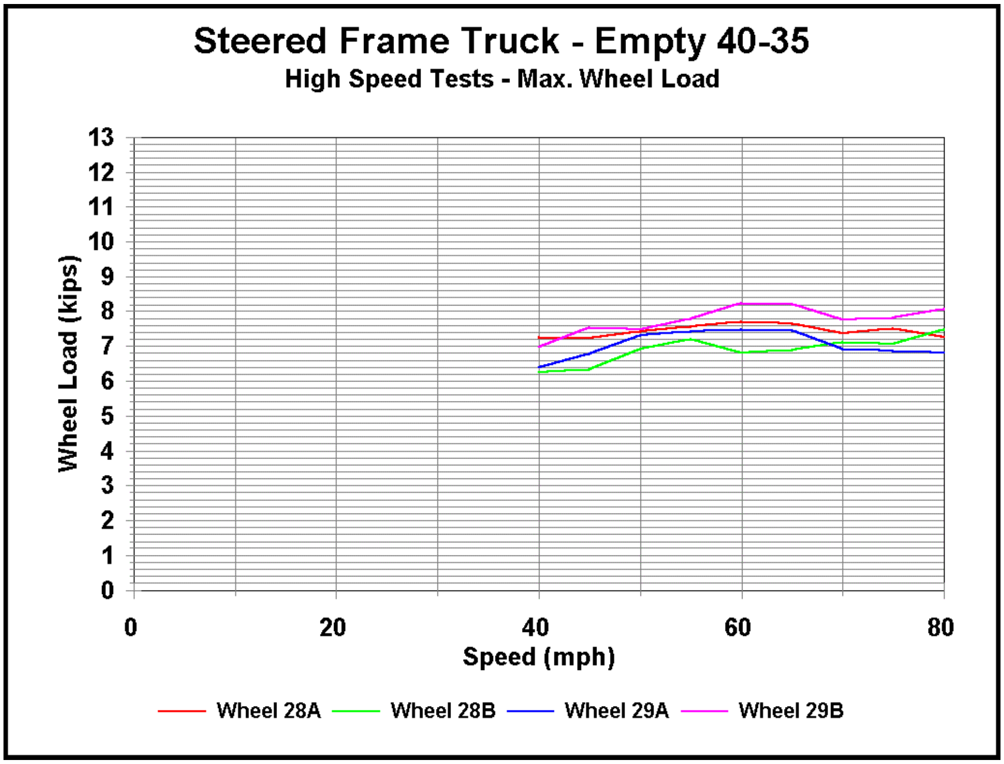

Figure 27. Maximum Wheel Vertical Force – High-Speed Tests (Empty) – Resco Truck, No Damping

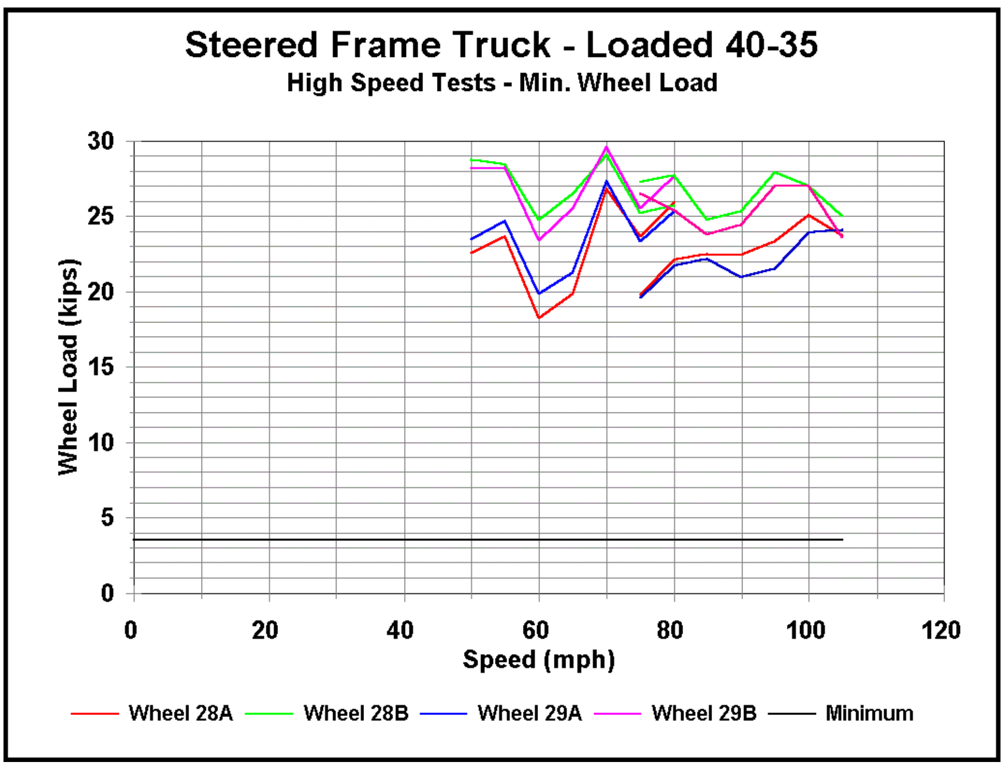

Figure 28. Minimum Wheel Vertical Force – High-Speed Tests (Loaded) – Resco Truck

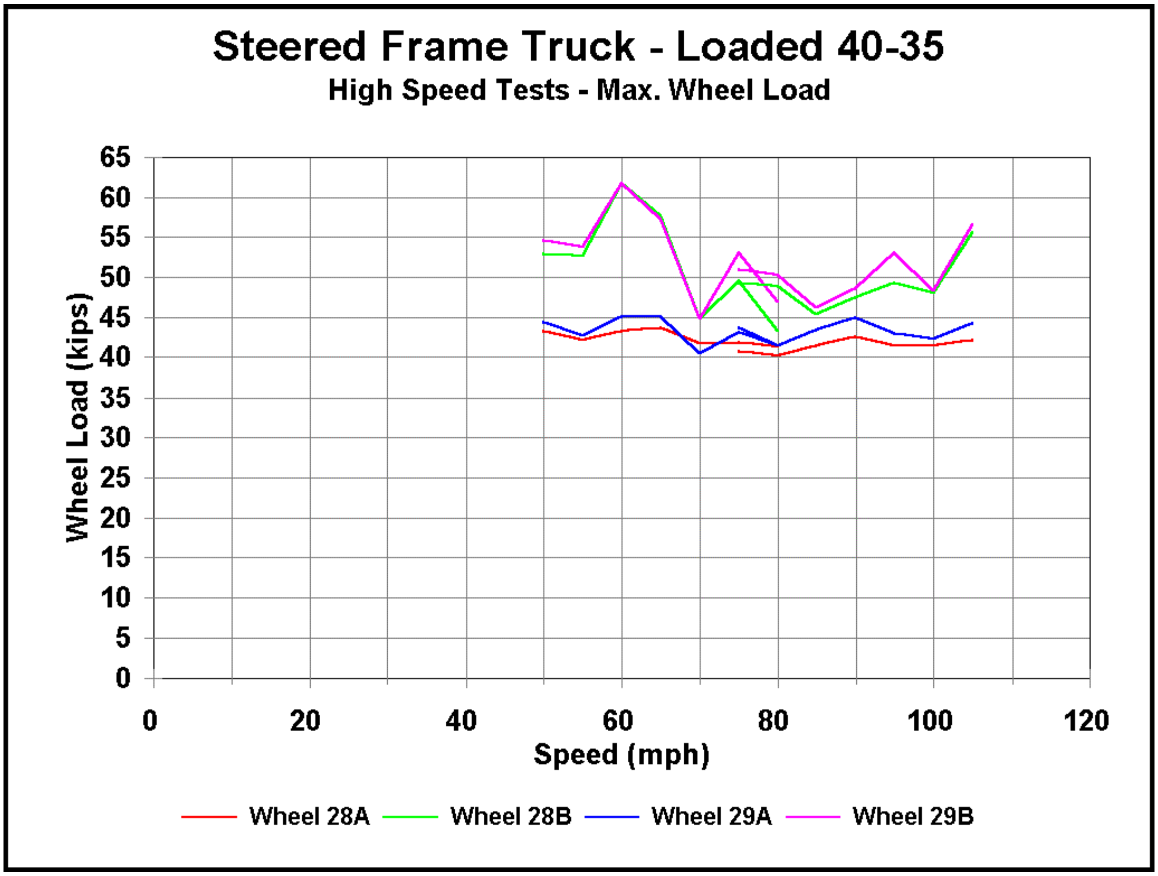

Figure 29. Maximum Wheel Vertical Force – High-Speed Tests (Loaded) – Resco Truck

Figure 29 shows the maximum vertical wheel loads corresponding to the minimums shown in Figure 28. Very little variation is seen all the way to 105 mph (168 km/h).

It is interesting to note, from both the minimum and maximum force measurements, that the A side wheels consistently have lower loads than the B side. It is felt that this must indicate a slight off-centre of the load because the track in this section is nominally level and tangent.

Figure 30 shows the minimum wheel load for the empty car in the high-speed tests with the rebuilt dampers. It can be seen here that, although the car lateral accelerations gave no indication of truck hunting and neither was there any visual indication of it, the minimum wheel load reduced rapidly at speeds above 70 mph (112 km/h). By 90 mph (144 km/h) the lower limit (10% of the static load) was reached and testing was terminated, even though no sign of truck hunting was present.

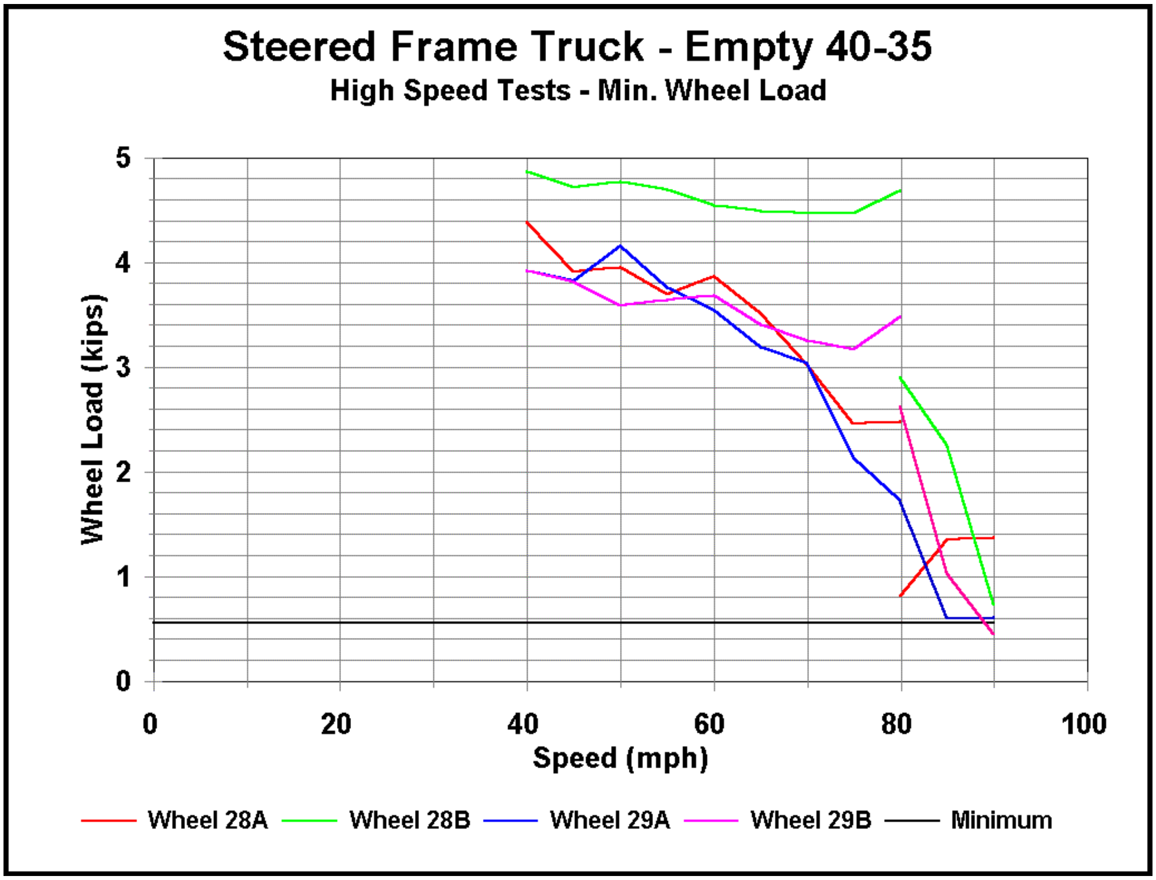

Figure 30. Minimum Wheel Vertical Force – High-Speed Tests (Empty) – Resco Truck

The maximum wheel load, seen in Figure 31, interestingly, did not increase significantly until speeds above 80 mph (128 km/h) were achieved, even though the minimum value started to decrease at 70 mph (112 km/hr). Above 80 mph (128 km/h), significant increases were seen, and by 90 mph (144 km/h) the peak load approached double the static load.

These results are very different from the results obtained earlier, with ineffective dampers installed (see Figures 26 and 27). This is discussed further in section 4.

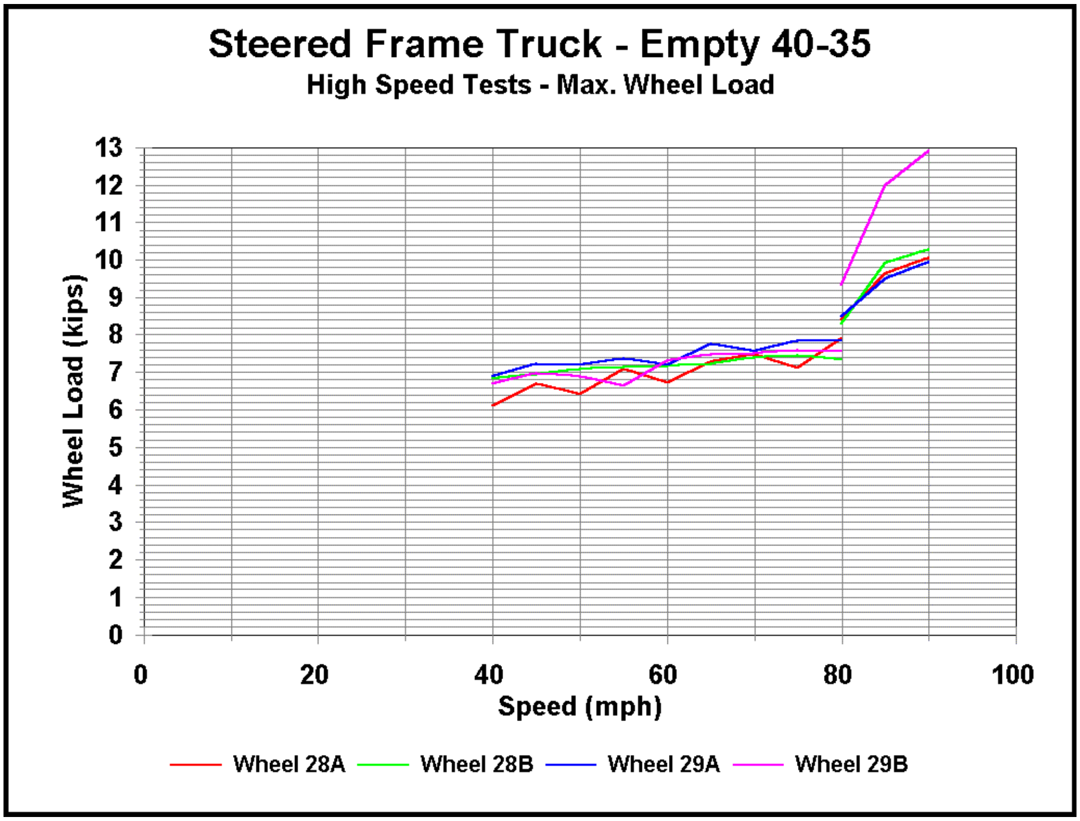

Figure 31. Maximum Wheel Vertical Force – High Speed Tests (Empty) – Resco Truck

2.6 Rolling Resistance

Initial rolling resistance measurements on tangent track, made in November 2000, showed a rather higher than expected value. The result was somewhat higher than a standard three-piece truck normally achieves. The curved track rolling resistance measurements, however, showed a very small increase in rolling resistance compared with the tangent track result, even on the 12º curve. Typically, even a well-behaved conventional or premium truck will produce a substantial increase in rolling resistance on curves, doubling the tangent track value by about 2 to 4 of curvature. The lack of a substantial increase in rolling resistance on curves was taken as a good sign that the steering mechanism was working properly.

A careful examination of the truck behaviour in tangent track showed that the leading truck was slightly "skewed" relative to the track alignment, whereas the trailing one was not. When the travel direction was reversed, the truck that had been aligned became the leading one and was slightly skewed, whereas the other truck, which had been skewed, was now the trailing one and was properly aligned. This is a phenomenon known as the "Weinstock Effect" [2] and it occurs with steered trucks when low conicity wheelsets are utilized: as the wheels wear and the conicity increases, the self-centring ability of the wheelsets overcomes the skewing and both trucks become fully aligned.

A simple design modification was installed in June 2001 that counteracts the Weinstock Effect without compromising the steering qualities and that, therefore, allows low conicity wheelsets to be used. The tests were repeated. A small amount of skewing was still present in the lead truck on tangent track but the tangent track rolling resistance was now lowered to below the levels found in standard trucks. It was felt that further improvement might be possible but that the truck was now ready, in this condition, for curving testing. Again it was found that there was almost no increase in the rolling resistance in curves, compared with that on tangent track.

The four graphs shown in solid lines in Figure 32 are from the data recorded in four runs through each of the curves and the tangent track. It can be seen that there is extremely good consistency in these results. The dashed black line represents a linear fit that follows the formula

$$\text{Force (lb./ton)} = 1.62 + 0.06\text{x}C$$

where C = curvature (degrees).

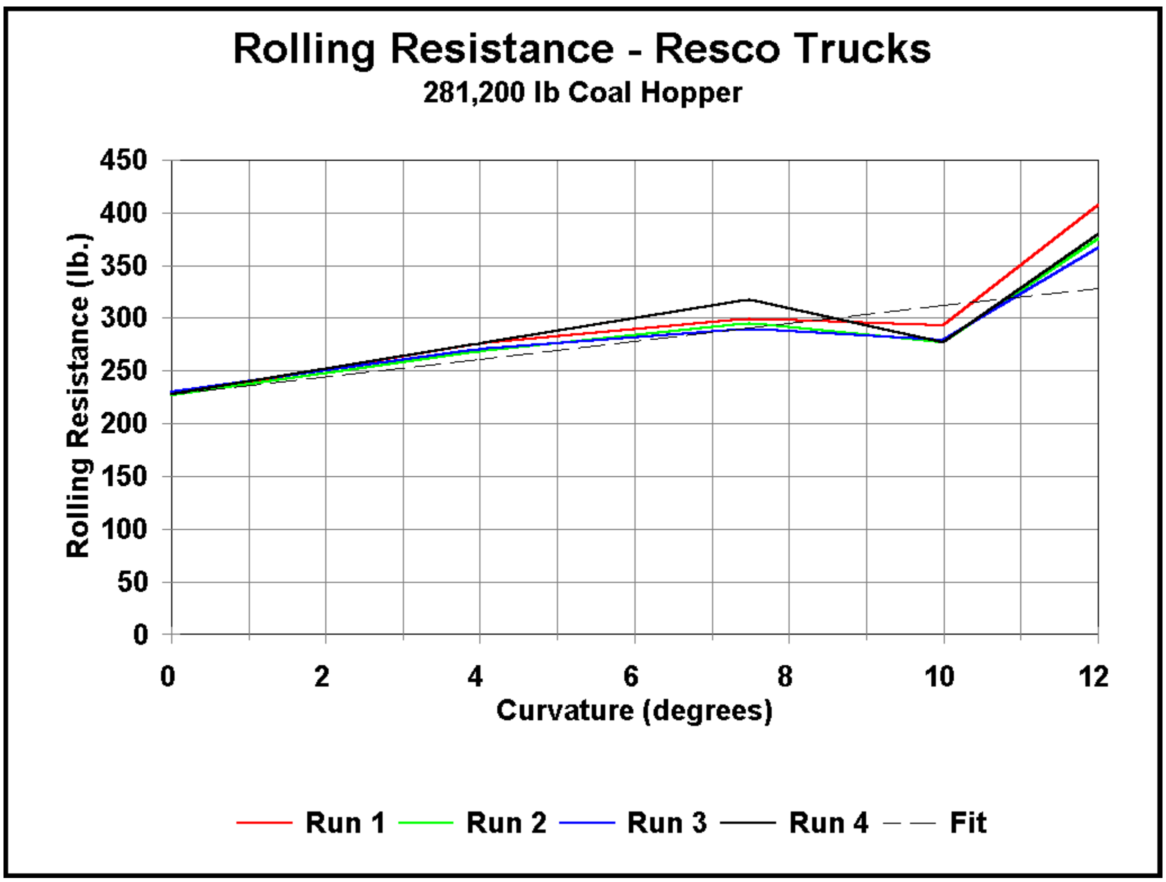

Figure 32. Rolling Resistance – Resco Truck

The reduction in resistance at 10 of curvature, relative to that at 7.5 , is interesting and quite consistent. No explanation for this is apparent; it may have something to do with the rail profile at that point or some other unknown factor.

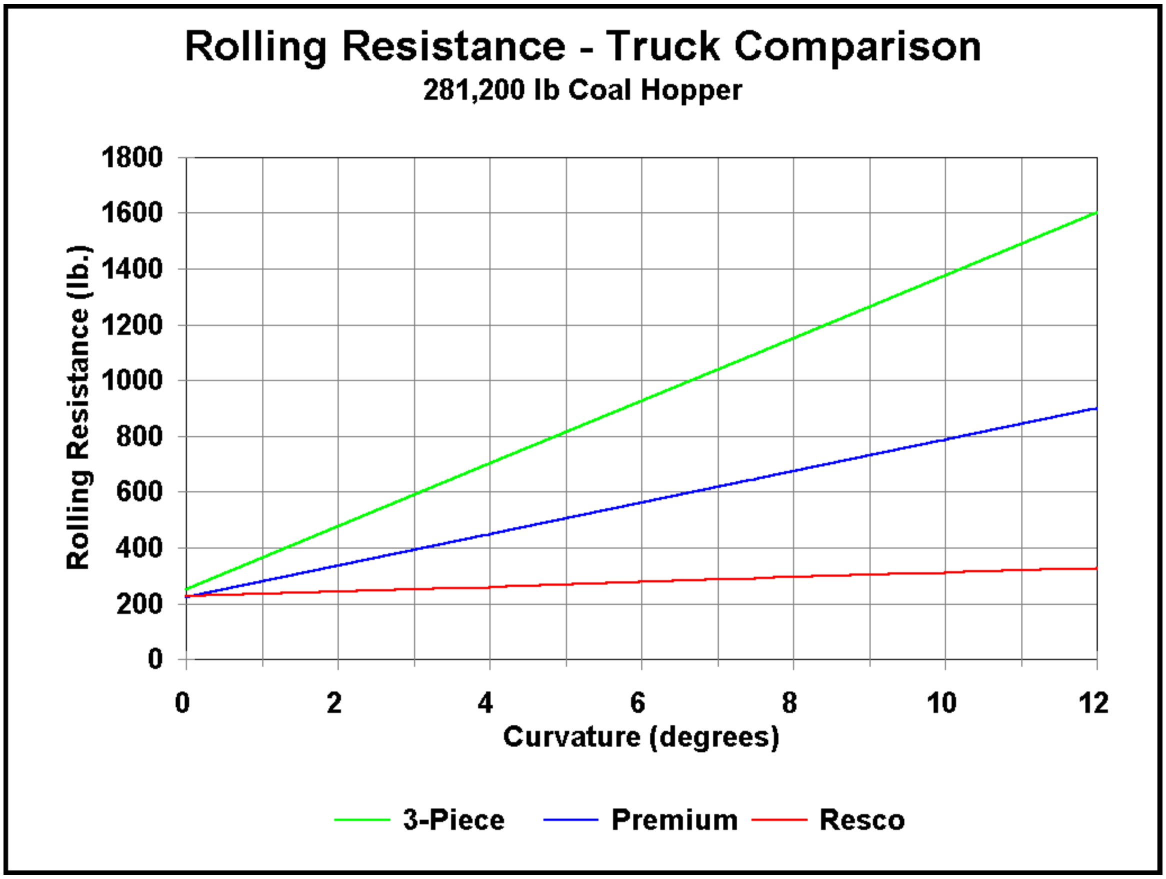

Figure 33. Comparison of Rolling Resistance

Figure 33 shows a comparison of the results from this testing with the values typically obtained from two other types of trucks. The standard three-piece truck typically produces a rolling resistance given by

$$\text{Force (lb./ton)} = 1.8 + 0.8\text{x}C$$

although this is often exceeded by trucks in less than optimum condition. The tangent track value of 1.8 often becomes 2.4 or even higher. A well-behaved premium truck (such as the frame-braced truck, for instance) will produce resistance given by

$$\text{Force (lb./ton)} = 1.6 + 0.4\text{x}C$$

The improvement in performance from the Resco truck, seen in Figure 33, is very pronounced. On a 12 curve the Resco truck produces approximately 20% of the rolling resistance of the standard truck and 36% of that of a premium truck. To put it another way, the rolling resistance for the Resco truck is 44% greater than the tangent track value on the sharpest curvature tested – 12 . By comparison, the standard truck's rolling resistance is 533% greater on that curve than on tangent and the premium truck's is 300% greater.

The effect of the reduction of rolling resistance on overall fuel consumption in a railway environment is discussed in section 3.

2.7 Curving Behaviour – Lateral Forces and L/V Ratios

Measurements of wheel/rail forces were made on the leading truck as it passed through the 4º, 7.5º, 10º and 12º curves in the WRM loop, as well as in the transitions and the tangents.

The results of these measurements for the lateral direction on the outside wheel, compared with other trucks and the baseline case, can be seen in Figure 34.

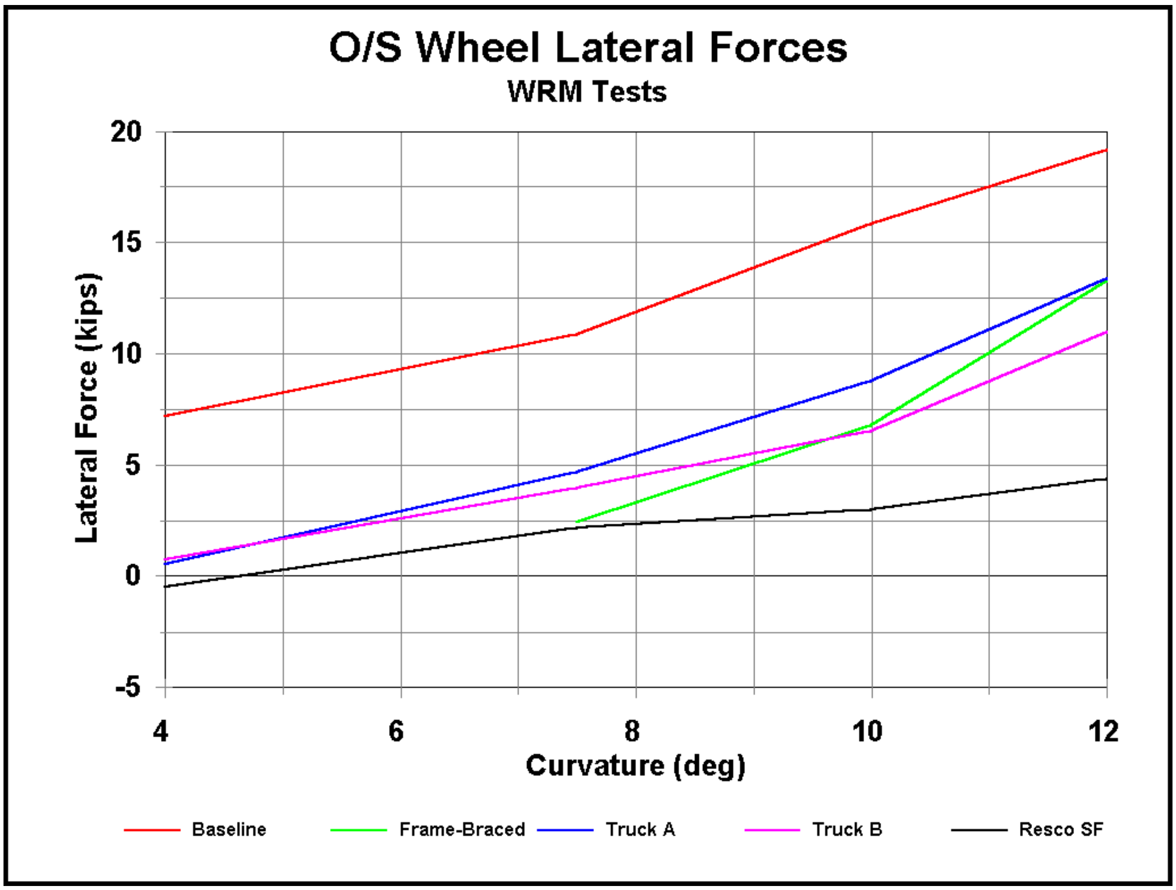

Figure 34. Comparison of Outside Wheel Lateral Forces

A very interesting aspect of the results for the Resco truck, apart from the obvious fact that the forces are lower than for all the other trucks, is the fact that the force is negative at the 4 curve. A positive force is one in which the rail is being spread outwards and an angular misalignment almost always creates a positive force on the rail. The only normal circumstances in which this is not the case is if an external force is being applied to the wheelset to hold it off the flange contact, in which case a positive angle of attack can lead to a negative net force on the rail. In this case, however, it seems unlikely that this is the explanation. It is more likely that the wheelset is almost perfectly aligned, creating little or no lateral force from angular creep, and that the negative force is produced by the small spin creep term due to the conicity. In the case of standard, non-steered trucks this component is so small that it is usually swamped by the lateral creep term. In this case it is visible due to the almost zero lateral creep.

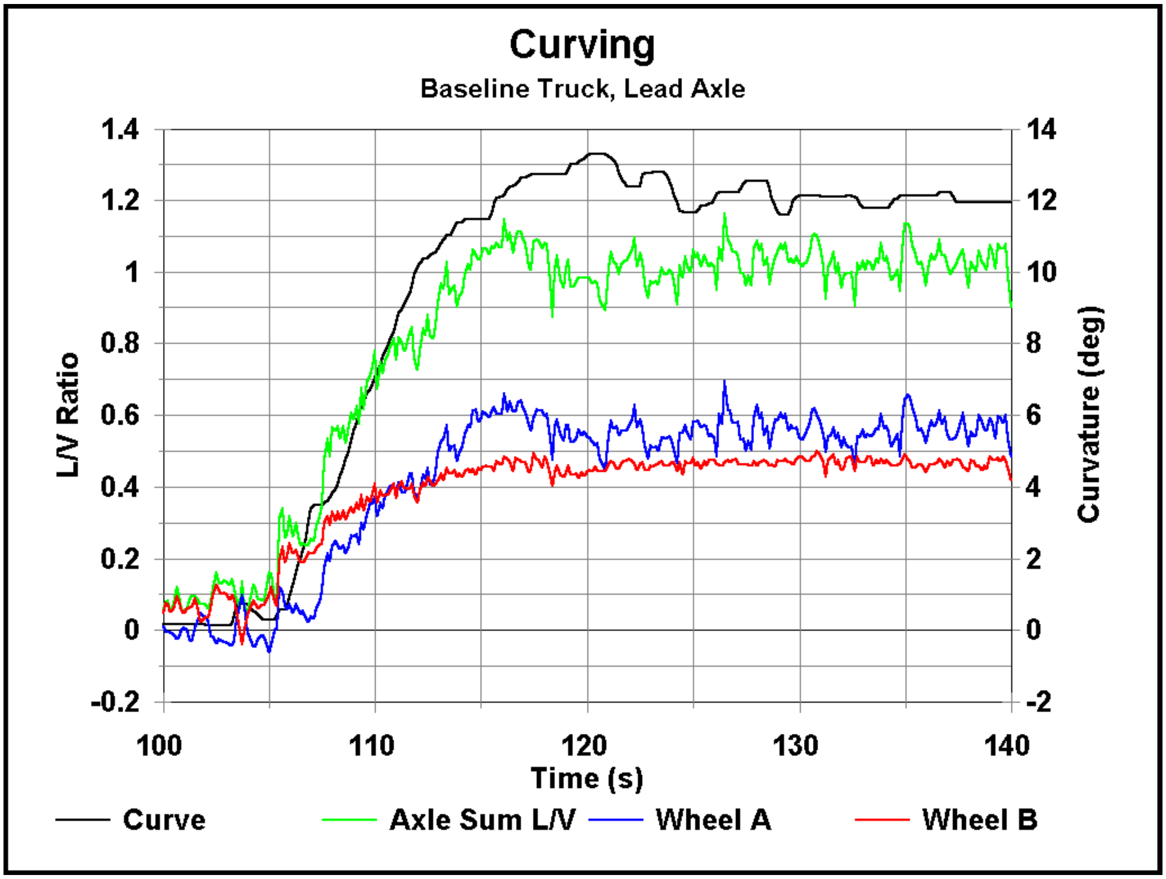

Figure 35. Baseline Truck – L/V Ratios on 12 Curve

Figure 35 shows the results of L/V measurements on the baseline truck for the leading axle as it enters and traverses the 12 curve on the WRM loop. The curvature is shown by the black line and the scale is on the right hand ordinate. It can be seen that all values start out, on the left of the diagram, at near zero, and that as the curvature increases, all of the L/V values increase virtually proportionally. The axle sum value exceeds 1.1 on several occasions and averages about 1.05 when fully into the curve. The worst individual wheel value approaches 0.7 and it averages about 0.58 when fully into the curve. The AAR limit for these values is 1.3 for the axle sum L/V and 0.8 for the maximum wheel L/V. The limit is not exceeded but is being approached quite closely.

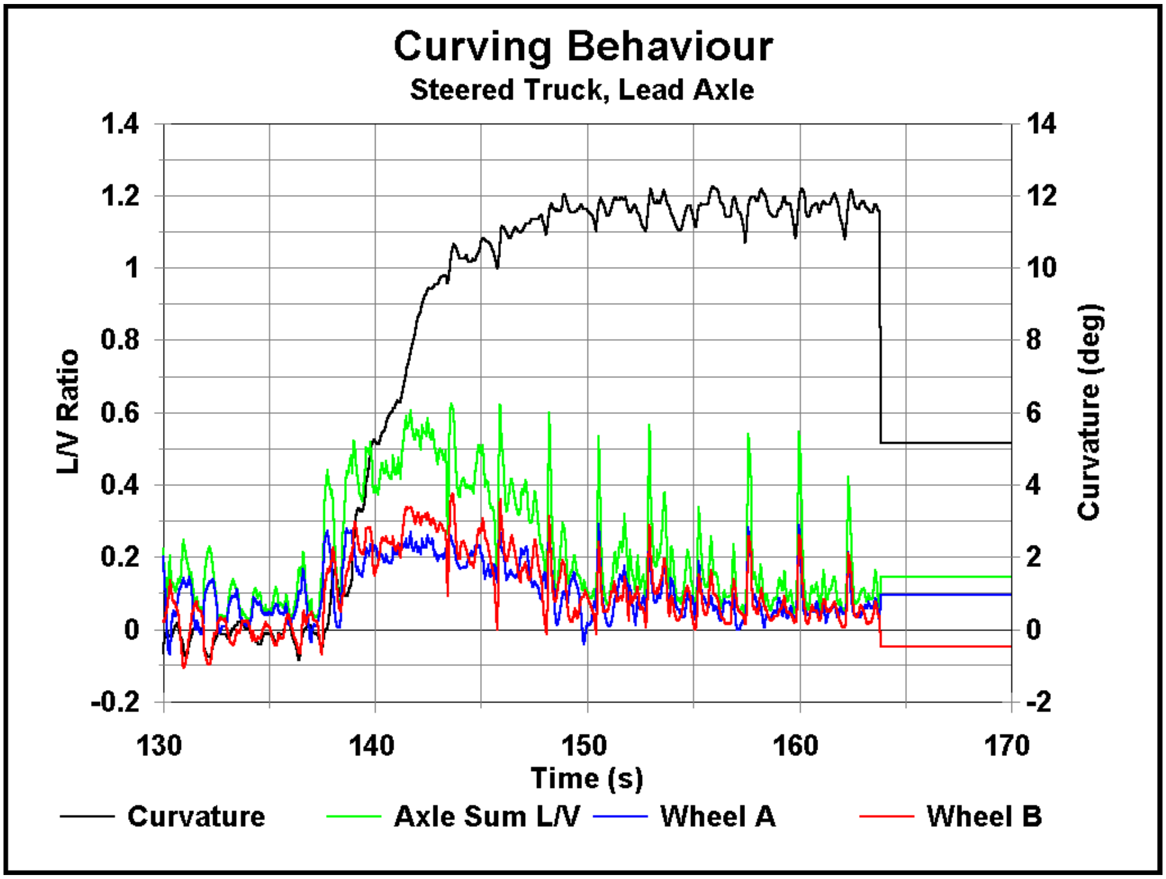

Figure 36. Resco Truck – L/V Ratios on 12 Curve

Figure 36 shows corresponding results to those in Figure 35, this time for the Resco truck. It is clear how much improvement there is for this case. The peak axle sum values are just over 0.6 and they occur in the spiral transition; the average is around 0.1 when fully into the curve. Similarly for the maximum single wheel value, the peak is around 0.38 and it averages around 0.05 when fully into the curve. These values are a very safe distance from the AAR recommended maximums, indicating an improved margin of safety against derailment on sharp curves as well as reduced wear and tear on the track.

It is interesting that the steered truck shows regular peaks in the L/V values at about every 2 seconds, corresponding to about every 79 ft. through the curve. It is not known what the cause of this phenomenon is. The distance corresponds almost exactly to 2 rail lengths but it is not known whether there is any disturbance at this wavelength on the WRM loop. Only very minor disturbances are seen in the baseline case, and not at a very well-defined wavelength.

2.8 Wheel Load Equalization

The measurements of wheel load as one wheel is lifted out of the plane of the other three are recorded in Table 7 for the Resco truck under a loaded car.

Wheel	Lift at Wheel 28B			
	0 in.	1 in.	2 in.	3 in.
28A	33,720 lb. 96%	30,740 lb. 87.5%	27,210 lb. 77.5%	25,080 lb. 71.4%
28B	35,990 lb. 102.5%	39,300 lb. 111.9%	42,780 lb. 121.8%	44,980 lb. 128.1%
29A	34,470 lb. 98.1%	34,480 lb. 98.1%	34,830 lb. 99.1%	34,480 lb. 98.1%
29B	36,330 lb. 103.4%	36,090 lb. 102.7%	35,600 lb. 101.4%	35,780 lb. 101.9%

Table 7. Wheel Load Equalization – Loaded Car – Resco Truck

Table 8 shows the results for a standard truck under an empty car.

Wheel	Lift at Wheel 28B			
	0 in.	1 in.	2 in.	3 in.
28A	3,324 lb. 113.1%	2,735 lb. 77.2%	1.818 lb. 62.5%	1,529 lb. 52.5%
28B	2,251 lb. 80.1%	3,494 lb. 124.6%	4,177 lb. 144%	4,540 lb. 157.1%
29A	2,460 lb. 88.3%	3,365 lb. 117.1%	3,409 lb. 118.7%	3,365 lb. 117.1%
29B	3,513 lb. 118.6%	2,354 lb. 81.1%	2,144 lb. 76.4%	2,114 lb. 74.8%

Table 8. Wheel Load Equalization – Empty Car – Standard Truck

The results for the standard truck are taken from the lifting direction of motion. As the wheel was lowered again, the loads were quite different. Figure 37 shows the variation of loads through a complete cycle of lifting and lowering. The effect of the friction at the wedges can clearly be seen in the hysteresis of the cycles. It is interesting to note that the wheel loads are quite unbalanced in the level condition. It is also remarkable that the results are extremely repeatable.

Figure 38 shows the results graphically for the Resco truck. The wheel loads started out very much more evenly distributed and varied much less throughout the lifting range than for the standard truck. There was no discernible hysteresis as the wheel lift was brought back to zero.

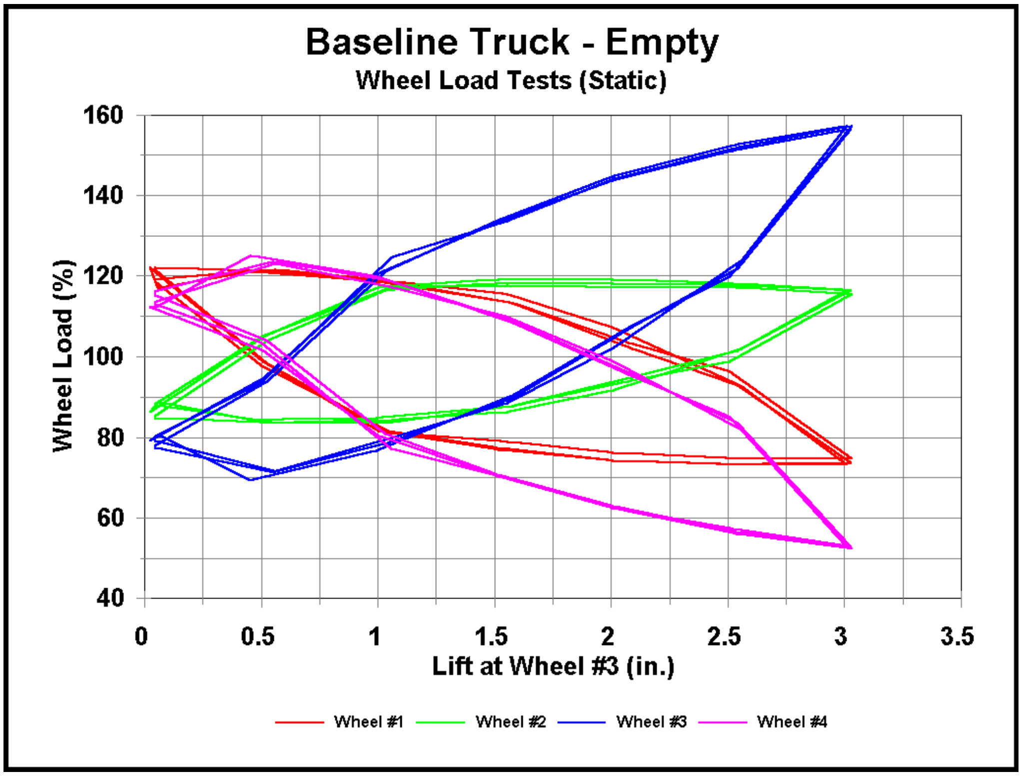

Figure 37. Static Wheel Unloading – Standard Truck – Empty Car

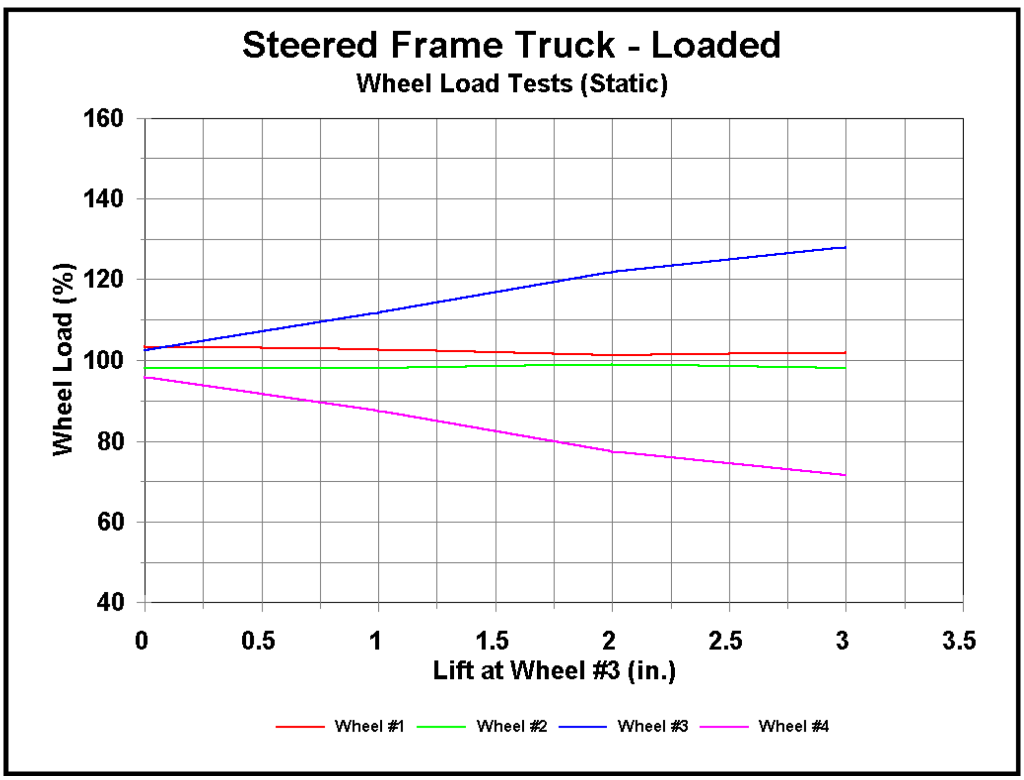

Figure 38. Static Wheel Unloading – Resco Truck – Loaded Car

It is unfortunate that there are no static wheel lift test results for the Resco truck under an empty car or for the standard truck under a loaded car. It is hoped that these results will be obtained at a later date. There was no opportunity during this test program to obtain those results, as this is not a normal test done during freight car truck testing and was not planned for the bulk commodity program. It is anticipated that the Resco truck under the empty car will produce very similar results to the loaded case, in terms of unloading, because the dual rate springs designed for this truck have a spring rate that bears the same relationship to the empty car mass as the loaded spring does to the loaded car mass. There might be some improvement of the empty car relative to the loaded car because the empty car spring is much smaller in diameter and thus will have proportionally lower bending stiffness than the loaded spring. The wheel unloading is affected by the bending stiffness of the spring because the base of the spring support takes up an angle relative to the car underfloor as the wheel lifts. For this reason it is felt that the empty car results for the Resco truck will be no worse, and probably better, than its loaded car results.

2.9 Dynamic Curving

In the dynamic curving tests there is a perturbation introduced in the track on a 10º curve, as described in section 1.1.5. The perturbation consists of a combination of vertical misalignment and lateral misalignment calculated to produce roll and sway motions while in the curving mode. It is important that the vehicle be able to negotiate track with this type of perturbation without excessive loss of load on any wheel.

The test was conducted at 12 mph (19.2 km/h) and then repeated at speeds increasing by steps of 2 mph (3.2 km/h) up to 24 mph (38.4 km/h). At 24 mph (38.4 km/h) a small wheel lift was observed on the trailing truck (i.e. not the truck with instrumented wheelsets). The performance at the leading truck, where measurements were being made, had not indicated loads below the 10% limit at that point. The behaviour was observed on a second pass through the section at 22 mph (35.2 km/h) and it was concluded that there was bottoming out of the lateral motion and/or the damper motions. In addition it was concluded that the lower centre roll motion, which produced the excessive motions, was underdamped with the arrangement as installed.

The cause of the inability of the suspension to negotiate the dynamic curving section of track at speeds above 24 mph (38.4 km/h) is discussed more fully in section 4.

2.10 Perturbed Tangent Track

2.10.1 Twist and Roll Section

Following the disappointing results in the dynamic curving test it was anticipated that similar difficulties would be encountered in the Twist and Roll tangent section on the PTT. This section has the same type of vertical perturbations as the dynamic curving section, except that they are more severe, but it does not have the curve or the periodic gauge widening. The results were as anticipated: at 22 mph (35.2 km/h) there was lateral contact of the drag link bolt heads against the sideframes and the test was terminated.

2.10.2 Pitch and Bounce Section

The Pitch and Bounce section of the PTT has the same magnitude of vertical disturbance as the Twist and Roll section. The difference is that the disturbances are in-phase on the two rails, thus exciting vertical and pitch motions of the car. From these tests it was clear that the effectiveness of the dampers was quite sufficient to control motions of the carbody. As speed was increased, however, there came a point when it became clear that solid contact was occurring at the bottom of the motions. It was not felt likely that the springs themselves were “bottoming out” but it was clear that some metal-to-metal contact was being made. The impact noise could be heard from wayside as the car passed. Testing was halted at 55 mph (88 km/h), when the impact noise became evident. After the trucks were removed from under the car, to be fitted to the empty car, it was discovered that the interference was between a projection on the underside of the traction beams and the top of the inner spring seat casting. The projection was there to provide for a surface against which the rubber lateral bump stop could act, provision could have been made in the lower casting to allow this to pass through. It was an oversight at the design stage that such provision was not made and that, therefore, these surfaces collided before contact was made with the vertical bump stops.

2.10.3 Yaw and Sway Section

Because it had been found on the Twist and Roll section that there was insufficient lateral clearance to permit the required motions, it was felt that testing on the Yaw and Sway section of the PTT would be futile as undesirable metal-to-metal contacts would almost certainly occur.

2.11 Dampers

As mentioned in section 2.5, the hydraulic suspension dampers, as initially installed, produced extremely small levels of damping. Prof. R. J. Anderson of the Mechanical Engineering Department at Queen’s University, Kingston, Ontario, conducted tests on the units to determine whether they met the manufacturer’s specifications.

The tests were conducted on a test rig available in the Mechanical Engineering department. It consisted of a stand on which the dampers could be mounted and the damper arm attached to a hydraulic actuator. The actuator was connected to a power supply, the controls of which were fed from a laptop computer. By this means the motions of the actuator could be tailored to meet any input requirement. For this test sinusoidal motions at various frequencies and amplitudes were used. It was found that the dampers produced very little damping effect.

The dampers were dismantled and it was found that screwed-in plugs, inserted in the main stems of the units, had in all cases worked completely out of their intended locations. This allowed free flow of the fluid within the units, thus preventing them from producing the desired damping effect.

The plugs were replaced and the units tested again. It was found that, although the damping action was now improved, there was a cut-off of the damping force at a level below the maximum desired for the freight car truck application. This was due to a spring-loaded relief valve inserted in the vanes of the unit for just that purpose.

In order to increase the maximum force that the units could produce, the relief valves were replaced by plugs that completely closed that flow path. When the units were tested again it was found that the maximum force had been increased, as desired, and that it did not increase excessively when higher speeds were applied. This was because of the effect of extraneous leakages within the unit (around the vanes, etc.), which acted as a natural limit to the maximum pressures (and, therefore, forces) that could be achieved.

3. Benefits

3.1 Fuel Savings

The fuel used to haul a train can be calculated from the formula

Fuel (gal./mi.) = $P/6283 + 27/S$

where P is the total drawbar pull in lb. and S is the speed in mph.

A 420 mi. (672 km) route containing a high degree of curvature and gradient, on the BC Rail system from Prince George to North Vancouver, was chosen to illustrate the savings accruing from the steered truck. The details of curvature, gradient and train speed were entered into a spreadsheet and the total drawbar pull and fuel consumption were then calculated. For a 286,000 lb. (130,000 kg) aluminum gondola car traveling this route, out loaded and home empty, the savings amounted to 3,129.45 US gal. (11,892 L) of fuel if the car operated 100,000 mi. (160,000 km). This represents an overall 21.7% saving on the total fuel usage for the car.

A similar analysis was also done for a 620 mi. (992 km) section of the Illinois Central Railroad's track between Galatia, Illinois, and Mobile, Alabama. This section of track has small gradients and shallow and infrequent curves. The 286,000 lb. (130,000 kg) car would save 1,152.35 US gal. (4,379 L) of fuel for every 100,000 mi. (160,000 km) traveled, out loaded and home empty, representing 20.7% of the total fuel used.

The fact that the percentage of fuel saved on the two routes was almost identical came as a surprise initially. It had been expected that the greater savings due to the steeper curves would produce a higher percentage savings on the BC Rail route. Further examination of the input data revealed the reason. On the more curvy route it was found that there were steeper and more frequent grades, which increased the total fuel consumed. As it happened, this increase in overall consumption almost matched the difference in the savings between the two routes and thus the percentage savings were almost equal. There are several interesting implications of this finding. Obviously, for less curvy routes with more grades than the values used here the percentage savings would be smaller. Also, for more curvy routes with fewer grades than the values used here the percentage savings would be greater. It is almost axiomatic, however, that areas with greater grades tend to have greater curvatures (in order to minimize the grade climbing) and that areas with few grades also tend to have few curves. For most scenarios, then, the values calculated here will probably be quite representative and so we can conclude that fuel savings of around 20% will be typical on most routes in North America.

The fuel savings can be expressed in another fashion, which can then be used to calculate fuel savings for different cars on these routes. The savings on the highly curved route amounts to 379 gal. (1,440 L) for every 1 million ton-mi. (1.45 million tonne-km) traveled. On the highly tangent route the savings are 139.7 gal. (530 L) for every 1 million ton-mi. (1.45 million tonne-km) traveled. A car that averages 80 tons (72.72 tonnes) and travels 50,000 mi. (80,000 km) a year will generate 4 million ton-mi. (5.82 tonne-km) in a year. Such a car would therefore save 1,516 gal./year (5,760 L/year) on the curvy route or 558.8 gal./year (2,123 L/year) on the less curvy route.

3.2 Savings Due to Reduced Mass

There are a number of ways of calculating the savings accruing from the reduced mass of the Resco truck. The first way, which was used by the AAR in the calculations summarized in section 3.4, is to simply calculate the reduction in fuel used as a result of hauling the lower weight. This assumes that the railroad would continue to load the same mass of commodity into the car.

The second method would be to assume that the railway would load more commodity into the car and then haul fewer trains. The benefit would be in the reduced costs associated with operating fewer trains. The assumption here is that no more commodity is available for hauling no matter what the capacity of the railway is for hauling it.

The third method is to assume that the railway would load more commodity into the car and generate more revenue for each train operated, effectively utilizing a larger capacity of the line. We will use this method here. If we assume that the railway charges US$15 for every ton (0.91 tonne) of commodity hauled through a distance of 1,000 mi. (1,600 km), and if we also assume that 4,800 lb. (2,182 kg) more commodity can be loaded into each car, then the increased revenue is US$15 x 2.4 x 50 = US$1,800 for each 100,000 mi. (160,000 km) traveled.

The increase in train capacity is 2% and the benefit is US$1,800/car per 100,000 mi. (160,000 km) traveled.

3.3 Drawbar Pull Reduction

For the same BC Rail route as was used above to calculate fuel savings, the maximum drawbar pull for a train of 286,000 lb. (130,000 kg) cars was 465,535.5 lb. (2,075,865 N) with standard trucks and 415,572 lb. (1,853,073 N) with the Resco steered frame trucks.

If the train length is limited by locomotive hauling capacity (rather than siding length, for instance) then **12% more cars could be hauled** by the same locomotive consist.

It should be noticed that the total force calculated here is greater than the maximum knuckle force allowable. Different railways use different strategies for overcoming this limitation. In some cases the trains are split at the areas of extremely high force requirement, in other cases "pusher" locomotives are used. For the above route the train is split at the highest pull location. For the rest of the route, where the train remains connected, the maximum drawbar forces are 246,454 lb. (1,098,960 N) with standard trucks and 214,208 lb. (955,173 N) with the steered trucks. This represents a potential increase in train length of 15%. As the other area showed a smaller gain it is that value that governs.

On the alternate route the maximum drawbar pull values were 331,280.7 lb. (1,477,210 N) for the three-piece trucks and 315,970.4 lb. (1,408,941 N) for the steered ones. This corresponds to a **4.85% increase in the length of the train** using the same locomotive consist. In the maximum force section of track, in this case, helper locomotives were used. In the areas where no helpers were required the maximums were 194,844 lb. (868,827 N) with standard trucks and 171,111 lb. (763,000 N) with steered trucks. This represents a potential train length increase of 13.9% so, again, the section where helpers are required governs.

If we assume that the benefit to the railway of increasing its train capacity by operating longer trains is similar to the benefit of increased capacity due to lighter trucks, we see that, on the curvy route there will be a benefit of US$10,800 per 100,000 mi. (160,000 km) traveled and on the less curvy route it will be US$4,365 per 100,000 mi. (160,000 km) traveled. This, of course, ignores the capital costs of the extra cars, but it is a useful first approximation to the potential benefit of the increased train length due to reduced train resistance.

3.4 AAR Economic Benefit Calculations

As part of its work on the bulk commodity test program, the AAR created a set of operational benefit models for the various criteria measured during that testing. These were published in November 2001 [3]. That paper contains a series of plots showing the dollar benefit to a railway from improvements in each of the test values measured. The benefits are given for a 286,000 lb. (130,000 kg) car and a 315,000 lb. (143,182 kg) car and for two routes, against the percentage of improvement relative to a similar car fitted with baseline trucks. If the percentage improvement of the Resco steered truck in each of the categories is calculated, then the dollar benefit to a railway can be determined from the charts in that paper. Route 1 contains a relatively high percentage of curves and route 2 contains only a moderate number of curves.

The results of this are seen in Figure 39, using the values for a 286,000 lb. (130,000 kg) car taken from the paper. On route 1 the saving is US$3,186 per 100,000 mi. (160,000 km) and on route 2 it is US$1,686. It can be seen that the economic benefits were divided into six separate categories. The wheel and rail wear, and the fuel savings on curve and tangent track, were all calculated from the reduction in rolling resistance measured for the truck. The tie damage was calculated from the reduction in lateral force on curves and the savings due to weight reduction was calculated from the reduction in fuel used to move the car.

From Figure 39 we see that the AAR calculation gives a value of the fuel saved as US$1,270 on the highly curved route and US$550 on the less curved route. This agrees very well, in terms of the ratio of the savings on the two routes, with the ratio obtained in section 3.1. The absolute value, however, seems rather low. The savings calculated in section 3.1 were 3,129.45 gal. and 1,152.35 gal., respectively. Depending upon what the cost of fuel is to the railway this could translate to 2 to 3 times the AAR estimate, approximately US$1,100 to US$3,800.

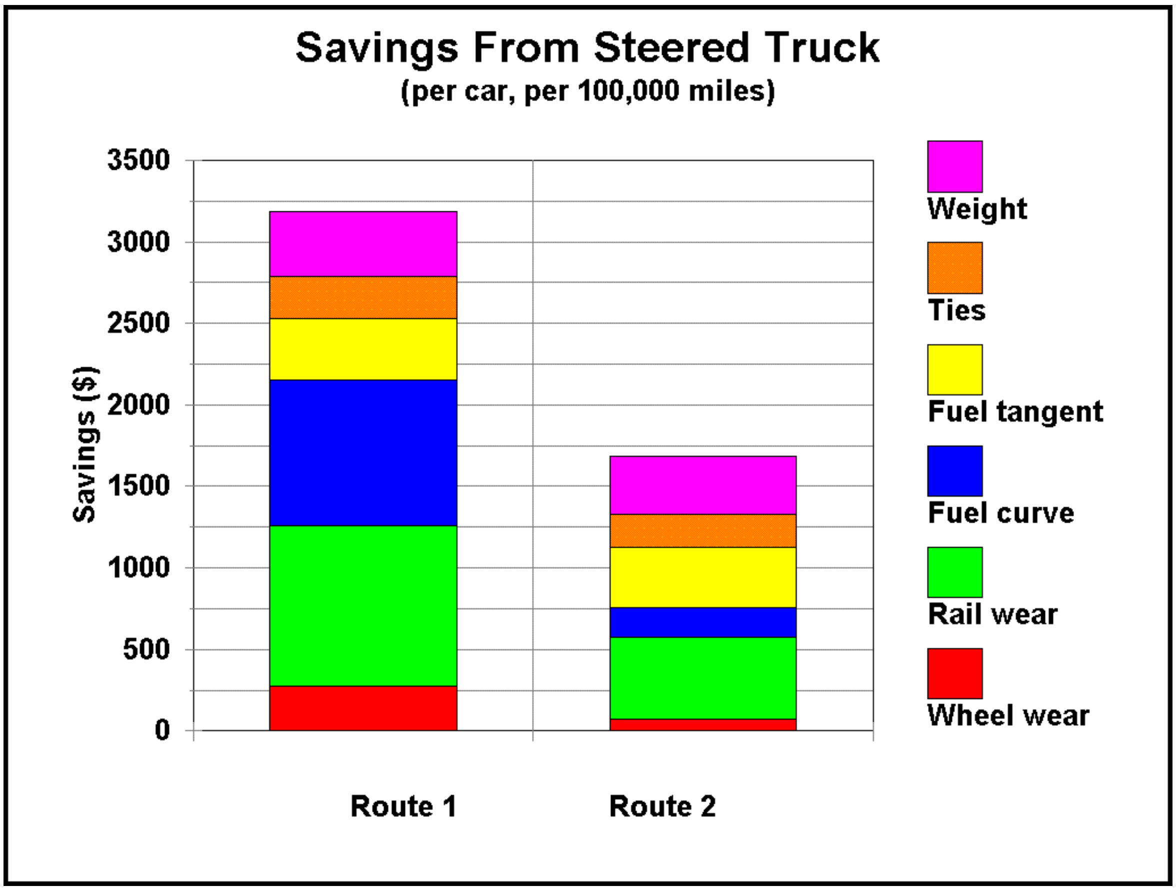

Figure 39. Savings for 286,000 lb. Car

One reason for the difference in the estimates of savings is that the estimate in section 3.1 uses the rolling resistance values measured during the tests, whereas the AAR estimate assumes that there would be a reduction in the average adhesion level, when running on service track, compared with that on the test track, and that the rolling resistance would be reduced as a result. The rolling resistance, however, is contributed to by many factors, including angle of attack, flange lateral force, wheel/rail creep coefficient, etc. It seems unlikely that the rolling resistance will reduce directly in proportion to the adhesion at the head of the rail.

The AAR's calculation of the value of the weight saving was done by assuming that the same amount of commodity would be carried in the car and then determining the fuel that would be saved from moving the lighter train. It seems more likely that in most bulk commodity operations, the amount of commodity carried would be increased so that the maximum allowable axle load for the car was met. The weight reduction would therefore be canceled by more commodity being carried, with a consequent increase in revenue for no increase in expense, as calculated in section 3.2.

If we assume that half the mileage is empty (non revenue) and half is loaded, then for every 1,000 lb. (454.5 kg) of extra commodity carried, the revenue traffic will increase by 25,000 ton-mi. (36,364 tonne-km) for each 100,000 mi. (160,000 km) of service by the car. If the railway charges US$15 per 1,000 ton-mi. (1,454.5 tonne-km) to carry the commodity, then the increased revenue due to the reduced weight of the trucks is US$375 per 1,000 lb. (454.5 kg) of weight reduction. For the steered frame truck the weight reduction is between 3,600 and 4,800 lb. (1,636 and 2,182 kg), so the increased revenue is between US$1,350 and US$1,800 for every 100,000 mi. (160,000 km) traveled. By comparison, the reduction in fuel costs due to reduced weight, calculated by the AAR, is US$400 for route 1 and US$360 for route 2.

Using the above two figures for the fuel savings and weight savings, and using the AAR figures for the other savings, without accounting for the increased train length benefit, we get total savings of between US$6,016 and US$7,766 per 100,000 mi. (160,000 km) for route 1 and between US$3,676 and US$4,876 per 100,000 mi. (160,000 km) for route 2.

3.5 Safety Against Derailment

It has been shown that the Resco steered truck offers substantially reduced L/V ratios in curving compared with the standard three-piece truck. The reduced lateral forces result in reduced track damage and the AAR, as discussed in section 3.4, has calculated the value of this. There is a further economic benefit, however, because reduced derailments result in reduced costs to the railway and improved delivery performance to the customer. In order to calculate the value of that economic benefit we need to know the number of derailments and the costs of those derailments, and the reduction in frequency of derailments. Similarly, the improved delivery performance could be calculated and it would be possible to estimate the added market penetration that would be possible as a result.

Let us make an estimate here for the value of reduced derailments. North American freight traffic volume is approximately 25 to 30 billion ton-mi. (36.4 to 43.6 billion tonne-km) annually. Let us assume that there is one major derailment in every 10,000,000,000 ton-mi. (14,544,000,000 tonne-km) and that the average cost of each such derailment is US$2,000,000. If the average mass of the freight car is 82.5 tons (75 tonnes) then it produces 8,250,000 ton-mi. (12,000,000 tonne-km) for every

100,000 mi. (160,000 km) traveled. The cost of derailments is thus US$1,650 per car per 100,000 mi. (160,000 km) traveled. If the rate of derailment can be reduced to 1 per 40,000,000,000 ton-mi. (58,176,000,000 tonne-km), then the saving to the railway will be US$1,230 per 100,000 mi. (160,000 km) traveled.

3.6 Higher Empty Speeds

The added stability of the Resco truck allows empty trains to be returned faster, thus improving their productivity. This benefit can be estimated as follows. Let us assume that the loaded speed of a train fitted with standard trucks is 50 mph (80 km/h) and the empty speed is 40 mph (64 km/h). Let us further assume that, if the train is instead fitted with Resco trucks, the loaded speed is unchanged because the locomotive horsepower limits it, but that the return speed is increased to 60 mph (96 km/h). The standard train will have an average speed of 45 mph (72 km/h) and the Resco-fitted train will have an average speed of 55 mph (88 km/h). This represents an increase of productivity of 22.2% for the same capital equipment and the same number of crew hours. At US$900 per car for every 1% increase in productivity for every 100,000 mi. (160,000 km) traveled this translates to US$20,000 per car per 100,000 mi. (160,000 km).

3.7 Summary of Economic Benefits

Table 9 summarizes the economic benefits of the Resco Steered Frame truck to an operating railway. The first three items use the figures calculated in sections 3.1, 3.2 and 3.3, the next three items use figures taken from the AAR calculations shown in Figure 38. The last two items are as estimated in sections 3.5 and 3.6.

		Route 1	Route 2
1	Truck Mass	US$1,800	US$1,800
2	Rolling Resistance	US$4,381	US$1,613
3	Drawbar Pull	US$10,800	US$4,365
4	Track	US$250	US$200
5	Rail Wear	US$1,000	US$500
6	Wheel Wear	US$250	US$100
7	Reduced Derailments	US$1,230	US$1,230
8	Higher Empty Speeds	US$20,000	US$20,000
	Total	US$39,711	US$29,808

Table 9. Summary of Economic Benefits of the Resco Truck for Each 100,000 mi. (160,000 km) Traveled

Clearly there are major differences in the cost benefits depending upon how a railway decides to calculate them. This is partly determined by the business philosophy of the organization involved. What is clear is that, even with the most pessimistic method of calculation, and even for railways having few curves, the steered truck still generates very substantial payback. If we only accept the figures in rows 2, 4, 5 an 6, and ignore the benefits that would require a more aggressive marketing and/or operational strategy, the totals still come to US$5,881 for route 1 and US$2,413 for route 2.

What is equally clear is that the railway that is willing to aggressively seek new business based on its increased capacity and improved service with the new trucks can reap very attractive rewards.

3.8 Present Value – Return on Investment

It is an easy matter to show that if a capital item is amortized over a period of *n* months, and produces a return of $M/month, and if the desired rate of return on investment (ROI) is X%/year (X/12 = x%/month) then the present value, P, for the item is given by the formula

$$P = M\{(1+x)^n - 1\}/x(1+x)^n$$

If the car travels 50,000 mi./year (80,000 km/year) and produces a return of $3,600 per 100,000 mi. (160,000 km) then M = $150/month. If the amortization period is 120 months and the required ROI is 30% per year (2.5%/month) then P = $5,690. This is the premium that could be paid for the trucks, over the price of current three-piece trucks, and the investment would be retired after ten years with a ROI of 30%.

Similarly, if the car traveled 100,000 mi. (160,000 km) per year and produced a return of $7,200 per 100,000 mi. (160,000 km) then M = $600 and P = $22,760. Clearly the net present value of the truck is highly dependent upon the routes and the intensity with which it is used.

The calculation can be done assuming a value for P (the premium over a standard pair of trucks) and determining the ROI. If P = $10,000, M = $300 and n = 60 months, then X = 26.2% per year. The truck will have repaid its premium in 60 months and produced a ROI of 26.2% per year at the same time.

In every case, of course, if the truck outlasts its assumed amortization period then the returns will increase.

3.9 Manufacturing and the Actual Premium Price

A short discussion of the flexibility of the design concept to various techniques of manufacture is appropriate at this point. The prototype trucks were built using a hybrid "cast-fabricated" structure. Wherever there were complex shapes and/or difficult stress concentrations, the structure was cast and then the cast pieces were welded into fabricated sections of simple geometry, the welds being located in areas of relatively low stress. Because of this design technique the trucks can be built quite readily in an extremely small, low-tech machine and fabricating shop. This demonstrated the feasibility and attractiveness of this type of construction, and was essential in keeping the costs of prototype construction within reasonable bounds. It also showed that the technique creates the situation where a multitude of manufacturing facilities can potentially provide serviceable railway trucks to the North American industry. This can greatly improve the potential for competitive pricing of the trucks. On the other hand, if a one-piece cast sideframe is required, due to a customer policy or specification concerning such things, nothing prevents such a structure being utilized and no weight penalty would be accrued.

The data presented in this report suggests that US$1,800/year is a very conservative estimate for the overall annual revenue/expense savings produced by the Resco truck, depending on the routes upon which it is operated. Many operations will generate several times this. This amount of savings would justify an investment of US$5,690 over the price of a standard truck and produce a ROI of 30%. The actual increased cost over that of a standard three-piece truck will, of course, depend very much upon the quantities of trucks sold, the manufacturing techniques used, and other factors. Given the reduced weight of the truck it is felt likely that, despite the added machining required by its nature, the total increase of cost, which would reflect a price increase of US$5,000 to US$6,000, is quite attainable within a reasonable period from its introduction. From a marketing point of view, however, it might be necessary to absorb much of the increased cost of the truck initially, meaning that a substantial investment might be needed in order to establish the truck in the marketplace.

4. Discussion of Results

4.1 High-Speed (Empty) and Perturbed Track Test Results

A general discussion of the test results follows in section 4.2, but it is felt that the high-speed (empty car) results warrant separate discussion because of the design implications that they produce.

The criterion most frequently used in AAR testing of freight cars, as an indication of whether truck hunting is taking place, is to record the carbody lateral accelerations. If the accelerations exceed 0.27g (rms) over the test section, that is taken as an indication that truck hunting has been established. This criterion, of course, assumes that the suspension system is working properly at the time of the test. In this case, the criterion for lateral acceleration was exceeded but visual observation of the truck showed that there was very little truck motion, even when the carbody motions were exceeding the limit. The same is true of the wheel load limit. Little or no flange contact or wheelset dynamic activity was observed even while the wheel vertical load was periodically reducing to within 10% of its static value. Indeed, it was observed at the time that, even when a substantial lateral excursion of the wheelset was created due to some track perturbation, it was immediately damped out and no periodic motions resulted.

In the case of the original high-speed empty testing, with ineffective dampers, the minimum wheel loads did not show the same reduction at speed as when the rebuilt dampers were re-installed (see Figures 26 and 30). It might be inferred that the dampers were therefore the cause of the phenomenon. In a way this is undeniably true, but not quite as it appears on the surface.

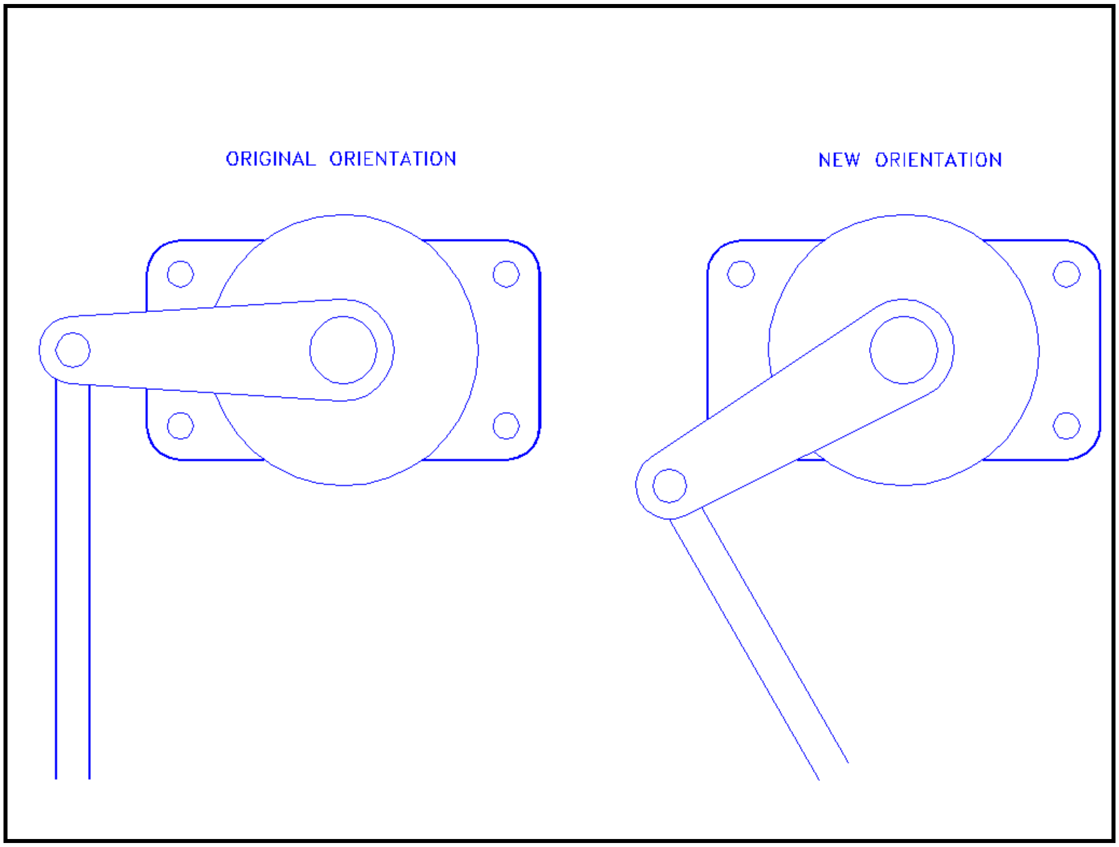

Figure 40. Damper Re-orientation

The rebuilt dampers were first installed under the loaded car, because that was the car under which the trucks were mounted at the time. When the car was first operated on the track, it appeared that the dampers had produced the required effect in the vertical and roll directions but that there was insufficient damping in the yaw direction. A quick modification was desired in order to correct this situation so that high-speed testing could continue. It was decided that it would be an easy matter to re-orient the damper arms so that they still provided nearly as much damping in the vertical and roll directions and also contributed something in the lateral direction. The re-orientation is shown in Figure 40. This arrangement was tested and found to control the yaw and sway motions, so high-speed testing was continued in that configuration.

When the change noted above was made, there were two choices available for the relative orientation of the dampers on either side of the car, as shown in Figure 41.

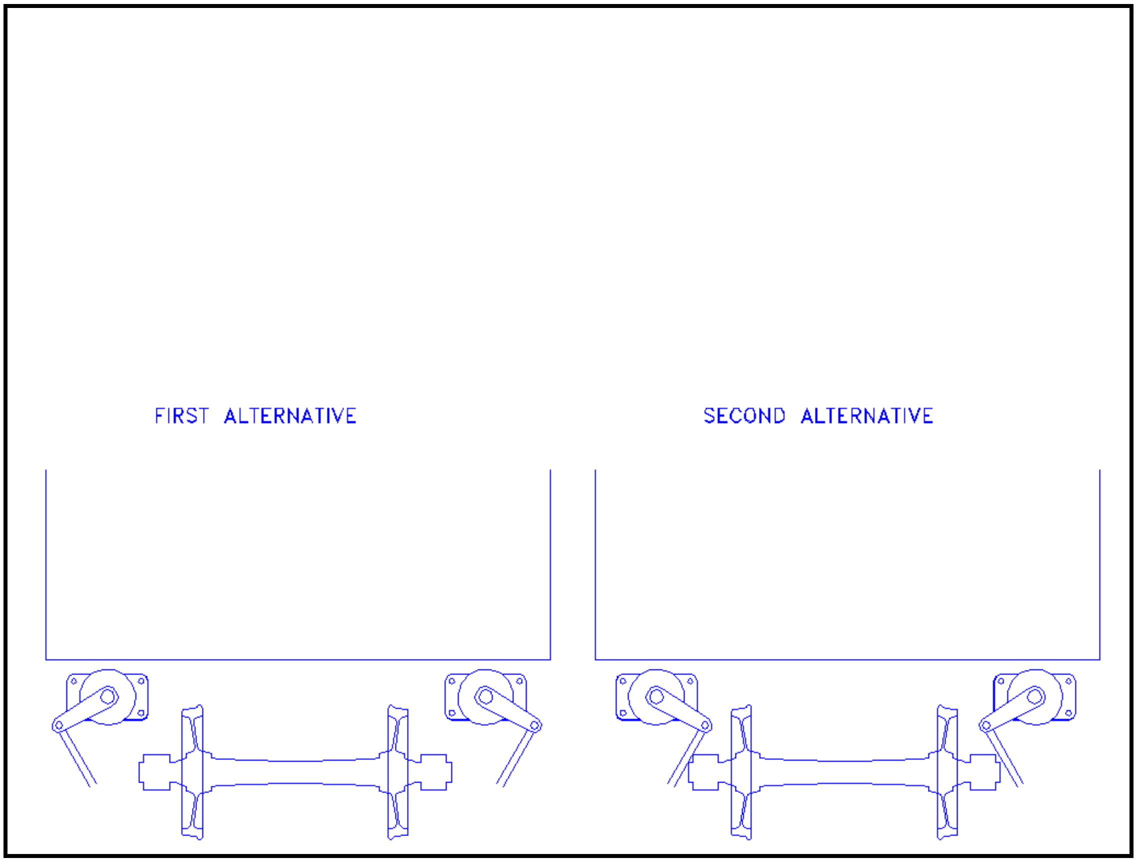

Figure 41. Alternative Orientations

The first of the two alternatives shown was the one chosen. Both would have been equally effective in controlling the yaw motions but the one chosen had the undesirable effect of forcing Lower Centre Roll (LCR) motions around a centre that was far below the spring location. The normal position for the roll centre in LCR motions is close to the spring height, or between there and the top of rail. With the roll centre in that location, the lateral movement at the spring height is usually quite small. By forcing the roll centre to be well below the rail head, two detrimental factors occurred. First, the damping for motions around that centre became extremely low – the motions simply created swinging of the links without any movement at the damper arm. That is what caused the roll centre to be forced down, of course. Second, the induced roll motions caused substantial lateral motions at the spring height. There is limited clearance in the prototype trucks between the ends of the traction beam (above the springs) and the inside of the sideframes, so metal-to-metal contact began to occur there and this became increasingly severe as the speed was increased. A further negative factor in this situation is that, when the roll centre was forced downwards (away from the carbody centre of gravity), the overall roll inertia for that mode was increased so the modal frequency was reduced. This meant that the mode became excited at a lower speed than it otherwise would have. It was not realized, of course, that this effect had inadvertently been produced until the dynamic curving tests, and subsequently the twist and roll tests, were performed. By that time it was too late in the program to rectify the situation and re-test.

If the alternative orientation for the damper arms had been chosen, this would have had the effect of increasing the height of the LCR roll centre and lowering that of the Upper Centre Roll (UCR) mode. The excessive LCR motions and bump stop contacts would not then have occurred. The UCR mode is one that is rarely excited from track inputs and is thus not as great a concern as the LCR mode. The relatively positive results from the Pitch and Bounce tests (up to the point where metal-to-metal contact was made) provide an indication that the damping level is sufficient as long as the geometry doesn't force an exaggerated LCR mode and as long as the mechanical clearances are adequate.

In the case of the empty car tests, the effect of the damper orientation on the car motions was even greater than in the case of the loaded car. In the empty case the springs were near the top of their extension, the car being empty, so the damper arms were near the bottom of their stroke. This was exaggerated by the fact that the damper link length did not have sufficient adjustment to allow for the fact that, when the orientation was changed, a greater link length was required. As a result, in the empty condition, the dampers were very near the end of their stroke and thus provided a very stiff constraint, forcing the body into lateral (LCR) motions whenever the end of the stroke was reached.

4.2 General Discussion of Test Results

The test techniques used at Pueblo to determine the performance of the steered truck were developed at the test centre over a number of years. A great deal of confidence can be placed in the results as a consequence of this experience. The testing at NRC Ottawa was much simpler, but the results match very well with those obtained at Pueblo. From this point of view these results can also be viewed with great confidence.

The rolling resistance measurements show a degree of fuel economy that is unprecedented in the railway industry. These measurements have been repeated on several occasions, including tests on widely separated days, and the results have been extremely consistent. There can be no doubt that the steered truck has produced a fuel economy that is currently unmatched.

The reduced lateral rail forces in curves were expected and have been used to calculate savings in track maintenance. The reduced vertical shock and vibration is an added benefit that was not included in the calculations of benefit. This would have most significance in terms of ballast tamping, railhead stresses and structures such as bridges.

5. Conclusions and Recommendations

The Resco Steered Frame Freight Car Truck has demonstrated a highly desirable set of performance results from the testing conducted at BC Rail, NRC Ottawa and TTCI Pueblo.

It can be concluded that the Resco Steered Frame Freight Car Truck has sufficiently greater economic benefits when compared with standard three-piece trucks, or other premium trucks, that a substantial premium price can be justified over either of those alternatives.

There remain some minor adjustments to be made to enable the truck to operate satisfactorily throughout the range of perturbed track conditions required by the AAR. These modifications involve primarily the provision for more motions at the suspension. Vertical interferences were discovered at the traction beam and the shear frame, and at the traction beam and the inner spring mounts. Lateral interference was seen between the traction beam and the sideframes. The damper orientation needs to be reversed, as discussed in section 4.1. The roller interface between the carbody underside and the top of the outer springs, while effective, had a tendency for migration of the rollers out of the loaded area. A new design with captive rollers is required.

Once the required design adjustments have been made, the perturbed track testing should be repeated and a program of endurance testing embarked upon.

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