MTMC REPORT TR 75-5

TECHNICAL REPORT

CARGO RESTRAINT CRITERIA FOR RAIL TRANSPORTATION

MAY 1978

MILITARY TRAFFIC MANAGEMENT COMMAND
TRANSPORTATION ENGINEERING AGENCY
NEWPORT NEWS, VIRGINIA 23606

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Prepared by

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ABSTRACT

This report discusses criteria for cargo restraint and cargo tiedown for the forces resulting from the rail transportation environment. The report covers methods for calculating the restraint forces for a broad range of cargo railcar combinations. Use of these methods will improve accuracy of restraint criteria and, in turn, will improve reliability of cargo restraints and reduce costs consequent to restraint over design.

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I. INTRODUCTION

Railcar cargos need to be secured, anchored, or tied to the railcar to prevent uncontrolled relative motion between the cargo and the railcar. During transit, vertical dynamic forces in the tiedowns are developed by the railcar wheels traversing irregular and elastic tracks. Principal longitudinal input forces are developed both by train coupler shock, which causes incremental train tension and compression during passage over vertical curves, and by braking. Lateral dynamic forces are generated from lack of levelness from one track to the other and from the combination of coupler tension and railcar truck action as a railcar traverses a horizontal curve.

The forces that have received the most attention for cargo tiedown are forces generated through the coupler, the car body, and the cargo during rail switching or railcar impact operations. Tiedown forces consequent to railcar impact are initially in the longitudinal direction at the coupler and in the longitudinal, vertical, and lateral directions at the cargo tiedown; this is due to misalignment of the resisting forces that result in force couples in the three principal directions. Tiedown shock forces generated from railcar impact are usually of the greatest magnitude of all operational forces; hence, they are often the only forces considered for tiedown design. Also, impact forces are the easiest to simulate for test application and are comparatively simple to analyze for criteria development. Particular comprehensive engineering analysis of all forces is required for high-value, sensitive, or hazardous cargos to reduce the risks of cargo damage or accident caused by lack of precision in cargo tiedown criteria. Current practice to insure adequacy of cargo tiedown for rail varies from no tiedown provision for cargos that need none, to elaborate and expensive test and evaluation programs for cargos that require a high degree of tiedown reliability. For most of the cargos that require tiedown attention, the blocking and bracing or tiedown is designed to withstand a proof test. Considerable blocking and tiedown experience has been acquired. This experience can be used to select the tiedown or blocking arrangements that will be sufficiently strong to pass standard impact tests and to identify areas of marginal strength. Standard railcar impact tests are conducted; then tiedown members or configurations that fail are strengthened or reconfigured until the system satisfactorily goes through the tests and receives approval based on visual inspection. To date, this procedure has been adequate and has produced an overall good record of cargo tiedowns at minimum cost.

The current method of controlling cargo tiedown, even though it is based entirely on experience and practical judgment, has certain basic mechanical factors built into the approval process. These mechanical factors are changing, and the restraint criteria must be updated. Impacts are made with 50-ton cars limited to 169,000-pound loads on the rail. The trend in railcar purchase in the last 10 years has been toward 70- and 100-ton railcars that are limited to 220,000- and 263,000-pound loads on the rail. Heavier weights might produce higher restraint criteria. Another trend in railcar acquisition has been to buy high-capacity, long-travel, draft-gear, and cushioned-underframe cars. These cars provide for greatly reduced forces to the cargo tiedown. Cargos tied down on trailers and shipped via rail piggyback also have reduced tiedown forces during impact due to the long-travel cushion built into the device connecting the trailer to the railcar. Greater cushioning usually produces lower restraint forces. The present method for controlling tiedown design provides neither for changes in the rail system, such as improvements in the railcar carrying the cargo, nor for introducing new heavyweight railcars that will impact into the railcar with the new tiedown design during switching operations. Cargo tiedowns designed and tested 10 or more years ago are overdesigned for cushioned-car use. Other old tiedown designs are inadequate to resist the impact forces of a 100-ton hoppercar. A design method, analysis, and test procedure must be developed for railcar cargo tiedowns different from the experience and go-no-go test check method. This new procedure should be flexible, to allow for system changes and improvements, without making obsolete much of the previous design analysis and test work investment.

The first requirement of tiedown design is restraint criteria to establish external forces and external loadings. The force criteria must include the mechanic variables that fix the size, direction, and characteristics of tiedown force. With sufficient expression of the mechanics of the system, the tiedown criteria can be formed to accommodate most system changes without scrapping previous tiedown design and test efforts. Also, tiedown criteria based on engineering mechanics should produce more balanced and economical tiedown designs because go-no-go tests check the failing load of the weakest tiedown member but do not indicate overdesigned tiedown members.

Cargos vary considerably in the degree of risk of tiedown failure that the shipper is willing to take. Most current tiedown criteria are expressed as the velocity of impact between two railcars undergoing rail tests.

The risk is accounted for by setting an 8-mile-per-hour impact for most cargos and a 10-mile-per-hour and above impact for cargos needing maximum protection or tiedown strength. This approach serves a purpose in that extra tiedown strength is provided when needed. This procedure allows neither for checks against tiedown members stronger than the weakest nor for strengthening tiedown components without redesign and retest.

Many organizations, military and industrial, perform tiedown design and rail impact tests for military and commercial cargos. Little uniformity exists in the tiedown criteria. They are not specific or precise, and different tiedown forces might result when applying the same broad criteria. Tiedown criteria must be stated in tiedown forces, or in methods to get tiedown forces, to obtain uniformity and consistency in basic design values.

Military Traffic Management Command (MTMC) recognized both the urgency and need for tiedown criteria. A program was set up to establish tiedown criteria by analysis and research and by utilizing existing approved test results. The program was planned to include study of tiedowns for all Army cargos because rail shippers are responsible for developing and applying cargo tiedowns. Most tiedown configurations have to be approved by the railroads prior to shipment. The railroads frequently require analysis, design calculations, proof, and/or tests for tiedowns of particular cargos prior to shipment. Either the railroad or the Association of American Railroads provides the inspection, approval, and general advice, but not the funds, to conduct the engineering design or criteria work.

Military Traffic Management Command (MTMC) conducted the library research and analysis of tiedown mechanics and practice from July 1974 to June 1975. Areas of importance that appeared to lack attention were emphasized in this study; they were engineering mechanics, balanced tiedown designs, costs, and risks.

II. OBJECTIVE

To develop cargo restraint criteria for analysis, design, and testing of blocking and tiedown structures used to affix cargos to railcars.

III. CONCLUSIONS

It is concluded that:

1. Restraint criteria, the forces required to design cargo restraints, can be determined with proper application of test experience and engineering mechanics.

2. Four methods for calculating restraint criteria--namely, coupler force, empirical, classical, and energy--all produce usable, accurate restraint forces.

3. No one method of calculating restraint criteria is perferred over the others for all cargo railcar systems. Each method has mechanical and structural considerations that will dictate its usage for calculating restraint criteria.

4. The flexibility of the cargo restraint structure determines, in part, the forces imparted to the restraint and the cargo. The physical arrangement of the restraint must be either known or specified to establish restraint criteria.

5. Improved accuracy for restraint criteria will improve both safety and reliability by providing adequate restraint protection, and will effect cost savings by eliminating overdesign in all or part of the restraint structure.

IV. ANALYSES

In order to give specific criteria, it is necessary to describe the high intensity shock occurring at the railcar floor or platforms on which the cargo is supported. These shock criteria are described for the longitudinal direction as ideal terminal peak saw-tooth pulses having a peak acceleration of 40g in logitudinal, 16g in vertical, and 8g in lateral directions with the pulse durations of 11 milliseconds. Design criteria shall require this shock pulse to be repeated three times in each direction. For reusable components, where fatigue effects are pertinent, design criteria will be 1,000 shocks of 40g amplitude and a pulse duration of 6 milliseconds.

Specific criteria for a range of frequencies in the three principal directions are:

Frequency cps	Peak Acceleration in g's: Longitudinal	Vertical	Lateral
2	10	4	2
10	50	20	10
20	63	45	20
40	150	85	40
60	200	130	50
80	300	180	50
100	400	250	55

Specific criteria represent the maximum shock values to be expected. Contributing to these maximums are maximum expected impact speeds, maximum unfavorable cargo-railcar weight combinations, maximum mismatch of component elasticities and no damping. Impact speed is independent of design, but all other factors can be minimized by mechanical designs to produce forces and accelerations substantially lower in severity than the specific criteria listed. Some Army cargos are sufficiently strong, or of such low value, as to be immune to the severities of maximum shock possibilities and require no attention in design of restraints to reduce shocks below the maximum possible.

Most Army cargos of transportability interest are those that require economic restraint structures with the greatest possible shock attenuation. These cargos require restraints that are analyzed and designed to result in the lowest possible forces acting on the restraint and on the cargo. Mechanical analysis for a specific cargo-restraint configuration should produce both more economic and more reliable restraint design than one resulting from application of the specific criteria.

There are several instances in which the Army would have been in a more defensible position if the basic mechanics criteria formula of this report had been available and applied. The accident report of the National Transportation Safety Board, 24 May 1973, describing the investigation of the freight train-munitions explosion at Benson, Arizona, discussed blocking and bracing failures. One of the munitions cars, MKT 539L, was cut from the train because of a broken floor shortly before the explosion of the 22 munition cars. Inspection of this car showed that the "bracing in the car had shifted and was loose at one end wall," and that "the blocking and bracing were damaged and loosened. Some of the banding on the pallets was loosened and others had shifted. Several of the bombs had contacted one another." The report states that "the Navy designed the blocking and bracing of munitions to withstand impacts of 8 miles per hour. Either this standard was not met in MKT 539L or the car was subjected to a potentially damaging impact which was not reported." The evidence from the damaged blocking and bracing was strong, but a strong analysis was not forthcoming. Had the restraint criteria been stated in quantities of force and frequency instead of in impact velocity, it could be stated that the design impact force was 100,000 pounds, for instance, and this force was exceeded. This engineering mechanics approach would give insight for restraint design and restraint application corrective action to improve safety and relate the cargo restraints in the inspected car to those in the exploded munition cars.

The US Army Materiel Command (AMC) Ammunition Center Report No. [illegible], May 19[illegible], on the JK-[illegible] internal cargo restraint system trial shipment, states: "The 105-mm portion of the JK-3 restraint system test load failed to successfully complete the trial shipment. Two of the hold-down straps failed." This in-service restraint failure occurred on the first test shipment, which concluded a comprehensive testing program in which the restraints withstood severe impacts in excess of 10 miles per hour with a 210,000-pound hammer car. Restraint fractures and cargo displacement observed after service-induced failure were different from those observed after test-induced failure, suggesting improper test simulation. It appears that the holddown straps might have failed due to intransit resonance and fatigue or to some other condition that was not simulated during the testing program. It would have been easier and more economical to pinpoint restraint overloads during the first test shipment if the original criteria for the ammunition restraint had been stated in terms of force and frequency for a continuous range of cargo weights and springing combinations. As it turned out, the trial shipment test failures were unexplained, and this failure terminated the comprehensive testing program for development of restraint criteria. The credibility of criteria developed by this program and by the rail and highway test procedures was greatly weakened by a failure of correlation in use of cargo restraint between the test-simulated environment and the in-service environment. It is probable that, with the mechanics approach, either the program would have been successful through the trial shipment, or any basic mechanics difficulty would have been detected early enough in the test program to minimize any investment in criteria and restraint development programs that were less than successful.

The Army let several contracts in 1974 and 1975 to develop removable restraint systems for shipment of ammunition in commercial containers that have no built-in restraint systems. According to the criteria given the contractors, the devices must survive a particular rail impact test conducted by the Army. The result has been that the contractors, using the contract funds, design and fabricate the restraint system based on their experience. The Army then tests the various systems and discovers that they are not strong enough.

If the restraint criteria expressed in formulas contained in this report were used in the requests for bids for contracts, the program would be less costly to the Army, and the chance for a successful development would be greatly increased. Before incurring prototype and testing costs, the Army could require an analysis showing that the proposed device is calculated to withstand the criteria limits expressed in the formulas. Sorting, ranking, and many comparisons of proposed devices could be completed by mechanical analysis. Contract prototype construction and Army testing would then be performed only for those devices disclosed by analysis to be the most promising for project cost reduction.

Designers have the most immediate application of restraint criteria expressed in formula form. They must start with external forces, and the criteria formulas give these external forces as well as frequencies and directions necessary to base a restraint or fitting design. Designers would also apply the criteria to provide a check on the forces by using one or more of the methods. The restraint criteria probably are the most important factors with which the designer has to deal, and ones over which he or she has no control. By applying the formulas and using the results to evaluate feedback, the designer has a sound basis for entire design. As basic changes are made in the design process--for instance, from chocked to tied-down cargos--and in the consideration of other loading generators, the formulas can be reapplied to predict the applicable input forces. Early in the design process, designers can apply the formulas to calculate component frequencies in order to stay away from unwanted resonances.

Transportation managers will apply the restraint criteria indirectly, but the formulas should prove useful to them. Transportation managers can request formulas and analyses for cargo restraints suspected to be either underdesigned or overdesigned. They will be less dependent on the designers and will have additional specific input for cargo restraint decisions.

Transportation planners would apply the rail restraint criteria by requesting that an analysis be made when major restraint changes are planned. Resonances and amplifications normally are not considered in the planning stages, but planners must provide resonance-free transportation for some selected cargos. The restraint criteria, as given, will allow for predicting the restraint conditions that would be highly susceptible to resonance. Planners can apply analysis, based on the criteria, to increase the value of their plans.

The greatest merit of restraint criteria is that it provides a basis on which to analyze the mechanics of railcar impact. Many times, data can be obtained or comparisons made from analysis alone, without physical tests. Costs of testing programs in this shock and vibration or mechanics area run from $25,000 to $250,000. Costs of analysis, which often is sufficient to obtain the information required, run from $10,000 to $100,000. The biggest cost-saving factor is that analysis does not require instrumentation, prototypes, testing, and data reduction.

The merit in expressing the restraint criteria by formula is that the analysis can be readily expanded, contracted, or changed without losing the benefit of the work done. Testing is entirely inflexible because test results from one cargo indicate little about how another cargo will respond with regard to weight, spring rate, and amplification.

An analysis based on restraint criteria provides a better library of information than do test data because it is not dependent on instrument calibrations, bands of frequency, or instrument improvements. Basic analysis never becomes obsolete because the mathematics never change. A principal merit for restraint criteria in formula form and for follow-on analysis is that the criteria formula provides information to designers in direct, usable form. Forces are the common denominator for communication among criteria preparers, designers, and analysts.

Many cargo restraints require tests regardless of the analysis performed. When tests are mandatory, the application of restraint criteria and analysis will be beneficial to the test program. Formulas can be used, to reduce test costs, and to extend test results for various other spring-mass arrangements. Analysis will reduce the experimentation necessary to find extreme conditions, and will hasten the pretest arrangements. It will give a better understanding of test results and assist in identifying and classifying component failures. Normally, test failures require additional action to remedy the weakness. Criteria and analysis will expedite this process when a test program is preceded by analysis and followed by corrections to the system based on analysis and test findings. The criteria analysis and test approach has the added advantage of being more scientific than the current practice of test and fix; the scientific approach is a process of calculation, measurement, and correction.

Some of the four methods of determining restraint criteria appear more applicable for particular types of cargo and railcars than others. The following display gives a general indication of which criteria methods appear to be most suited to different types of cargos. For the listing, G is good, F is fair, and P is poor.

	Criteria Method			
Cargo Types	Coupler Force	Empirical	Classical	Energy
1. General, boxed	F	F	F	F
2. General, rigid	P	P	F	F
3. Vehicles	P	F	F	G
4. Ammunition	P	F	G	G
5. Piggyback	P	P	G	G
6. Bulk	P	G	P	P
7. Equipment containers	P	P	G	G

Cargo Types	Criteria Method			
	Coupler Force	Empirical	Classical	Energy
8. Tanks, bulldozers	G	P	F	F
9. Nuclear waste	G	P	F	F
10. Missile systems	P	P	G	F
11. Communication systems	P	P	G	F

Many benefits are possible from application of the restraint criteria presented. Testing costs can be greatly reduced by use of this more precise, straightforward approach. Design costs are held to a minimum because criteria forces are supplied in a form that can be readily applied to design. Also, criteria forces expressed in formula form reduce the effort required for design optimization or for major design changes. Better and more comprehensive criteria for designers will contribute to simpler restraint designs and should greatly reduce costs of applying cargo rstraints.

Restraint criteria will benefit the Army in the area of cargo safety. With quantitative assignment of forces to the restraints for the particular systems, the Army will be in a more defensible position if an accident results from a suspected restraint failure. With the criteria analysis system, there exists a better description of the strength and forces of all restraint components; this will more quickly pinpoint a safety hazard. Also, after-the-fact analysis of accidents will provide feedback, in terms of fractures and failures that can be compared with forces and stresses, to isolate the problem item and to improve and check on the basic criteria and analysis.

Restraint criteria should produce time savings in several areas. The time it takes to apply cargo restraints has always been important because it adds to the costly time of transportation. More accurate criteria, with resultant more accurate design, will reduce restraint application time. Time for analysis and test for restraint systems will be reduced because the criteria will be prepared in more usable form. Approval time also will be improved because it will involve principally a comparison of a criterion force, a design stress, and a measured test force. Tiedown criteria can be established by a variety of methods. Each method has applicability and improved accuracy for particular cargo configurations. The Army cargos cover a complete range of weight, elasticity, damping, and rigidity. No one tiedown-criteria method is conservative or safe for all cargos. Selection of the proper method would be based primarily on the mechanical characteristics of the cargo and the tiedown. All methods require background and experience to make proper assumptions, and some criteria methods require an understanding of basic engineering mechanics.

The four methods of establishing tiedown restraint criteria that appear to have the most applicability to Army cargos will be discussed. The target of all methods is development of the external forces required for a safe tiedown structure.

Coupler Force Method

The Association of American Railroads (AAR), Specifications for Design, Fabrication, and Construction of Freight Cars, adopted 1 September 1964, includes the following new requirement in paragraph 4.1.10.2:

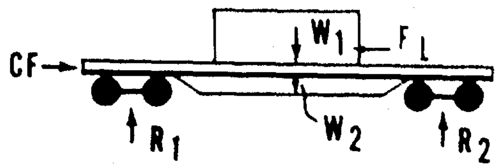
"For cars in this group using end-of-car cushioning, the car shall have a structural capability of withstanding the following coupler forces applied at one end of the car:

Length of Travel in Buff	Coupler Force
Less than 6 inches	1,250,000 pounds
6 inches but less than 9 inches	1,000,000 pounds
9 inches but less than 14 inches	750,000 pounds
14 inches and greater	600,000 pounds."

This specification covers cargo restraint and tiedown items that are part of the railcar, such as fixed and movable bulkheads, inflatable dunnage, and tiedown fittings. For many Army cargos, especially heavy items, a balanced design would suggest making the tiedown force criteria consistent with the force criteria for the other car components. The main purpose of listing coupler forces for each amount of buff travel is to insure that benefits of high-cushion railcars are used as criteria for all structural components that will be subjected to the reduced shock loadings. Tie-downs also require less strength for high-cushion cars than for standard draft-gear cars.

For the coupler force method, the appropriate coupler force (CF) is distributed to the system component masses (M) in proportion to their share of the weight (W) that opposes the CF. For a first approximation, vertical and lateral forces may be estimated as follow: take 40 percent of the longitudinal force for the vertical force and 20 percent of the longitudinal force for the lateral force. This proportion assumes that the forces in the three principal directions occur at the same time. This assumption approaches fact for railcars with long travel draft gears, because the longitudinal force duration is long enough to take in the longi-tudinal and vertical forces that are usually behind the coupler force pulse.

An example of the coupler force method for tiedown criteria would be an 80,000-pound cargo transported on a 65,000-pound, lightweight railcar with a standard draft gear, with less than 6 inches length of travel.



$$CF = 1,250,000 \text{ lb}$$
$$W_1 = 80,000 \text{ lb}$$
$$W_2 = 65,000 \text{ lb}$$

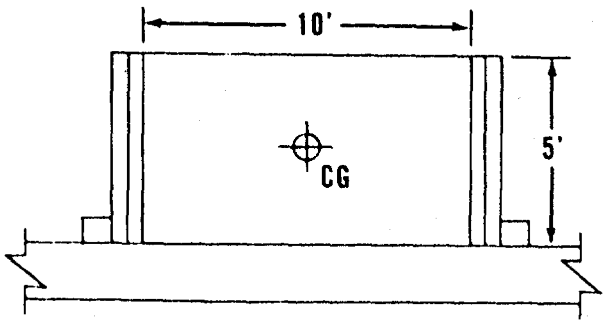
$$F_L = \text{longitudinal force, tiedown} = \frac{(CF)(W_1)}{W_1 + W_2} \quad (1)$$

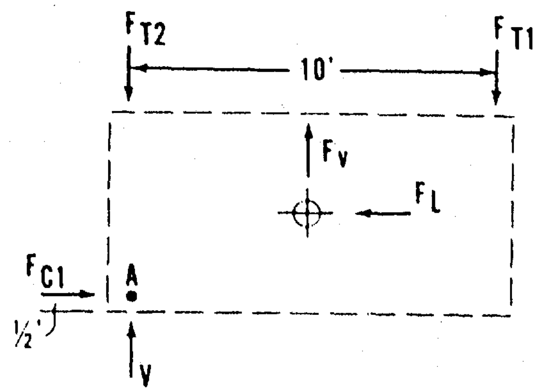
$$= \frac{(1,250,000)(80,000)}{80,000 + 65,000} = 689,650 \text{ lb}$$

$$F_v = \text{vertical force, tiedown} = 0.40(F_L)$$
$$= 0.40(689,650) = 275,860 \text{ lb}$$

$$F_{LA} = \text{lateral force, tiedown} = 0.20(F_L)$$
$$= 0.20(689,640) = 137,930 \text{ lb}$$

The restraint forces are dependent on configuration of the restraint. The size of the mass and the magnitude of the restraint forces suggest a tie-down design comprised of chocking at the railcar floor and tiedown with steel cables. With this restraint arrangement, the forces are calculated as shown on the following page:





$$\Sigma H = 0$$

$$F_{C1} = F_L = 689,650 \text{ lb}$$

where

F_{C1} = Force on chock

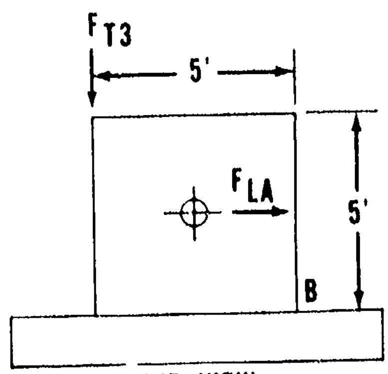
F_{T1} = Force on tiedown

$$\Sigma M_A = 0$$

$$F_{T1}(10) = F_L(2) + F_v(5)$$

$$F_{T1} = \frac{689,650(2) + 275,860(5)}{10} = 275,860 \text{ lb}$$

The portion of the tiedown forces, F_{T3}, contributed by the lateral component of the external force, F_{LA}, can be computed separately and added to the affected tiedown forces by superposition.



END VIEW

$$\Sigma M_B = 0$$

$$F_{T3}(5) = F_{LA}(2.5)$$

$$F_{T3} \text{ (lateral)} = \frac{137,930(2.5)}{5} = 68,965 \text{ lb}$$

$$F_{T1} \text{ (total)} = F_{T1} \text{ (long \& vert)} + F_{T3} \text{ (lateral)}$$

$$F_{T1} \text{ (total)} = 275,860 + 68,965 = 344,825 \text{ lb}$$

For this example, the coupler force method called for a maximum force on the chocks of 689,650 pounds and a maximum force on the tiedowns for one end of the cargo of 344,825 pounds. This force would require an impractical number of cable tiedowns and a reconfiguration of the tiedown structure is indicated. The next design configuration would relocate the structural restraint members to partition the chocking forces and use a bracing arrangement in place of the tiedown members. The resulting forces would be calculated by starting with F_L, F_V, and F_{LA}, which values are independent of the restraint configuration, and would recalculate the forces in the restraint members.

The coupler force method is useful for heavy cargos that, for comparison purposes, are out of the range of cargos that have had previous design and test work. The method is also useful, as in the example, as a quick check on the basic design configuration to determine if type of planned securement, such as wood blocking, cables, or metal fabricated structure, can resist the forces with a reasonable number or size of restraints. The coupler force method establishes tiedown forces that could be larger or smaller than actual forces because it assumes there is no relative movement between the cargo and railcar. However, some relative movement always exists, introducing such factors as spring rate, damping, and frequency. These factors, in particular combinations, could result in increased tiedown criteria forces and could also afford opportunities to reduce tiedown forces by proper design of the spring mass system. Cushioned-underframe railcars, piggyback railcars, and inflatable dunnage are examples of designing elastic properties into the system to greatly reduce forces to the tiedowns.

Empirical Method

The empirical method for developing tiedown and restraint force criteria is based on results from instrumented rail impact tests modified by pertinent specifications. This method can be only as good as the test data. Test results applied for the empirical method, to give weight to the resulting criteria, must be generally acceptable to the technical community and be published by an approving or regulatory agency. Most of the factors shown are substantially based on AAR test results from Report No. MR 443, dated August 1965. The remaining factors are based on AAR Specifications for Design, Fabrication, and Construction of Railcars.

The general procedure used is to multiply the mass by nominal acceleration, and then to multiply by a series of factors that will make the force agree with the trend of test results. Considerable interpolation and extrapolation are required because of the small number of points from test results that appear to be accepted and consistent. Empirical procedure expressed in algebraic terms is:

$$F_L = C_1 \cdot C_2 \cdot C_3 \cdot C_4 \cdot C_5 \qquad (2)$$

where

F_L = restraining force
C_1 = cargo weight
C_2 = nominal acceleration of cargo relative to railcar
C_3 = maximum rail load limit divided by actual rail load
C_4 = cargo elasticity factor, 0.5 for wood-boxed cargo
1.0 for bundled lumber cargo
1.5 for steel-boxed rigid cargo
C_5 = risk factor, impact velocity in miles per hour squared, divided by 100

The first term, F_L, or force acting at the center of gravity of the cargo, is the force that must be resisted by the cargo restraints. F_L is independent of the type of restraint used. Actual restraint and tiedown forces will need to be calculated after the general scheme of the tiedown design is developed.

The weight of the cargo to be restrained is denoted as C_1. If the cargo is unitized in several size groups, the tiedown force must be calculated separately for each cargo group. If the restraints secure parts of two different cargo groupings, separate calculations must be performed to distribute and superimpose the external forces.

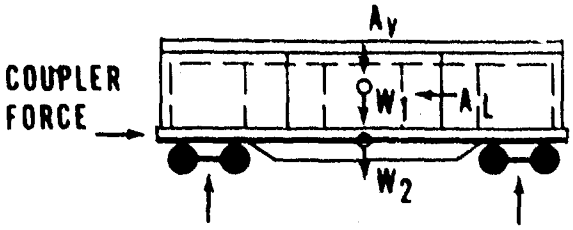
Nominal acceleration caused by railcar impact is designated as C_2. For purposes of the empirical method, the respective values are 6g, 7g, and 8g for impacts with a 50-, 70-, and 100-ton hammer car. Nominal vertical accelerations are 2g, 2g, and 3g for 50-, 70-, and 100-ton hammer cars. Lateral accelerations are 2g for all three weights of hammer car.

C_3 is a coefficient to adjust the tiedown forces relative to the total weight of cargo on the railcar. This coefficient is the ratio of the maximum weight allowable for the railcar on the rails to the actual shipping weight of all cargo on the railcar plus the light weight of the railcar.

Cargo elasticity has a pronounced effect on the magnitude of the tiedown forces. C_4 is a coefficient based on test results to correct the tiedown or restraint forces for various elasticities. C_4 ranges from 0.5 for relatively elastic cargos to 1.5 for relatively stiff or dense cargos.

C_5 is a coefficient to account for the impact speed that is frequently specified for criteria. C_5 is proportional to the kinetic energy of the hammer car at various speeds. It ranges from 0.36 for a 6-mile-per-hour to 1.44 for a 12-mile-per-hour impact velocity. C_5 enables adjustment of the restraint forces to be consistent with the risk or value of the cargo. Because of the low-value consequence of restraint member failure, many military cargoes require restraint protection as low as 6 miles per hour. Other military cargos are so important that cargo restraint protection to 12 miles per hour is needed. C_5 is set up as a separate coefficient so that multiple restraint forces can be given for various risk or impact-speed conditions.

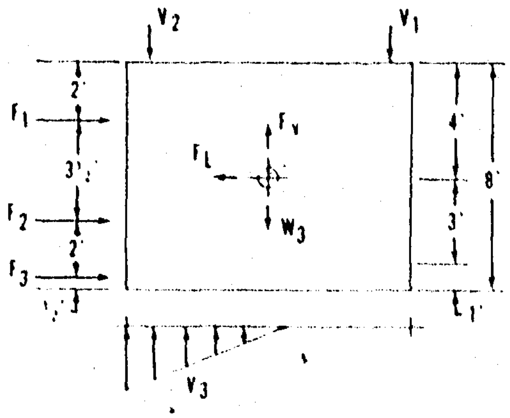
As an illustration of the empirical method of determining restraint criteria, the following transportation system is assumed. Consider boxed ammunition to be shipped in 70-ton standard draft-gear rail box-cars with a light weight of 60,000 pounds. Each railcar will contain 135,000 pounds of cargo, restrained separately in 27,000-pound unitized loads. Consider further that the cargo is specified to withstand 8-mile-per-hour impacts with a 70-ton hammer car. Assume that some unitized loading is near a railcar end, to give an external vertical acceleration of 2g. The external forces on the car are:



where

W_1 = cargo weight
W_2 = car light weight
A_V = inertial vertical force
A_L = inertial longitudinal force

and the external forces on the unitized load are:



where

W_3 = weight of unitized load inertial
F_L = longitudinal force
F_v = inertial vertical force
F_1, F_2, F_3, V_1, V_2, V_3 = resisting forces

The basic formula for the empirical method is:

$$F_L = C_1 \cdot C_2 \cdot C_3 \cdot C_4 \cdot C_5 \qquad (2)$$

where

$C_1 = W_3 = 27,000$ lb
$C_2 = 7g$
$C_3 = \dfrac{\text{Max rail load}}{W_1 + W_2} = \dfrac{220,000 \text{ lb}}{195,000 \text{ lb}} = 1.13$
$C_4 = 1.0$ (boxed ammunition)
$C_5 = 0.64$ (for 8 mph)

$$F_L = (27,000)(7)(1.13)(1.0)(0.64) = 136,580 \text{ lb}$$
$$F_v = 2(27) = 54,000 \text{ lb}$$

F_L and F_v are the basic inertial forces of restraint criteria. The external resisting forces are dependent on the tiedown configuration, as in the first example. The AAR specifies for bulkhead design that restraint forces should be distributed 100 percent over the entire bulkhead, 80 percent over the lower half of the bulkhead, and 60 percent on the bottom 12 inches of the bulkhead. If the unitized ammunition is 8 feet high, the distribution of the external longitudinal forces F_1, F_2, and F_3 is as follows:

$$F_1 = \frac{1.0}{2}(F) = 68,290 \text{ lb}$$

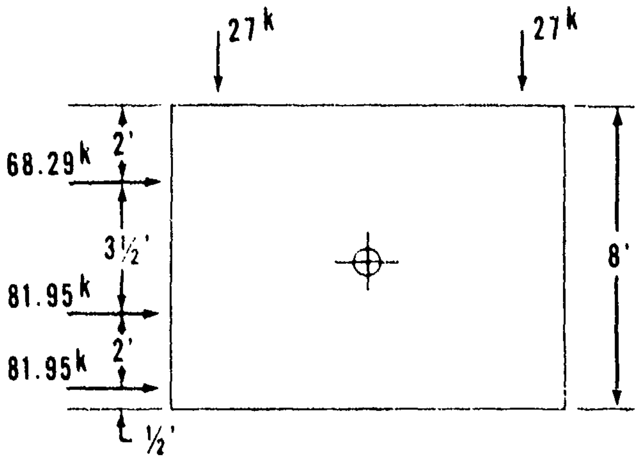
$$F_2 = 0.80(3/4)(F) = 81,950 \text{ lb}$$

$$F_3 = 0.60(F) = 81,950 \text{ lb}$$

F_1, F_2, and F_3 were not obtained from equilibrium equations, but from factors to account for restraint loads concentrating at the bottom due to cargo elasticity, and should not be used in equilibrium equations. Vertical tiedown loads normally are taken to be only the vertical inertia load. The vertical force due to rotation of the unitized load usually is calculated separately, or is assumed to dissipate by free rotation. Also, since the 2g vertical is taken from test data for this example, the magnitude is 2g, independent of the 1g gravity or weight of the cargo.

$$V_1 = V_2 = \frac{F_v}{2} = \frac{54,000}{2} = 27,000 \text{ lb}$$

The resulting tiedown criteria, expressed as forces, acting on the unitized cargo in the positions shown, is:

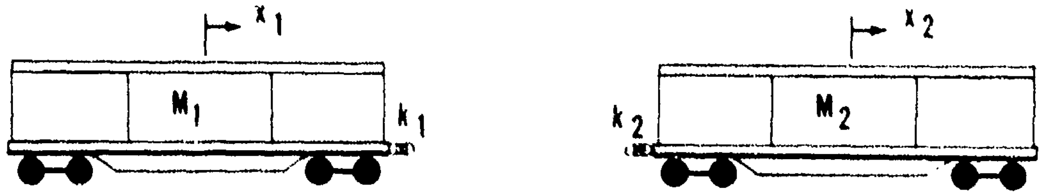


The empirical method produces the best accuracy for restraint criteria when considerable test data exist for cargo with mechanical characteristics similar to those of the cargo needing restraint criteria. Ammunition, boxed goods, and vehicles are examples of cargos with existing test data. When cargo is intended for shipment by standard draft-gear railcars, the empirical method is applicable because tests are usually conducted with standard cars that normally establish the most severe tiedown forces.

All of the factors and coefficients comprising the empirical method are entirely sensitive. If the information needed to select a coefficient for the cargo is unknown, a conservative selection is usually made. The resulting estimated restraint criteria can be so high that the tiedown structure would be impracticable. When this happens, one of the other methods is suggested.

Spring Mass Method

During rail impact tests, measured accelerations on the cargos vary from 2g to 60g for the same basic impact velocities. The acceleration spread is caused by different masses, elasticities, and spring rates. To establish meaningful cargo restraint criteria, it is necessary to compute the effects of various masses, elasticities, and frequencies of components of the spring mass system. This method is basically the classical method of establishing the motions of the system as a function of time to establish the proper interaction of the forces and the effects of the system frequencies. Consider two railcars whose masses are M_1 and M_2 during impact. The cars have draft gears or cushioning denoted by k_1 and k_2 and the respective car displacements are indicated by x_1 and x_2 as shown:



For calculation purposes, the two active draft gears will be considered as two springs, in series, with the combined spring rate k, or,

$$k = \frac{1}{\frac{1}{k_1} + \frac{1}{k_2}} \tag{3}$$

The equations of motion are:

$$M_1\ddot{x}_1 + k(x_1 - x_2) = 0 \tag{4}$$

$$M_2\ddot{x}_2 + k(x_2 - x_1) = 0 \tag{5}$$

where $\ddot{x}$ denotes the acceleration of each mass.

The solutions to these equations are of the exponential type:

$$x_1 = A_1 e^{\omega t}$$

$$x_2 = A_2 e^{\omega t}$$

where ω is the natural circular frequency of the system.

For the spring mass given, ω reduces to:

$$\omega = \sqrt{\frac{k}{M_1} + \frac{k}{M_2}}$$

Constants A_1 and A_2 are determined from the initial conditions of the example. The expression of displacement may also be written as:

$$x_1 = C_1 \sin\omega t + C_2 \cos\omega t$$

$$x_2 = C_3 \sin\omega t + C_4 \cos\omega t$$

where C_1, C_2, C_3, and C_4 are new arbitrary constants replacing A_1 and A_2. By substituting the initial condition, the constants are found to be:

$$C_2 - C_4 = x_o = 0$$

$$C_1 - C_3 = \frac{v_o}{\omega}$$

where x_o is the compression of the springs, and v_o is the initial velocity of the hammer car.

The spring deflection $(x_1 - x_2)$ is given by the equation:

$$(x_1 - x_2) = \frac{v_o}{\omega} \sin\omega t \tag{7}$$

A is the amplitude of vibration and for these specific conditions is:

$$A = \frac{v_o}{\sqrt{\frac{k}{M_1} + \frac{k}{M_2}}} \tag{8}$$

The acceleration is a maximum when the draft gears are fully closed, or when $A = x_1 - x_2 = -(x_2 - x_1)$ substituting in:

$$M_1\ddot{x}_1 + k(x_2 - x_1) = 0 \tag{9}$$

$$\ddot{x}_1 = \frac{-kA}{M_1} = \frac{-kv_o}{M_2\sqrt{\frac{k}{M_1} + \frac{k}{M_2}}} \tag{10}$$

For cargos with nearly rigid tiedowns or restraints, $\ddot{x}_1$ may be used to calculate the tiedown forces.

For an example of the spring mass method, consider a cargo loaded in a railcar to 130, 000 pounds rail weight. Restraints are to be designed to resist an 8-mile-per-hour impact with a 100-ton car, 260, 000 pounds maximum rail load.

$$M_1 = \frac{130,000 \text{ lb/ft/sec}^2}{32.2} \quad M_2 = \frac{260,000 \text{ lb/ft/sec}^2}{32.2}$$

$$k = \frac{1}{\frac{1}{k_1} + \frac{1}{k_2}} = 187,000 \text{ lb/in}$$

$$\omega = \sqrt{\frac{kg}{W_1} + \frac{kg}{W_2}} = 29 \text{ rad/sec} \tag{11}$$

$$f = \frac{1}{2\pi}\omega = 4.5 \text{ cps} \tag{12}$$

$$\text{pulse time} = \frac{T}{2} = \frac{1}{2f} = \frac{1}{2(4.5)} = 0.111 \text{ sec} \tag{13}$$

$$A = \frac{v_o}{\sqrt{\frac{k}{M_1} + \frac{k}{M_2}}} = \frac{(8)(22)}{(15)(29)} = 0.40 \text{ ft or } 4.9 \text{ in.} \tag{8}$$

$$\ddot{x}_1 = \frac{-kA}{M_1} = \frac{(187)(12)(0.40)(32.2)}{130} = -222 \text{ ft/sec}^2$$

$$= \frac{-222}{32.2} = -6.90g \tag{10}$$

If the cargo is rigidly supported to the railcar and has a high natural frequency, the restraint criteria would be 6.90g at 4.5 cycles per second.

If the cargo has a natural frequency anywhere near the restraint criteria frequency, it might cause serious amplification of the restraint forces.

The magnification factor is given by the expression:

$$\frac{1}{1 - \frac{\omega^2}{\omega_n^2}} \qquad (14)$$

where ω is the frequency of the impact force, and ω_n is the natural frequency of the mass receiving the impressed dynamic loading. This factor can go to infinity or resonance. Damping in the system reduces this magnification. Most cargos have large damping or friction built in the restraint process so the maximum amplification should not be larger than 2.

If, in the preceding example, the natural frequency of the cargo were 10 cycles per second, the restraint criteria would have to be modified as follows:

$$\omega = 29 \text{ rad/sec}$$

$$\omega_n = 10(2\pi) = 62.83 \text{ rad/sec}$$

$$\frac{1}{1 - \frac{\omega^2}{\omega_n^2}} = \frac{1}{1 - \frac{(29)^2}{(63)^2}} = 1.27 \qquad (14)$$

$$1.27(6.90g) = 8.75g$$

Magnification does not change the frequency. The revised restraint criteria for the example would be 8.75g at 4.5 cycles per second. As in the other examples, the actual restraint forces are dependent on the restraint design and on the mass of the cargo to be restrained. From this point, the calculations proceed as in the first two examples.

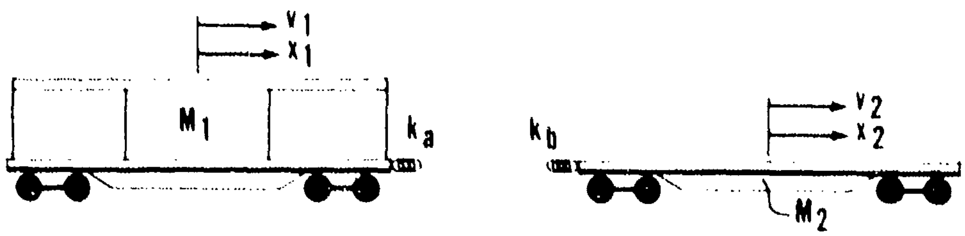
Damping has not been considered for the basic spring-mass restraint criteria method for several reasons: first, it is beyond the scope of this discussion; and second, damping rarely plays an important role in the first cycle or impact. Even with impact, if the natural frequency of the cargo and the input frequency are close, the amount of damping required to reduce the restraint forces to a reasonable level must be calculated and provided for in the design. For most cargos with relative motion between the cargo and the railcar, the spring-mass method can be used successively. During impacts, the shocks are not simultaneous so that the force and frequency output from one system can be considered the input forced vibration to the next system, and good accuracy can be obtained. The principal advantage of the spring-mass method of restraint criteria is that it develops the equations of motion for the system. The system frequencies are quite sensitive and, to a large extent, determine the restraint forces. Once the equations of motion are determined, based on needed frequencies and spring-mass relationships, changes and improvements can be made to the design to obtain optimum results in safe restraints and minimum shocks and vibrations to the cargo. The spring-mass method does not work well with crude restraint designs. In conditions of acceptable cargo crushing, uncontrolled displacements, cargo bouncing, and other erratic movement, either the coupler force or the empirical method for developing restraint criteria is preferable.

Energy Method

For many cases of restraint criteria, only the maximum values are of interest, and the equations of motion need not be developed. The energy method equates the resultant work to the kinetic energy change in the system from initial contact to time of peak coupler force. The energy method is most applicable when the mechanical system experiences more than one impact before the cargo restraints are stressed. For rail piggyback, the first impact is between the hammer car and the railcar carrying the trailers.

The only interest in this impact for restraint purposes is to get the result of the impact in terms of force and frequency. Some time after the first impact, or after peak coupler forces, a second impact occurs, which is between the test car and the trailers. It is the second impact that produces the maximum force on the trailers and the cargo restraints. This same sequential-type impact occurs with most standard draft-gear car arrangements and with any cargo that is independently sprung in the shipping configuration.

The energy method is based on the fact that the change in kinetic energy is equal to the energy absorbed in the springs at the time of maximum spring deflection. Consider the impact between a standard freight car and a piggyback car carrying two trailers. M_1 denotes the mass of the hammer car; M_2 the mass of the piggyback railcar; and M_3 and M_4, the mass of two trailers carried by the piggyback car. The spring rate of the hammer-car cushion is indicated by k_a, and k_b denotes the spring rate of the piggyback car cushion. The combined effect of k_a and k_b is denoted by k_1. The velocities throughout the impact cycle are denoted by l_1. The velocities throughout the impact cycle are denoted by v_1 for the hammer car and v_2 for the piggyback car. x_1 and x_2 are the displacements for the hammer car and the piggyback car, respectively.



$$k_1 = \frac{1}{\frac{1}{k_a} + \frac{1}{k_b}} \tag{15}$$

The energy absorbed in the system due to the impact is assumed to be entirely absorbed by the springs. The springs are assumed to be linear, and no account is taken for their precompression. Also, any elastic action of the railcars or cargo that absorbs energy is included as part of the draft gear. The combined spring rate, k_1, is taken from instrumented test data that account for extraneous potential energies. The maximum spring travel of $x_1 - x_2$ is denoted as x_{1m}, and the maximum velocity difference $v_1 - v_2$ is v_{1m}. The absorbed energy is expressed as ΔE_1.

$$\Delta E_1 = \frac{1}{2} k_1 x_{1m}^2 \tag{16}$$

The equation for the spring deflection in terms of the maximum deflection would have to be a sine function of time, or:

$$(x_1 - x_2) = x_{1m} \sin \omega t \tag{17}$$

The velocity of the spring compression is found by differentiating the displacement formula, or:

$$(v_1 - v_2) = \omega x_{1m} \cos \omega t \tag{18}$$

The maximum velocity difference would have to be at $t = 0$, where $\cos \omega t = 1$.

$$v_{1m} = \omega x_{1m} = v_o$$

Before impact, M_2 is motionless and M_1 contains all of the kinetic energy of the system. At the time of maximum spring deflection, M_1 and M_2 form one mass traveling at the same velocity, and this single mass contains all of the kinetic energy of the system. The kinetic energy change is:

$$\Delta E_1 = \frac{M_1 v_1^2}{2} - \frac{(M_1 + M_2)}{2} v_2^2 \tag{19}$$

Using the principle of the conservation of momentum:

$$M_1 v_1 = (M_1 + M_2) v_2 \tag{20}$$

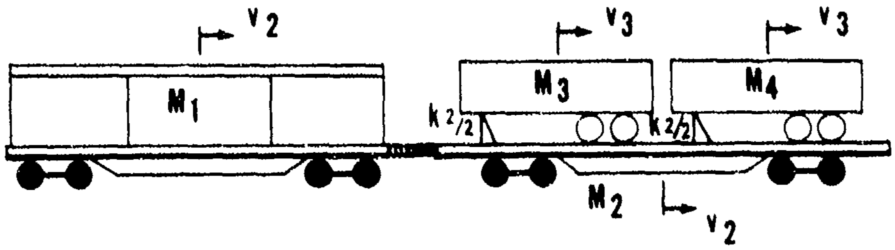
$$\Delta E = \frac{M_1 v_1^2}{2} \left[1 - \frac{M_1}{M_1 + M_2} \right] \tag{21}$$

$$\frac{1}{2} k_1 x_{1m}^2 = \frac{\omega_1^2 x_{1m}^2}{2} \left[M_1 - \frac{M_1^2}{M_1 + M_2} \right] \tag{22}$$

$$\omega_1 = \sqrt{\frac{k_1}{M_1} + \frac{k_1}{M_2}} \tag{23}$$

$$x_{1m} = \sqrt{\frac{M_1 v_1^2}{k_1} \left[1 - \frac{M_1}{M_1 + M_2} \right]} \tag{24}$$

The same procedure is used for the second impact, where the hammer car and the piggyback railcar join together as one mass and impact the trailers:



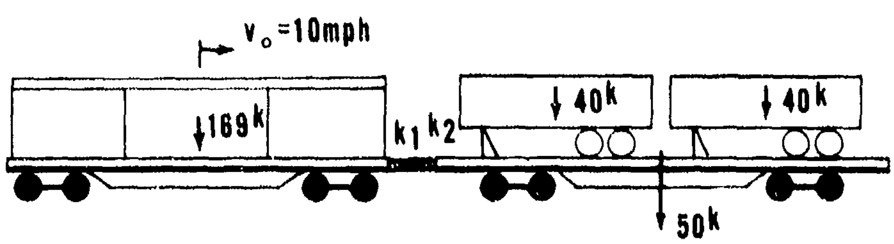
$$\omega 2 = \sqrt{\frac{k_2}{M_1 + M_2} + \frac{k_2}{M_3 + M_4}} \tag{25}$$

$$\frac{x_2(m_2)^2}{2} = \frac{M_1 v_1^{\ 2}}{2}\left[\frac{M_1}{M_1 + M_2} - \frac{M_1}{M_1 + M_2 + M_3 + M_4}\right] \tag{26}$$

With the double impact, the output force from the first impact is the impact force to the second impact, which can cause amplification to the cargo restraint forces. This magnification is expressed by:

$$\frac{1}{1 - \dfrac{\omega_1^{\ 2}}{\omega_2^{\ 2}}} \tag{27}$$

For an example, consider cargo weighing 35,000 pounds restrained rigidly in 5,000-pound trailers. The piggyback railcar weighs 50,000 pounds and carries two trailers. The restraint forces are required to protect against 10-mile-per-hour impact with a 50-ton hammer car, 169,000-pound maximum rail load.



$$k_1 \text{ (both draft gears)} = 166,600 \text{ lb/in.}$$

$$k_2 \text{ (both hitch cushions)} = 19,100 \text{ lb/in.}$$

$$\omega_1 = \sqrt{\frac{k_1 g}{W_1} + \frac{k_1 g}{W_2}} = \left[\frac{(166.6)(32.2)(12)}{169} + \frac{(166.6)(32.2)(12)}{50}\right]^{1/2} \tag{28}$$

$$= 40.84 \text{ rad/sec or } 6.5 \text{ cps}$$

$$x_{1m} = \sqrt{\frac{M_1 v_1^2}{k_1}\left[1 - \frac{M_1}{M_1 + M_2}\right]} \tag{24}$$

$$= \sqrt{\frac{(169)(14.66)^2}{(166.6)(12)(32.2)}\left(1 - \frac{169}{169 + 50}\right)}$$

$$x_{1m} = 0.359 \text{ ft or } 4.3 \text{ in.}$$

$$\omega_2 = \sqrt{\frac{k_2}{M_1 + M_2} + \frac{k_2}{M_3 + M_4}} \tag{25}$$

$$= \sqrt{\frac{(19.1)(32.2)(12)}{169 + 50} + \frac{(19.1)(32.2)(12)}{40 + 40}}$$

$$\omega_2 = 11.22 \text{ rad/sec or } 1.78 \text{ cps}$$

$$x_{2m} = \sqrt{\frac{M_1 v_1^2}{k_2}\left[\frac{M_1}{M_1 + M_2} - \frac{M_1}{M_1 + M_2 + M_3 + M_4}\right]} \tag{26}$$

$$x_{2m} = \sqrt{\frac{169(14.66)^2}{(19.1)(12)(32.2)}\left[\frac{169}{169 + 50} - \frac{169}{169 + 50 + 40 + 40}\right]}$$

$$x_{2m} = 1.0 \text{ ft or } 12 \text{ in.}$$

The coupler force, F_1, is equal to the draft gear travel, times the spring rate, or:

$$F_1 = x_{1m} k_1$$

$$= 4.3(166,000) = 716,400 \text{ lb}$$

and the hitch force, F_2, is found:

$$F_2 = x_{2m}k_2$$

$$F_2 = 12(19,100) = 229,200 \text{ lb (for 2 trailers)}$$

by applying the amplification factor:

$$\frac{1}{1 - \frac{\omega_1{}^2}{\omega_2{}^2}} = \frac{1}{1 - \left(\frac{20.19}{11.22}\right)^2} \qquad (27)$$

$$= -0.45$$

In this example, the magnification factor is negligible. This indicates that the piggyback system was designed to operate out of phase and prevent amplification when carrying an average-size cargo. Here again, the cargo restraint must be checked for its natural frequency, ω_3, and compared with ω_2 for possible magnification at the restraint.

The restraint criteria for the example can be determined by proportioning the hitch forces to the cargo and trailer. If F_3 is the force on the cargo and W_5 is the weight of cargo in one trailer, and W_3 is the weight of cargo plus trailer, then:

$$F_3 = \frac{F_2}{2}\left(\frac{W_5}{W_3}\right)$$

$$= \frac{229.2}{2}\left(\frac{35}{40}\right) = 100,300 \text{ lb (per trailer load)} \qquad (29)$$

For the example, the resulting criteria would be 100,300 pounds longitudinal force, acting at the center of gravity of the 35,000-pound trailer cargo, with a frequency of 1.78 cps.

Summary

Cargo restraint criteria, when expressed as acceleration of the cargo for transportation, will vary from 2g to 60g depending on the weights and elasticities of the cargo, on the restraint structure, and on the degree of protection required. Cargo restraint criteria can be calculated to obtain safe and balanced cargo restraint designs.

There are at least four methods to calculate cargo restraint criteria. Each is applicable to particular types of cargos and restraint arrangements. The methods can be used separately or in combination, and parts of each method can be mixed to obtain criteria. Two methods normally are calculated to afford a check on the resulting criteria.

The empirical method compares the cargo to be restrained with similar cargo arrangements that have been tested. The restraint criteria are as good as the similarity between what is needed and the existing test results. The empirical method is applicable to crude restraint systems with poorly defined restraint members, poor structural connections, and uncontrolled cargo motions.

The coupler force method proportions, by weight, the maximum specified coupler force to the car body and cargo. This method is dependent on rigid cargo restraints, as it assumes that all forces are peaking at the same instant. The coupler force method is particularly applicable to extremely heavy railcars carrying rigid, shear-connected cargos such as transformers and nuclear casks. With lightly or partially loaded railcars, the coupler force method gives excessive rail restraint criteria forces.

The spring-mass method is the classical one for mechanical vibrations. It enables the cargo restraint criteria to be presented as equations of motion. By this method, the frequencies and forces throughout the entire impact cycle are calculated. The spring-mass method is particularly applicable in the restraint design stage where restraint criteria can be given for both force and frequency. The desired restraint frequency can be provided in the design to prevent force magnification.

The energy method considers only maximum values to determine restraint criteria. Frequencies can be computed with this method, and they require a little less calculation than with the spring-mass method. The energy method is quite adaptable in computing restraint forces consequent to multiple impacts as with piggyback systems or cargos with built-in cushioning.

Restraint criteria forces must be distributed from the cargo center of gravity to the restraint members. Restraint design will, to a degree, establish the magnitude of criteria forces. This is particularly important when the cargo restraint is of wall or floor chock type. Due to differences in the restraint structure flexibility, the restraint forces are greater at the bottom of the restraint than at the top.

This report covers the development of the formulas for rail restraint criteria. The Military Traffic Management Command Transportation

Engineering Agency (MTMCTEA) plans to publish the criteria formulas of this report, with complementing test procedures and other technical information, in a condensed form to comprise a military standard covering cargo transportability by rail mode.

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

This report discusses criteria for cargo restraint and cargo tiedown for the forces resulting from the rail transportation environment. The report covers methods for calculating the restraint forces for a broad range of cargo railcar combinations. Use of these methods will improve accuracy of restraint criteria and, in turn, will improve reliability of cargo restraints and reduce costs consequent to restraint overdesign.

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