REPORT *of*

DRAFT GEAR TESTS

United States Railroad Administration
Inspection and Test Section

Preface by
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PREFACE

When the United States Railroad Administration decided in the spring of 1918 to enter upon its car and locomotive building program, one of the problems which early came before the Committee on Standards for Locomotives and Cars and the Central Advisory Purchasing Committee was the selection of draft gears to be used and the allocation of orders among the several manufacturers. The Committee on Standards and the Purchasing Section were both embarrassed, owing to a lack of definite and positive knowledge as to the relative merits of the different gears as well as the relation between mechanical value and cost. Much information on the subject of draft gears was presented by the various manufacturers, but a comparison of the information presented soon developed the fact that each manufacturer had prepared his information on a basis of his own selection and that it was impossible to correlate or co-ordinate the various tests in any comparable manner. The reports of the Draft Gear Committee of the Master Car Builders' Association and the files of the mechanical associations failed to give any definite information on the subject.

In the absence of real information, the Committee on Standards adopted the wording of the M.C.B. specification for draft gears for Class III and Class IV tank cars which provides that the gears purchased shall have a "minimum capacity of 150,000 lb." The committee later defined this requirement in the following words:

> "A 150,000 lb. draft gear should be defined as one that will sustain a drop of 16 in. (including travel of the gear) of a 9,000 lb. weight without shearing the rivets of one or both lugs which are to be secured to suitable members by nine ½ in. rivets of .15 carbon or under, driven in $\frac{9}{16}$ in. holes."

When gears were tested under this requirement, it was found that no useful information was obtained. Gears of widely varying characteristics and excellence passed the prescribed test and it was soon appreciated that the specification requirement as well as this test were useless in obtaining draft gear information.

When this absolute dearth of reliable knowledge on the subject was fully realized by the Committee on Standards and the Purchasing Committee, they joined in requesting the Inspection and Test Section of the Division of Operation to conduct such a series of tests as would determine the mechanical value of each make and type of friction draft gear then regularly offered for sale to railroads.

In addition to the various tests which have been completed and which are given in the report, the Section had definite plans made for train operation tests and service tests. Had the time been available and had circumstances permitted, the Section would have completed these tests.

It is much to be regretted that conditions on the railroads throughout the country during the war and immediately thereafter prevented the carrying out of these tests and this, to a degree, operates to render the present work inconclusive.

The information covering the tests which have been made on new gears is definite and final. To a limited degree, tests were made on gears which had seen considerable service but the service tests themselves and the train operation tests were not made for the reasons given.

It is much to be hoped that arrangements will be made to complete the full series outlined by the Section and thereby render available accurate information concerning the action of gears in train operation and the ability of each type of gear to stand up in service. With this added information, mechanical officers and purchasing agents would be able to equate value and cost and to understandingly purchase a definite amount of protection for a definite amount of money.

If the present report does no more, it gives reliable and entirely comparable and unbiased values for new commercial gears of the various types. The values given should supplant the widely variant figures frequently given out in the past as a result of inaccurate, unscientific or incomparable tests.

Attention should be called to the fact that this report must be used as a whole in order to obtain accurate and definite information concerning the draft gears. The picking out and exploiting of an idea shown here or there throughout the test and which favors one or the other of the draft gears tested, should be heartily discouraged and those who use the report should guard themselves against errors of this kind. The pros and cons of all gears must be thoroughly balanced by those who are looking for the truth.

In the chapter entitled "Grading of Average Commercial Gears" will be found the only place where personal opinion has in any manner entered into the report. The assignment of the number of points of excellence to the various functions of the gears is on the basis of the ideal gear and engineers who study the work may not entirely agree with this assignment.

Attention is also called to the fact that on plate 69 where these points of excellence are used to rate the various gears, a column covering wearing qualities has not been included. Engineers will, of necessity, record their opinions and observations as to wearing qualities and in so doing may materially change the grading of the gears as shown on this table.

In making these tests the path was entirely unbroken, the trail was unblazed. It was necessary to avoid many previous methods of testing that are erroneous and misleading. It was also necessary to forget at the outset the values of the several gears as generally reported and accepted. It was necessary to lay aside all prejudices and personal preferences.

With one or two exceptions the tests were welcomed by the draft gear manufacturers, and their full co-operation was freely given.

The importance of the type and design of the draft gear attachments is often not fully appreciated. The report covering the tests of attachments and of reinforced and unreinforced wooden car construction gives, probably for the first time, reliable figures for the comparison of these features of construction, and also gives some slight hint of the wealth of information on general car construction that can be developed from actual impact tests, if carefully made and reported.

Acknowledgment is made of the services and hearty co-operation of Messrs. B. W. Kadel, E. M. Richards and L. H. Schlatter in the active conduct of the test in the field as well as the working up of the data contained herein.

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Manager of the Inspection and Test Section of the Railroad Administration during Federal Control.

Chicago, Ill.,
January 20, 1921.

DRAFT GEAR TESTS OF THE U. S. RAILROAD ADMINISTRATION, INSPECTION AND TEST SECTION

The draft gear tests of the United States Railroad Administration were originally undertaken at the request of the Committee on Standards for Locomotives and Cars and the Central Advisory Purchasing Committee for the purpose of determining the relative merits of the several commercial gears in order that mechanical excellence and costs might be evaluated. The Inspection and Test Section, as a preliminary to any work, carefully studied all of the common methods of testing draft gears. Letters on the general subject were also addressed by the section to all of the draft gear manufacturers and to a large number of prominent mechanical officers of the roads, the replies to which showed a wide difference of opinion, not only as to the proper method of testing draft gears, but as to what performance should be expected from a gear.

A comparison of the many test reports submitted, showed an entire inconsistency in results, supposedly obtained under similar conditions. It became evident that a test of all gears under exactly the same conditions, removed from any proprietary influence, was essential, and also that the tests should be conducted in such a manner as not only to determine the comparative value of the several gears, but to obtain all the exact information possible with respect to draft gear action, and to extend the study as far as possible toward the ultimate determination of the ideal draft gear. With such a program in view, the co-operation of the A. R. A. Committee on Draft Gears was felt to be desirable, and upon invitation from this section, this committee has taken an active part in the test work and in analyzing and compiling the results.

The present report covers in a rather extensive manner the action and comparative merits of the various gears when considered from the viewpoint of impact and buffing. The opportunity for the investigation of draft gears in train starting and similar operations has not developed as was hoped for, so that it is impossible at this time to present definite information in this latter respect. It is desired accordingly, that this report, which compares the several commercial gears and deals extensively with the question of cushioning and absorbtion, shall be considered only as a part of an extended investigation into the action of draft gears, not only in buffing and impact, but also in train starting and handling.

The full investigation of draft gears should include not only the laboratory and impact tests of the present report, but also a wide range of train operation tests and service tests, from the results of which should ultimately be determined:

1. The minimum amount of movement necessary between cars for starting trains, and whether this movement may be free slack, as between coupler knuckles, or whether it should be resisted movement.

2. Whether the beginning of draft gear compression should be an easy movement or a stiff movement, and whether there should be an initial compression to prevent movement from slight shocks.

3. The effects of recoil and what amount of release force is desirable.

4. The desired capacity, travel, and ultimate resistance of the gear, as well as the shape of the curve representing draft gear resistance for both buffing and train starting.

5. The coupler horn clearance and coupler shank clearance.

6. The life, together with the rate of wear and loss in gear capacity attending it, that should be expected from an acceptable draft gear, as well as the setting of a measure, either in time, mileage, or loss of capacity, when a draft gear should be removed from the car and be repaired or scrapped.

DRAFT GEAR TESTING

The following discussion on the general subject of draft gear testing is given for the benefit of any who may be called upon to do similar work in the future.

It is important to have a full knowledge of the condition of each test gear before putting it into a test. Check measurements should be made, such as spring heights, barrel or housing dimensions, initial spring compression, initial friction compression, absolute free height, absolute friction height, and solid height, keeping a record of possible travel at any of the previously mentioned gear heights. By having such a record it will later be possible to check up the gear conditions and to know whether any loss in travel is due to set of springs, wear of friction members or deformation of parts of the gear. Depreciation in any of these respects should be reported in equivalent loss in coupler or gear travel.

It is important to protect the friction surfaces of test gears from any grease, rust or moisture. Even the handling of the friction faces with bare hands may leave enough grease or moisture on them to lower the gear capacity. After taking a new gear apart it should be reassembled with the parts always in their original relationship, and the gear should then be operated not less than ten times before making a regular test. Any rust on the friction surfaces should be removed by sand papering, and the gear should then be operated not less than twenty times if comparable and consistent results are to be obtained. This does not mean that the friction faces of draft gears do not have deposits of rust and other foreign material on them in service, but is given as a rule for conducting comparative tests of new gears.

In testing draft gears, the gear should not be loaded beyond the solid point. Few gears will stand much service beyond their normal capacities, especially under the drop machine. The determination of the solid point, however, is often quite difficult. Sometimes the spring coils, or other internal gear parts, will go solid before the gear is fully closed. The result is that a greater load or drop is required to fully close the external portions of the gear than would be required if normal action obtained throughout. The static test is best suited to accurately fix the limit of normal gear closure. In tests of other characters, such as the drop test, the gear should be closed only to the travel determined from the static cards as the limit of normal gear action.

All gears, irrespective of construction, should be set up and restrained in a suitable testing frame, corresponding in dimensions to the draft gear pocket in the car. The frame should be so designed that the influence of its yield will be minimized, The gear should rest in the frame upon pieces of metal corresponding to the stop faces of the gear draft lugs or other stop member. A striking plate of the same size as the coupler butt should be placed on top of the gear for receiving the blow. This will develop whether or not the gear construction is substantial enough to receive the coupler butt forces in service. Where followers are regularly used with a gear, they should, for comparative purposes, be set up with the gear in the testing frame. In all respects service conditions should be simulated in the testing frame, as in no other manner will the weak or strong points of a gear be shown. It is more convenient to test gears such as the Miner, Westinghouse and similar types without a frame, but a frame is necessary for some other gears, such as the Cardwell, and in any impact testing the yield of the frame, no matter how carefully constructed, may slightly increase the results. It is therefore only fair that all gears should be tested under similar conditions.

On the subject of heating but little needs to be said. It is not often that a gear will be operated fast enough to heat it sufficiently to affect the results unless a wear test or endurance test is being made. In such a test the gear should not be allowed to become more than just warm to the hand.

It is a noticeable fact, however, that if a friction gear is brought for testing from a cold place into a warm room, the capacity will be low; and if brought from a warm room to a colder outside atmosphere, the capacity will be higher. This is due to the deposit of moisture on the colder metal, or the abstraction of moisture from the friction surfaces of the warmer metal, as the case may be. In general the humidity of the air is a decided factor in testing, and an instance is known of a depreciation of 20 per cent in a gear which could be explained in no other manner.

Another point of interest is that when a gear is to be given a static test without a frame, and the free height of the gear is greater as set up than the pocket length in the car, the gear should first be compressed to slightly below the pocket dimension and then released to the exact pocket length. The compression test should then start from this released point.

In impact testing, where the load passing through the gear to the supporting device is measured or compared, the gear should never be tested beyond the closing point. This rule applies particularly to rivet shearing tests and car-impact tests. It should be remembered that after a gear goes solid its normal functioning ceases, and further testing is only of the gear housings or barrel. Hence in over-solid testing the greater deformation of a weaker gear barrel offers additional protection to the rivets for the time being, and also offers more yield in the car tests. Any considerable repetition of such over-solid blows would, however, shortly destroy the gear. On the other hand, a sturdy gear will usually shear the rivets at the first over-solid blow and will similarly produce a sudden change in car velocity, but the sturdy gear will not be so quickly destroyed. In practice, no one would knowingly use a weak draft gear in order to protect draft lug rivets, but draft gear tests are frequently made with this object in view. A weak gear barrel will show up well enough for the few over-solid blows given it in a laboratory, but will shortly be depreciated or destroyed from the repetition of such blows as occurs in service. In fact, if a gear of sturdy design should shear the ½ in. rivets at say a total fall of 16 in., it would be entirely practical to increase this figure several inches by simply reducing the thickness of the barrel or other part receiving the solid blow. For a full knowledge of the functioning of a gear it is necessary to know only its capacity up to the point of closure and the character of its action within that capacity. Any yield or cushioning beyond the solid point is due to deformation or spring of the heads or barrel, and is obtained only at the expense of strength and life of the gear.

The suggestion is frequently made that all gears be tested to determine the point where a force of say 500,000 lb. is set up in the sills. On the face this would appear to be entirely reasonable and a proper test for the grading of gears. But for the same reasons as before, a premium would be placed upon a weak gear construction. Furthermore, it is a fundamental principle of mechanics that there can be no force set up in any structure greater than the resistance offered by the structure. It therefore follows that if a gear were constructed with an ultimate strength value of 400,000 lb. it would be physically impossible to apply 500,000 lb. through it to the car. Hence, the only over-solid draft gear tests that should be made are those that will discover the weakness of a gear rather than credit it with false merit. The destruction and endurance tests are the only over-solid draft gear tests known that will correctly rate the gears in this respect.

Another practice from which wrong conclusions are often drawn is that of testing gears against sills of different sizes and conditions. It is not fair to set up one gear on heavy channels and another on light channels, as again, the force developed will depend upon the yield and the resistance offered by the channels. Thus if a test were made upon 20-lb. channels it would be unreasonable to expect as high a force as upon 30 lb. or 40 lb. channels, for not only is there a greater yield of the channel, but the elastic limit of the material in the lighter channels might be reached and passed, which would preclude the possibility of reaching as high a force as might be shown in the heavier channels. In other words, it is impossible to put more load into the light channels than they will stand, as the force is limited by the resistance of the structure supporting the gear.

THE TEST PROGRAM

The following general program was decided upon for the present tests as offering the best means of investigating the comparative action of the gears:

9,000 lb. Drop Tests—Solid Anvil.

- Closing gears by drops of 1 in. increments.
- Recoil tests.
- Investigation of influence of foreign material on friction surfaces.
- Investigation of rivet shearing tests.
- Destructive tests.

Static Tests.

- Closing gears at a rate of ⅛ in. per minute.
- Closing gears at a rate of ¾ in. per minute.
- Closing gears at a rate of 3 in. per minute.

Car Impact Tests.

- Calibrated gear in one car only, solid buffer in another car.
- Calibrated gears in both cars.

In general three each of 18 different types of draft gears are embraced in the tests. The table of Fig. I has been prepared to identify the gears and to give other data of prime interest in connection with them. Fifty-nine gears in all were used because of gear failures developing during the test as follows:

Westinghouse NA-1 gears No. 4 and No. 5 failed in the slow static test.

Sessions K gear No. 9 failed in the slow static test.

Bradford gears No. 43 and No. 44 failed in the drop test.

TEST GEAR NUMBER	MAKE AND TYPE OF GEAR	NOMINAL TRAVEL	NOMINAL LENGTH	AVERAGE WEIGHT ONE GEAR	NUMBER OF STD. FOLLOWERS REQD. PER CAR	COMPARATIVE WEIGHT PER CAR
	(1)	(2)	(3)	(4)	(5)	(6)
1, 2, 3	WESTINGHOUSE D-3	2 7/16"	20 1/8	200#	4	684#
4, 5, 6, 7, 8	WESTINGHOUSE NA-1	3"	22 3/8"	368#	2	878#
9, 10, 11, 12	SESSIONS K	2 1/16"	20 1/8"	252#	4	788#
13, 14, 15	SESSIONS JUMBO	3"	24 5/8"	433#	0	866#
16, 17, 18	CARDWELL G-25-A	2 3/4"	24 5/8"	440#	0	880#
19, 20, 21	CARDWELL G-18-A	3 3/16"	24 5/8"	440#	0	880#
22, 23, 24	MINER A-18-S	2 1/2"	22 3/8"	346#	2	834#
25, 26, 27	MINER A-2-S	2 1/2"	20 1/8"	207#	4	698#
28, 29, 30	NATIONAL H-1	2 1/2"	24 5/8"	428#	0	856#
31, 32, 33	NATIONAL M-1	2 1/2"	24 5/8"	372#	0	744#
34, 35, 36	NATIONAL M-4	2 1/2"	24 5/8"	322#	0	644#
37, 38, 39	MURRAY H-25	2 3/4"	24 5/8"	376#	0	752#
40, 41, 42	GOULD 175	2 1/2"	22 3/8"	337#	2	816#
43, 44, 45, 46, 47	BRADFORD K	2 1/2"	24 5/8"	386#	0	772#
48, 49, 50	WAUGH PLATE	2 1/4"	24 5/8"	420#	0	960#
51, 52, 53	CHRISTY	2 1/2"	22 3/8"	442#	2	1026#
54, 55, 56	HARVEY 2-8"x8" SPGS.	1 7/8"	7 7/8"	104#	8-1 1/2" Followers	670#
57, 58, 59	COIL SPRINGS 2-8"x8"-CLASS G	1 7/8"	7 7/8"	110#	8-1 1/2" Followers	682#

Fig. 1—Identification of Gears in Test

DESCRIPTION OF GEARS

All of the gears in the test, with two exceptions, are of such dimensions as to go into the standard 9⅛ in. x 12⅞ in. x 24⅝ in. draft gear pocket and no gears were included except such as had been developed to the state of being in use, at least to a limited extent, on one or more railroads. To properly identify the several gears a brief description and an analysis of each of the types will be given.

Westinghouse Type D-3

Gears No. 1, 2 and 3

This is the well-known friction draft gear of the Westinghouse Air Brake Company, and is the same gear as applied to 25,000 of the United States Railroad Administration cars. It has a nominal travel in the car of 2-7/16 in. the first ⅝ in. of which is spring travel, the remainder being friction travel. In addition to this the gear is placed in the car under ⅛ in. initial spring compression. Thus as soon as wear equivalent to ⅛ in. coupler movement has occurred, the gear and the friction members will be loose in the car.

The malleable iron friction barrel has a plurality of V-shaped ways on its interior surface and the composite segments or splines, eight in number, are wedged outwardly into these ways to produce frictional resistance against longitudinal movement of the splines. The gear is arranged with a pressure-limiting feature, so that when a predetermined load has been applied to the friction wedge, the follower comes directly into contact with the outer ends of the splines and additional wedging is prevented. On release the friction strips are arranged to be started serially so that sticking will be less likely to occur. No provision is made for taking up wear and lost motion in this gear.

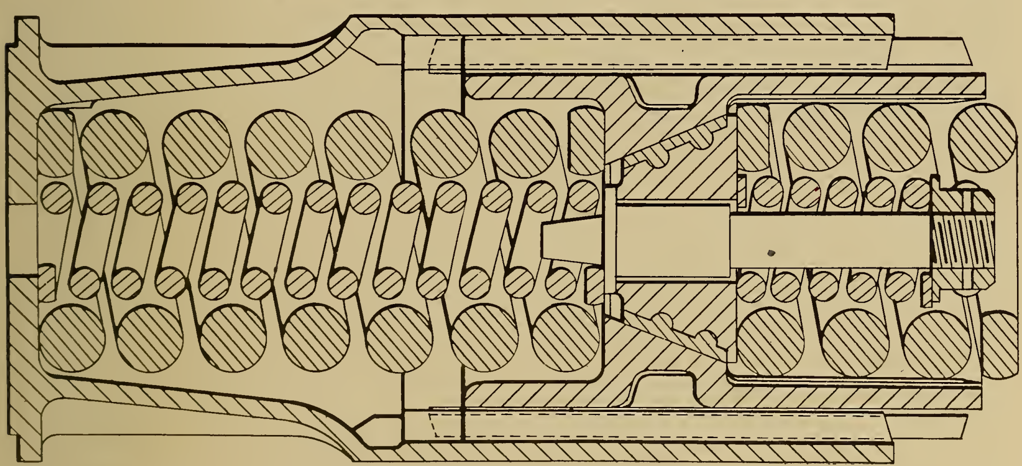


Fig. 2—Westinghouse D-3 Gear

Each gear is made up of a total of 32 parts, 25 of which are subject to wear, one of these being the barrel or housing. Considerable grinding and other machine work is done on this gear so that it may be termed a finished gear. It is not self-contained but can fall apart when dropped from the car, although it is assembled at the factory and shipped and applied as a single unit. A peculiar feature of this gear is that should wear equivalent to ¾ in. coupler movement occur, the wedging action would cease except for a slight amount resulting from the tapered ways of the barrel. After this the gear would be substantially a spring draft gear.

The gear has a friction spring value of 19,500 lb. and an additional independent release spring value of 6,000 lb. The preliminary spring has an active value of 14,800 lb. The friction area of this gear no additional metal being presented for this load.

The nominal length of the gear is 20⅛ in., so that two followers of 2¼ in. thickness are required for it. The gears as furnished weigh on the average about 200 lb. each, or 400 lb. per car, to which must be added for comparative purposes four followers per car, weighing 71 lb. each, giving a per car weight for this gear of 684 lb.

Westinghouse Type NA-1

Gears No. 4, 5, 6, 7 and 8

This new gear of the Westinghouse Air Brake Company is made with a cast steel barrel of rectangular cross-section. The

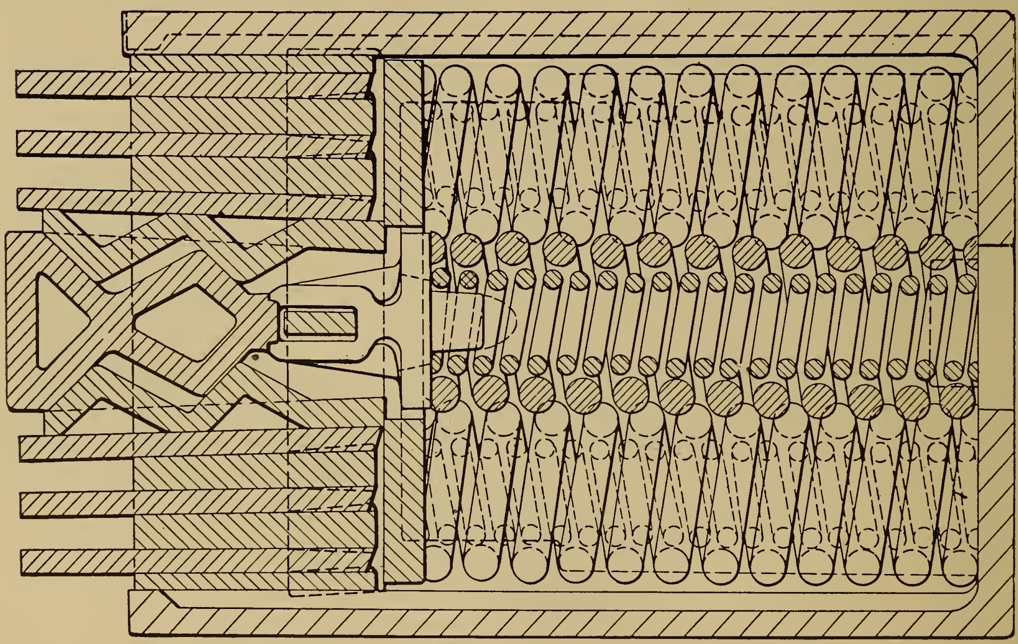


Fig. 3—Westinghouse NA-1 Gear

increases as the gear closes, additional metal of both the moving and the stationary elements coming into contact. The solid blow on this gear is carried by the same metal that resists the frictional movement, friction elements comprise one series of stationary plates and another series of relatively movable plates alternating therewith, the plates being loaded through a set of central wedge members. Four friction springs are used, in addition to one central release spring. All five of the springs are duplicate and the gear has a friction spring value of approximately 17,000 lb. plus an additional release spring value of approximately 4,300 lb.

This gear has an absolute free length of 23-7/32 in. and is held to a compressed normal length of 22⅜ in. by means of a key arrangement. The gear is thus under an initial compression of 27/32 in. in the car. The first 7/64 in. of this is spring compression and the remainder is friction compression. This means that the friction elements can wear an amount equal to 47/64 in. coupler travel before the friction parts of the gear become loose. The capacity of the gear, however, will begin to depreciate as soon as any wear takes place. The gear has a nominal travel in service of 3 in.

The parts of the gear are held, when new, in compressed position by means of the key arrangement and hence the gear is self-contained. Wear of the friction parts, however, will cause the movable friction plates to loosen, so that they can be lifted out of the gear barrel. Considerable fitting and grinding is done in the manufacture of this gear. There are a total of 28 pieces per gear, 12 of which are subject to wear, these latter being small parts, however, and easily renewable. There is no wear on the barrel of this gear and no wear at any point that cannot be compensated for by the insertion of simple plate liners. Any permanent set or shortening of the friction springs will produce loss of capacity and slack and this cannot be taken up by the liners.

The friction area of this gear is constant for all points of its travel, the pressure per square inch increasing as the gear is compressed. On the stationary friction elements the same area is in engagement at all times. The engagement portions of the moving plates change at all points of the travel of the gear. The solid blow comes on the side walls of this gear, at places not highly loaded by the frictional resistance of the gear, so that for the solid blow at least some additional metal is presented.

The gear as manufactured is 22⅜ in. long, so that one 2¼ in. follower is required with it. These gears as furnished weigh on the average about 368 lb. each, or 736 lb. per car, to which, for comparative purposes must be added two followers per car, weighing 71 lb. each, giving a per car weight for this gear of 878 lb.

Sessions Type K

Gears No. 9, 10, 11 and 12

This is the well-known Sessions gear as manufactured by the Standard Coupler Company and as used on 50,000 of the United States Railroad Administration cars. The bellmouth friction box is of drop forged steel, the friction blocks being of cast iron. The spring barrel is a section of steel tubing. The gear has a nominal travel of 2-1/16 in., the first ⅛ of which is spring travel and the remainder friction travel. The gear is put into the car under approximately ⅛ in. initial spring compression, but no friction compression, the friction elements being loose when the gear is first applied. This gear has a friction spring value of approximately 23,000 lb. No separate release springs are used.

Each gear consists of eight parts, four of which are subject to wear. Wear of the parts cannot be taken up except by renewal of parts. The gear is not self-contained but will fall apart when removed from the car. But little fitting or grinding is done in the manufacture of this gear and it is usually shipped loose, to be assembled in the car. The friction area of the gear increases as it is compressed and the solid blow is delivered upon the same metal that receives the friction load.

The normal length of this gear is $20\frac{1}{8}$ in., requiring two $2\frac{1}{4}$ in. followers with each gear. The average weight of one gear is 252 lb., or 504 lb. per car, to which must be added for comparative purposes, four followers per car, weighing 71 lb. each, giving a per car weight for this gear of 788 lb.

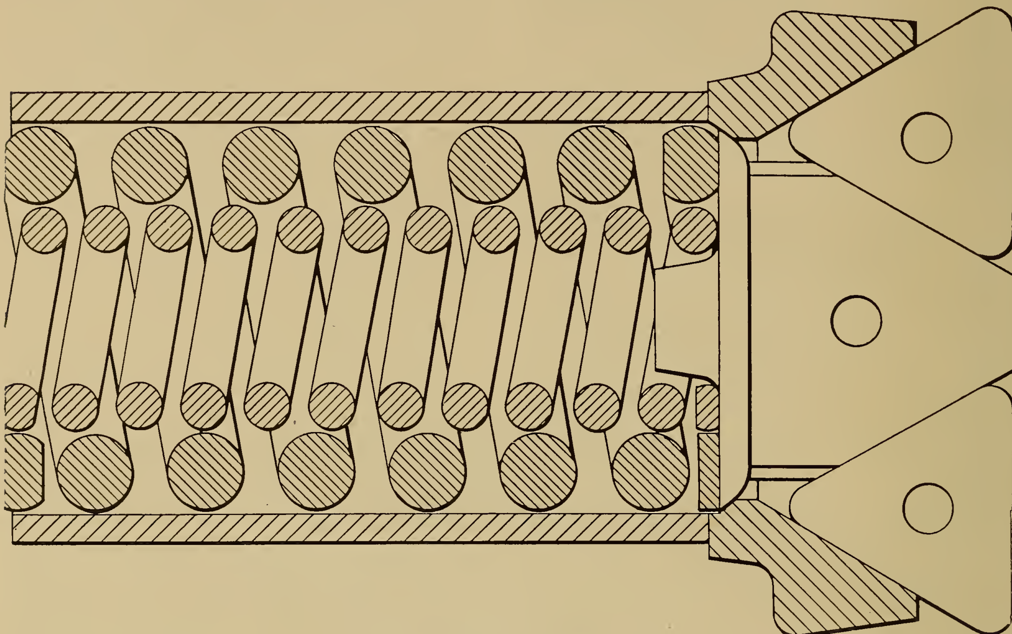


Fig. 4—Sessions Type K Gear

Sessions Jumbo

Gears No. 13, 14 and 15

This is a heavier gear, and of 3 in. nominal travel, recently developed by the Standard Coupler Company. In general, it follows the same principle of wedge blocks as the older Type K gear of this company, there being changes, however, in the angles of the wedge blocks. The gear also includes both followers, being $24\frac{5}{8}$ in. nominal length.

The friction box is of drop forged steel and the spring barrel of cast steel with a closed bottom. There are six double coil friction springs, having a combined value of 30,000 lb. These springs are graduated, the inner coils being shorter than the outer coils.

The friction box, spring barrel, spring plate and center friction block are held together as a self-contained unit by means of a rivet and key arrangement. The side friction blocks and the follower are loose, however, so that the gear is not entirely self-contained.

Each gear has a total of 22 parts, five of which are subject to wear; three of these are the cast iron friction blocks, the other two being the drop forged friction box and the drop forged follower. The gear has but little fitting or grinding done on it. The solid blow is taken by the same metal that carries the friction load, and wear will slightly reduce the value of the gear to resist solid blows. The friction area of this gear increases slightly as the gear is compressed.

The free length of this gear is 24¾ in., hence the gear is put in the car under ⅛ in. initial compression, all of which is spring compression, the friction elements being loose when the gear is first applied to the car. Of the 3 in. gear travel, when new, the first 3/16 in. is spring travel, at which point the friction blocks first become tight. The remainder is friction travel. The average weight of one of these gears is 433 lb. and as no extra followers are required the comparative per car weight is 866 lb.

Cardwell Type G-25-A
Gears No. 16, 17 and 18

This is the regular pattern Cardwell gear of the Union Draft Gear Company, but with the parts slightly modified to give a nominal travel of 2¾ in. It is used on 19,000 of the United States Railroad Administration cars. The gear is of the double end type, the two friction casings, sometimes termed "housings" or "followers," being of malleable iron. There are seven contained friction members of cast iron. The customary transverse spring arrangement is used, with malleable iron spring-seat nuts threaded on the ends of the spring rod. The free length of this gear is 25-11/16 in. as against a pocket length of 24⅝ in., so that the gear as assembled in the car is under an initial friction compression of 1-1/16 in. This means, in other words, that the gear can wear an amount equal to 1-1/16 in. coupler travel before actual lost motion in the gear occurs. Of course, the ultimate resistance of the gear as well as its capacity will have been reduced, but it is possible to recover this in a large measure by adjusting the exposed spring-seat nuts. There is in addition to this an initial spring compression of ⅜ in. so that each spring

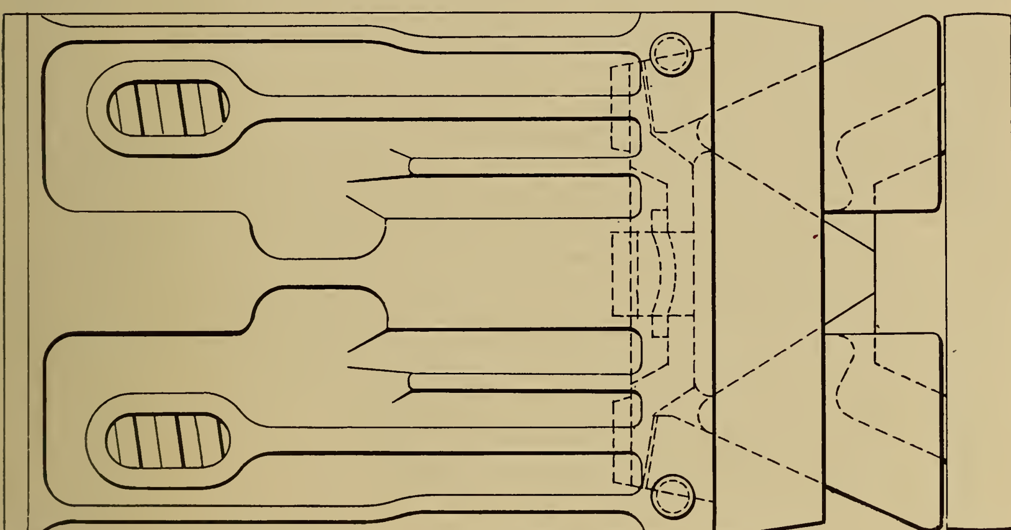


Fig. 5—Sessions Jumbo Gear

can, in addition to the wear above noted, take a permanent set of 3/16 in. before the friction elements become loose on the spring rod. The magnitude of the initial compression of this gear gives a high starting resistance and a stiff compression curve at the beginning of the gear travel.

The gear has no independent release springs and the friction springs have a value of approximately 29,900 lb. When a solid blow comes on this gear some additional metal is presented to receive it. The friction casings, which alone receive the solid blow, are castings with rather thin walls. There are a total of 20 parts per gear, nine of which are subject to wear. Seven of the wearing members are of cast iron and two are malleable iron, the latter being the main friction casings or followers. This gear is not self-contained but must be built up in the car. It is probably the most difficult of the gears to apply. All of the parts are rough with but little grinding or fitting done to them. The normal length of this gear is 24⅝ in. and no followers are needed. The average weight of one gear is 440 lb., giving a comparative per car weight of 880 lb.

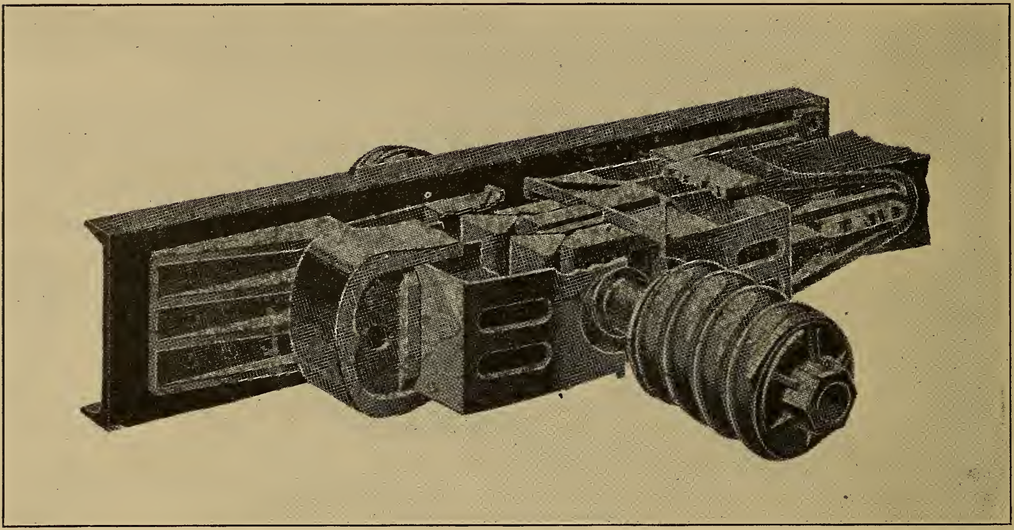


Fig. 6—Cardwell Type G-25-A Gear

Cardwell Type G-18-A
Gears No. 19, 20 and 21

This is the regular Cardwell gear of the Union Draft Gear Company, designed to fit in the standard 24⅝ in. draft gear pocket and of 3-3/16 in. nominal travel. The remarks in general concerning the Cardwell Type G-25-A gear are applicable to this gear also. Each gear has a total of 20 parts, nine of which are subject to wear, seven of these being of cast iron and the other two being the main malleable iron heads or followers.

The gear has a free length of 25¼ in. as against a pocket length of 24⅝ in. so that the gear, as assembled in the car is under an initial friction compression of ⅝ in., meaning that when wear equivalent to ⅝ in. coupler travel occurs the friction elements become loose in the car. The springs are in addition under a combined initial compression of a ⅜ in., or 3/16 in. per spring. The value of the friction springs is approximately 29,900 lb. The nominal length of this gear is 24⅝ in. and no followers are required. The average weight of one gear is 440 lb., giving a comparative per car weight of 880 lb.

The relative performance of this gear and of the Cardwell G-25-A should be of interest inasmuch as the only difference in the two gears is in the length of the travel. All of the parts of both gears are the same except the two heads or followers and these are designed in the case of the G-25-A gear to take up the first 7/16 in. of travel as compared with the G-18-A gear, giving heavier initial compression but leaving the ultimate resistance practically the same for both gears. The G-25-A, therefore has a reduced travel but higher starting resistance. It may possibly show a very slight loss in capacity due to this but on the other hand has the advantage of having the friction elements held in positive engagement during a longer period of wear. Whether or not high initial resistance prevents wear that may otherwise occur from the multitude of slight movements of the easier moving gear may also be indicated by service tests of these two gears.

Miner Type A-18-S

Gears No. 22, 23 and 24

This is a slightly modified arrangement of the well-known A-18 gear of W. H. Miner and is the design as applied to United States Railroad Administration locomotive tenders. The location of the friction shoes has been changed as compared with the A-18 gear. The present gear has a nominal travel of 2½ in.

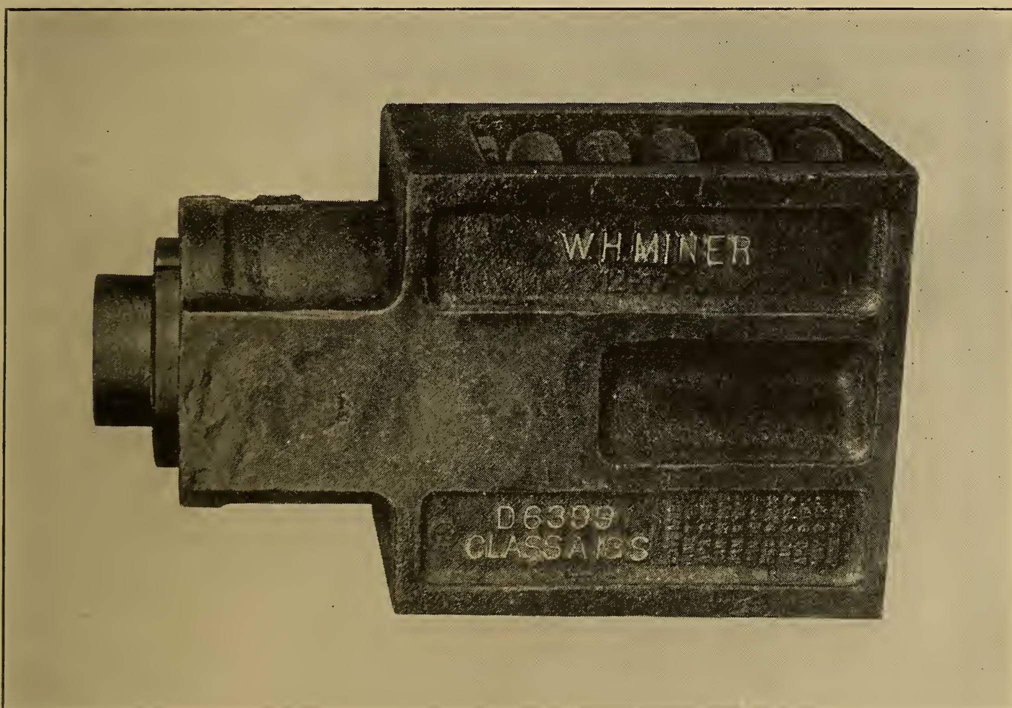


Fig. 7—Miner Type A-18-S Gear

The barrel of the gear is of malleable iron and contains, by an interlocking arrangement, the two double coil friction springs and malleable iron spring plate or follower. The regular drop forged, hardened friction shoes, three in number, are used, with the central wedge of cast steel. The Miner rollers, three in number, and of 1 in. diameter tempered tool steel, are interposed between the central wedge and the friction shoes to allow greater friction pressures with possibly no greater tendency of the gear to stick. The entire friction pressure is transmitted through these rollers.

As applied to the car the main springs are under an initial compression of ⅞ in. and the preliminary spring of ¾ in. The function of this preliminary spring should not be confused with purely spring gear action, as the A-18-S gear starts off immediately as a friction gear of high initial resistance. Inward movement of the friction shoes is resisted first by the preliminary spring and subsequently by the main spring. Wear will increase the movement of the friction shoes upon the preliminary spring and decrease the movement upon the main springs. The travel of this gear should remain practically constant, irrespective of wear, and as wear occurs the friction shoes, which in the new gear extend ¾ in. outside of the friction barrel, will protrude farther because of the spreading action resulting from the preliminary spring. This will continue until wear equivalent to ⅜ in. coupler movement occurs when the friction shoes will extend 1⅛ in. outside of the barrel and the shoes will then loosen. Up to this point however, the full travel of the gear will be realized as friction travel, although the capacity and ultimate resistance of the gear will be reduced. It should be possible, however, to compensate for wear by inserting one or more ring washers between the inner ends of the friction shoes and the spring cap or followers, thereby recovering the movement upon the main springs and restoring the original capacity.

The gear has a main spring value of approximately 42,000 lb. and in addition a preliminary spring value of approximately 5,300 lb. It is held to the correct length and as a self-contained unit by means of a single ⅞ in. retaining bolt. The gear has a total of 18 parts, four of which are subject to wear, one of these being the main barrel or cylinder. Wear on this part will reduce its ability to withstand solid blows. The friction area of this gear increases as the gear is compressed. The gear has considerable grinding and fitting done to it during its manufacture.

The normal length is 22⅜ in. so that two followers are required per car. The average weight of one gear is 346 lb., to which must be added for comparative purposes the weight of the two followers, giving a comparative per car weight of 834 lb.

Miner Type A-2-S

Gears No. 25, 26 and 27

This is a slightly modified arrangement of the A-2 gear of W. H. Miner, the nominal travel being 2½ in. The gear has the regular malleable iron cylinder with three hardened, drop forged friction shoes and a single cast steel central wedge, the customary rollers of the Miner design being interposed between the central wedge and the friction shoes. One double coil friction spring is used. The rollers, three in number, are of tempered tool steel 1 in. in diameter by 3 in. long. The rollers in this gear, as in the A-18-S gear, are not directly cushioned by the springs, but receive the entire friction pressure.

The absolute free length of this gear is 21 in., but it is held compressed to its normal length of 20⅛ in. by the retaining bolt. The gear is thus under an initial friction compression of ⅞ in. Before this much wear could occur, however, or if wear equivalent to ⅜ in. of coupler movement should occur, the inner end of the central wedge would strike the spring cap, and the gear would then become purely a light capacity spring gear and further wear would be arrested. In this gear, as in the A-18-S, the total travel of the gear can never be reduced by wear, although the capacity and ultimate resistance will be decreased. The friction shoes will also extend farther out of the barrel as wear progresses.

The gear has a total of 13 parts, four of which are subject to wear, one of these being the main barrel or cylinder. The friction area of this gear increases as the gear is compressed. The solid blow is taken upon the same metal that receives the friction load and wear will materially weaken the cylinder for taking care of the solid blow. The gear is self-contained so that it is applied to the car as a single unit. This gear has a friction spring value of approximately 22,800 lb. It is also fitted and bulldozed during the process of manufacture. The average weight of one of these gears is 207 lb. and there are required two followers with each gear, weighing 71 lb. each, giving a comparative per car weight of 698 lb.

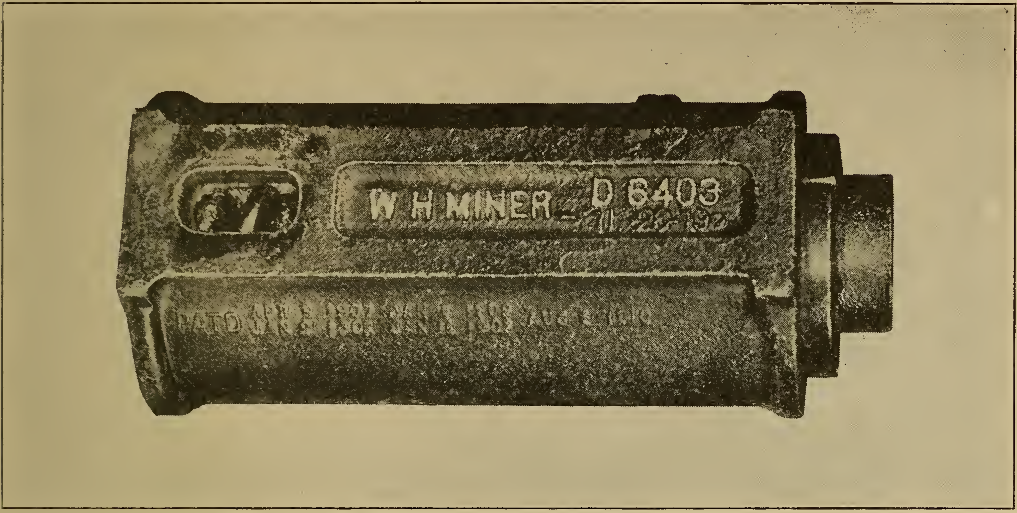


Fig. 8—Miner Type A-2-S Gear

National Type H-1

Gears No. 28, 29 and 30

This is a new gear of 2½ in. nominal travel, manufactured by the National Malleable Castings Company. A central friction column with four ways in it is cast integral with the one follower of the gear. In these ways are four friction segments or shoes. The other, or movable follower, is arranged to wedge these shoes inwardly into the ways of the column and as the gear is closed the longitudinal movement of the shoes is resisted by a single coil friction spring that surrounds the friction column below the wedges. Four independent corner posts of 1⅝ in. diameter steel are provided to receive the solid blow so that this force is received on entirely different metal. An independent release spring surrounds each of these corner posts. The gear is held to any desired length and as a self-contained unit by means of two ¾ in. rods with castle nuts.

All of the principal parts of this gear, including the friction members, are made of Naco Electric steel, the corner posts being of tempered knuckle pin steel. All of the friction members are hardened. The gear has a friction spring capacity of approximately 29,200 lb. and an additional release spring capacity of approximately 16,000 lb. The absolute free length of the gear is 24-25/32 in. so that it is put into the car under 5/32 in. initial compression, all of which is friction compression. The gear can thus wear an amount equal to 5/32 in. travel or the spring take a set of 5/32 in. before the friction shoes become loose in the car. The capacity of the gear, however, will begin to depreciate as soon as any wear takes place.

An interesting feature of this gear is that on release, the first action is a tendency to shift the friction shoes outward from their engagement with the center friction column, thus allowing greater pressures with possibly no greater tendency to stick. This is accomplished by having the bearing of the shoes upon the spring seat at a subtracting angle. Bronze pressure pads are provided for the contact spots on the spring seat and the outer head. These are not subject to wear, but to pressure only. The outstanding feature of this gear is that the friction elements are wedged inwardly, the outward reactions all being included in the box-shaped movable follower. There is no wear upon this member and wear should not noticeably affect the strength of the gear. Wear can be taken up by means of ring washers beneath the friction spring.

The friction area of this gear is constant, the pressure per square inch increasing as the gear is compressed, the entire bearing surface of the friction blocks sliding along the ways or flutes in the center column. This gear has a total of 26 pieces, 5 of which are subject to wear, one of these being the main center column. Considerable grinding, fitting and working constitute a part of the manufacture of this gear and it may be termed a finished gear.

The normal length is 24⅝ in., so that no followers are required. The average weight is 428 lb. or a comparative per car weight of 856 lb.

National Type M-1
Gears No. 31, 32 and 33

This gear is similar in construction to the National Type H-1, the most noticeable difference being that but two release springs are used instead of four as in the H-1 gear. Otherwise the same description of parts, materials, and operation serves for both gears. The nominal gear travel is 2½ in.

This gear has a friction spring value of 16,700 lb. and an additional release spring value of 9,100 lb. The free length is 25⅛ in. so that it is put into the car under ½ in. compression, the first 5/16 in. of which is spring compression, the remainder, 3/16 in., being friction compression. Thus the gear can wear an amount equal to 3/16 in. coupler travel before the friction shoes become loose in the car. There are a total of 26 pieces per gear, five of which are subject to wear and one of these being the main center column. As in the H-1 gear, the wearing surfaces are of hardened steel and are constant in area.

The normal length is 24⅝ in., no followers being required. The average weight per gear is 372 lb. or a comparative per car weight of 744 lb.

National Type M-4

Gears No. 34, 35 and 36

This gear is of the same general construction as the two preceding National gears but has three flutes in the center column with three friction shoes, and has no independent release springs. Otherwise the general description is the same as heretofore given for the other National gears. The nominal travel is 2½ in.

The gear has a spring value of 25,000 lb. The absolute free length is 25-3/16 in. so that it is put into the car under 9/16 in. compression, all of which is friction compression. This means that the gear can wear an amount equal to 9/16 in. coupler travel before the friction shoes become loose. It is held to its normal compressed length of 24⅝ in. by means of two ¾ in. rods with castle nuts. The gear has a total of 17 pieces, five of which are subject to wear, one of these being the main center column. As in the other National gears, the wearing surfaces are of hardened steel and are constant in area.

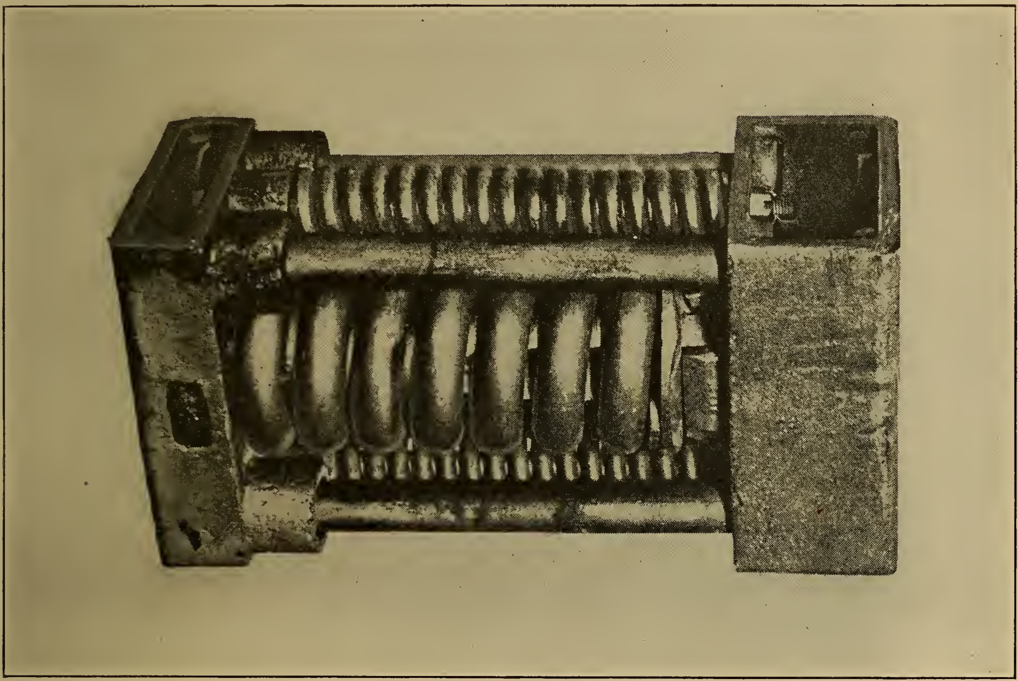


Fig. 9—National Type M-1 Gear

The normal length is 24⅝ in., no followers being required. The average weight per gear is 322 lb. or a comparative per car weight of 644 lb.

Murray Type H-25

Gears No. 37, 38 and 39

This gear is of the regular Murray pattern, without wear blocks, as manufactured by the Keyoke Railway Equipment

Company, but has been specially designed to give a nominal travel of 2¾ in. for use on 6,000 of the United States Railroad Administration cars. The followers, the side wedges and the cam blocks are of cast steel and are in general of sturdy design. the triple coil friction spring has a value of approximately 43,000 lb. These gears as included in the tests had no provision for taking up wear, but it is understood that a similar type is made with renewable wear blocks.

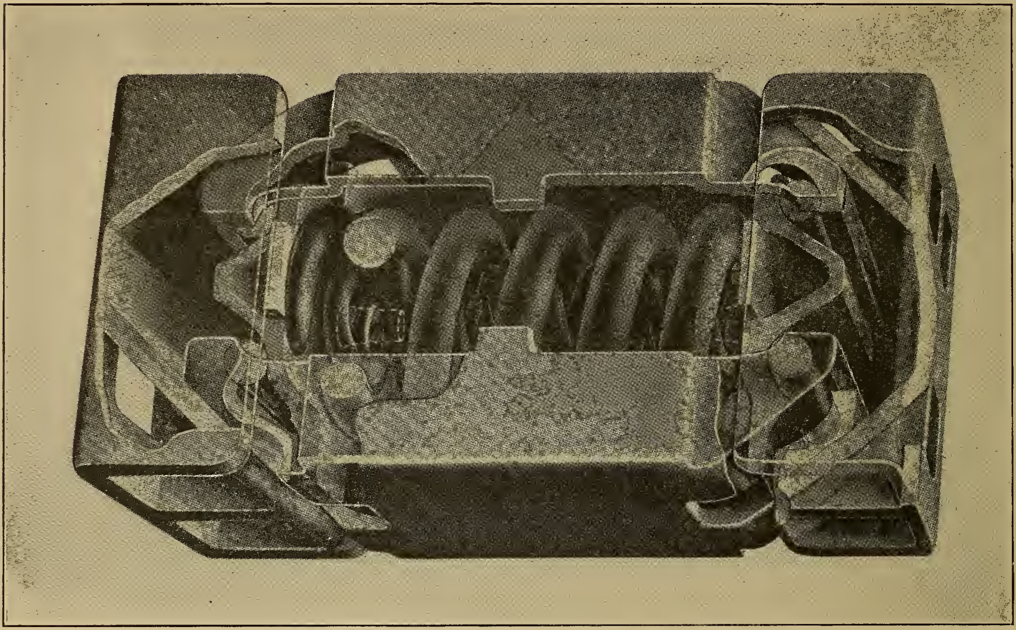


Fig. 10—Murray Type H-25 Gear

This may be termed a double end gear, inasmuch as there are friction elements in series at each end.

The longitudinal movement of the heads or followers produces a corresponding inward movement of the side bars or wedges, this latter movement, through the four rollers and two cam blocks, producing an endwise compression of the friction spring. The rollers, which actually rotate during a compression of the gear, can never be loaded beyond the capacity of the spring, unless the spring should go solid before the limit of normal gear action is reached. No separate release springs are used and

Each gear has a total of 13 parts, four of which are subject to wear, these being the four largest parts, *viz.*, the cast steel end heads and the cast steel side bars. The friction area of this gear increases as the gear is compressed.

The free length is 24-15/16 in. as against a pocket length of 24⅝ in., so that the gear as assembled in the car is under an initial friction compression of 5/16 in. This means that the gear can wear an amount equal to 5/16 in. coupler travel before actual lost motion in the gear occurs. The ultimate resistance and the capacity of the gear will, however, have been reduced by such wear. In addition to this the spring is held under an initial compression of 11/16 in. The solid blow in this gear is taken by the same parts that receive the friction load and the bearing surfaces between the followers and the side wedges for the solid blows are not as extensive as in some other gears. The gear is not self-contained and requires some special fitting up and manipulation to apply it to the car, and some special apparatus is required to assemble the spring and rollers in their inter-locked positions between the side wedges. The parts of the gear as finished are rough with but little grinding or fitting.

The normal length is 24⅝ in. and no followers are required. The average weight of one gear is 376 lb., giving a comparative per car weight of 752 lb.

Gould Type 175

Gears No. 40, 41 and 42

This gear of the Gould Coupler Company has a cast steel barrel with a rectangular bellmouth for the cast steel friction wedges. These latter, two in number, are case hardened and are pressed outwardly against the friction faces of the barrel by a friction spring of leaf type, which is made up of two half-elliptic springs placed back to back between the wedges, each half being composed of eight plates, 3/16 in. by 7 in. by 8¼ in. long. This spring is applied under a slight initial compression but not enough to compensate for wear of any moment. The gear is supplied in addition with a double coil release spring of 38,000 lb. value. The gear is applied to the car without initial compression. It has a nominal travel of 2½ in.

There are a total of 22 parts to each of

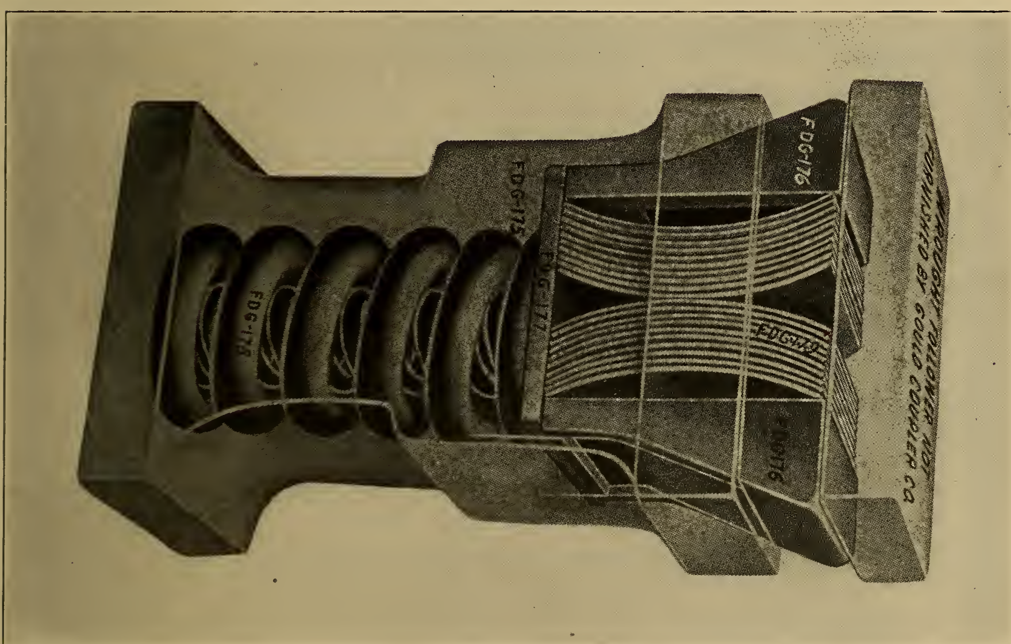


Fig. 11—Gould Type 175 Gear

these gears, four of the parts being subject to wear, one of these being the main cast steel barrel or housing. Wear can be readily taken up by the insertion of liners back of the friction wedges. The solid blow is delivered upon the same metal that receives the friction load, and wear also tends to weaken the gear. The friction area is constant. The gear is self-contained when new, but wear will shortly loosen the parts so that it will fall apart when removed from a car.

The normal length is 22⅝ in., requiring a 2¼ in. follower with each gear. The average weight of one gear is 337 lb., to which must be added for comparative purposes two followers per car, weighing 71 lb. each, giving a per car weight for this gear of 816 lb.

Bradford Type K

Gears No. 43, 44, 45, 46 and 47

This gear is manufactured by the Bradford Draft Gear Company. It is of the rocker type, having malleable iron heads or spring housings at each end with two pairs of inter-engaging, rotative knuckles between the springs. The action is such that each spring is compressed between one of the heads and the adjacent face of the rocker. Friction is obtained by one rocker sliding upon another and also by the rockers rotating in seats in the housings. The gear has a nominal travel of 2½ in., but is put in the car under ¼ in. initial spring compression. The friction elements are just tight when the gear is put into the standard pocket.

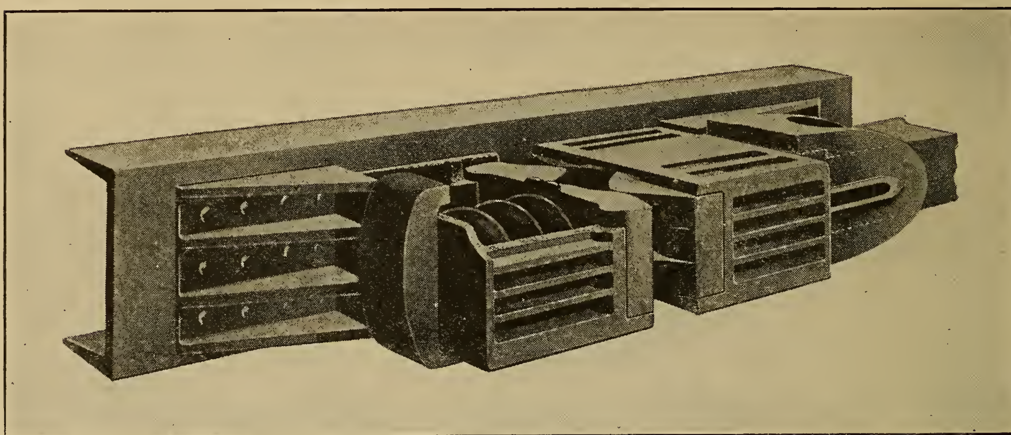


Fig. 12—Bradford Type K Gear

There are a total of 10 pieces to each gear, six castings and four springs. It is noticeable that every piece of the gear, except possibly the springs, is subject to wear. Upon a strict analysis the rocker ends of the springs might even be included, as the rockers move across the end faces of the springs. The friction area is constant and where one rocker slides upon another there is practically line contact. The solid blow is taken by the same metal that receives the friction load. The gear is not self-contained but must be assembled in the car. The two friction springs are regular A. R. A. Class G springs, working in series, and the spring value of the gear is accordingly 30,000 lb. The parts of the gear as furnished are rough.

The normal length is 24⅝ in. and no followers are required. The average weight of one gear is 386 lb. or a comparative per car weight of 772 lb.

Waugh Plate Type

Gears No. 48, 49 and 50

This is the well-known plate gear of the Waugh Draft Gear Company. As included in the tests each gear was made up of four sets of plates in series, each set consisting of 15 spring steel plates ¼ in. by 6 in. by 11⅞ in. Half oval followers of cast steel are supplied at each end, and two separators and one full oval complete the gear proper. In addition, however, two guide plates or wear plates, are supplied for each gear, these being bolted or riveted to the draft sills to fill out the 12⅞ in. sill spacing and to hold the parts of the gear in alignment.

The nominal length is 24⅝ in. and the nominal travel 2¼ in. The gear has a friction area of great extent and it is hardly probable that wear would ever materially reduce the travel or capacity. If the spring plates should take any permanent set, however, the travel and capacity of the gear would be decreased. Every gear has a total of 65 parts, or 67 parts including the wear plates.

In this gear it is difficult to give a relative per car weight because of the difference in yoke dimensions required. Each gear weighs 420 lb. without the two wear plates, which latter will weigh about 30 lb. each. Yoke spacers should then be added, so that a comparative per car weight of 960 lb. has been allowed for this gear.

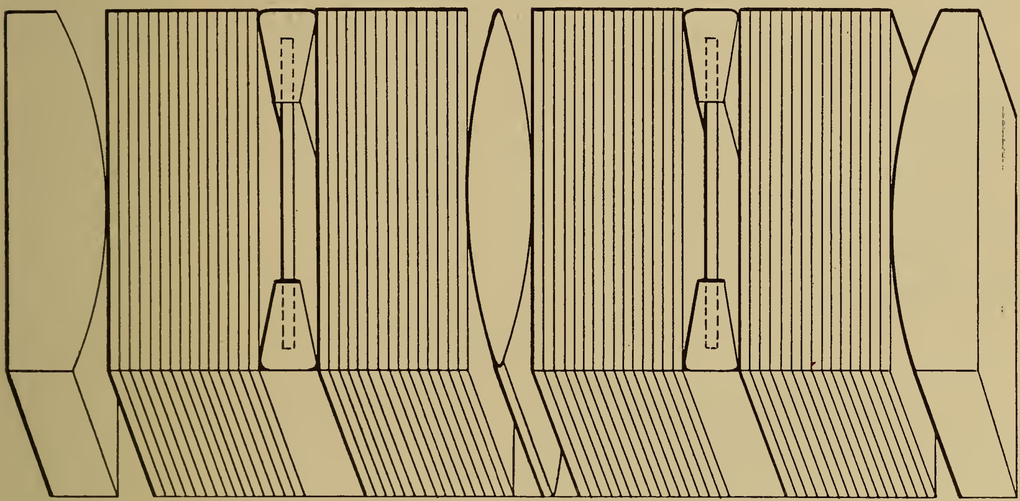


Fig. 13—Waugh Plate Gear

Christy

Gears No. 51, 52 and 53

This gear is under development by the American Car Roof Company. It had not, up to the time of the beginning of the tests, been developed to a commercial stage and has been included in these tests only upon the request of the mechanical department of one of the railroads. The gear, which has a nominal travel of 2½ in., follows in general the better-known Sessions principle of wedge blocks, except that the center block is made in halves with a roller between them to form a fulcrum. Wear is to be compensated for by using a roller of a larger size.

The outstanding feature, and the point wherein it differs from all other gears in the test, is that the frictional resistance of the gear is compounded. In most draft gears the friction movement is obtained, and to a greater or less degree the frictional resistance is controlled by the direct compression of springs. In this gear the outer, or main friction members, are resisted, not by the spring directly but by other friction members, and these latter are then resisted by the spring itself, the frictional resistance being thus multiplied. This should result in a gear of very high resistance but may also result in uncertain and uncontrolled resistance.

The frictional resistance of the gear is thus compounded by having the inward movement of the halves of the center wedge block seat upon and expand the additional pair of friction shoes which press upon the inner faces of the spring portion of the barrel. These last named friction shoes rest upon and compress the friction spring which is graduated, the inner coil being of 2½ in. less length than the outer coil. The friction spring has a total value of 27,000 lb. No separate release springs are used. The friction box and spring barrel are in one piece, of cast steel, with a removable head bolted on the spring end of the barrel. All of the parts of this gear are exceedingly heavy, the walls of the spring barrel, for example, being of 1 in. stock. The gear has a total of 28 parts, 8 of which are subject to wear, among these being the main cast steel barrel or housing. The gear is not self-contained but the friction members can fall out when the gear is removed from the car. The solid blow is taken upon the same metal that receives the friction load and wear will reduce the strength and value of the gear to resist solid blows. The friction area is practically constant, although some new surfaces are constantly coming into bearing and others going out of bearing as the gear is compressed.

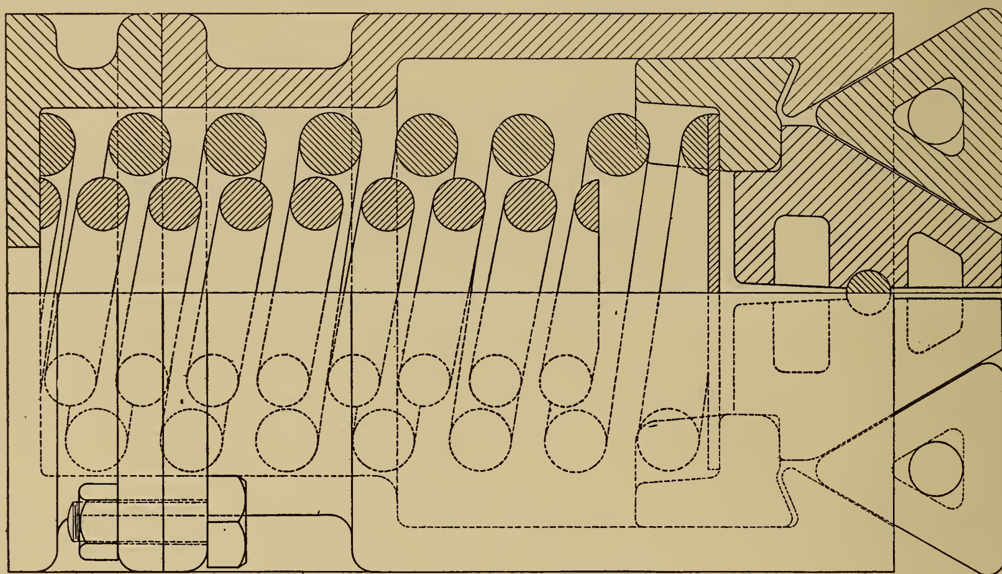


Fig. 14—Christy Gear

The absolute free length is 22-7/16 in. as against a pocket length of 22⅝ in., so that the gear in the car is under but 1/16 in. initial compression. Upon very slight wear, therefore, or set of the springs, the friction members will be loose. The average weight of one of these gears is 442 lb. and having a nominal length of 22⅜ in. there must be added to this the weight of two followers per car, giving a comparative weight of 1026 lb. per car.

Harvey Friction Springs
Gears No. 54, 55 and 56

These are the regular interwound Harvey friction springs as manufactured by the Frost Railway Supply Company. Each gear, as included in the tests, consisted of two of these springs set in twin fashion, side by side. The free height of each interwound spring group is 8 in., and so wound as to allow 2 in. of movement from this height, thus having a nominal travel in the car of 1⅞ in. Each group has a plain centering

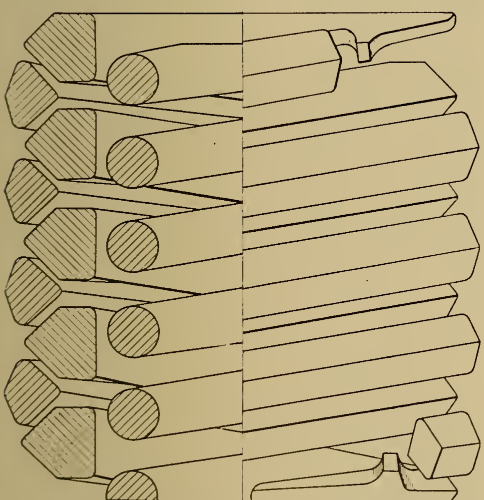


Fig. 15—Harvey Friction Springs

coil of ⅞ in. diameter bar wound on a 2⅝ in. diameter mandrel and of 7½ in. free height. In receiving the solid blow the main, or inner, of the two specially shaped friction coils goes solid. This bar is made with flattened contact faces to receive the solid blow.

This type of gear will not work in the standard pocket without special housings. The average weight of a group of these springs is 52 lb. or 208 lb. per car. It is difficult to give a comparative per car weight but in order to compare the arrangement with the other gears of the test there has been added eight followers, each 9 in. by 12 in. by 1½ in., weighing 45 lb. each, two yoke abutments, weighing 40 lb. each, and four rivets for the yoke abutments, weighing 5½ lb. each, giving a comparative per car weight of 670 lb.

A. R. A. Class G Springs
Gears No. 57, 58 and 59

Regular A. R. A. Class G draft springs drawn from ordinary railroad stock have been included in the tests. Each gear as numbered above was composed of two complete inner and outer coil springs, tested in twin fashion.

The details of each spring group are as follows:

Outer Coil

1-9/16 in. diameter bar.
8 in. outside diameter coil.
7⅞ in. free height.
5¾ in. solid height.

Inner Coil

1 in. diameter.
4⅝ in. outside diameter coil.
7½ in. free height.
5¾ in. solid height.

Each group has thus a possible deflection of 2⅛ in. at a load of 30,360 lb. or a deflection of 2⅛ in. per gear at a load of 60,720 lb. The average weight of a group is 55 lb. or 110 lb. per gear. To this is added for comparative purposes the same parts as for the Harvey springs, giving a comparative per car weight of 682 lb.

SELECTION AND CONDITION OF TEST GEARS

At the beginning of these tests the various manufacturers were asked to furnish gears for test purposes, so that the gears as tested were in each instance procured directly from the proprietor, with full knowledge on his part that they were for test purposes. Whether or not gears of average manufacture were furnished must be decided from previous or additional experience with the several gears and from a knowledge of the manufacturing practices of the concerns. Unless a definite statement to the contrary appears in this report it is to be understood that gear conditions and performances as developed during the tests are in accordance with what is believed to be average conditions.

Immediately upon receipt of a test gear it was given a test number and then taken apart. The parts were marked, and measured for comparison with the manufacturer's drawings and for later comparative tests measurements. The gears were reassembled with the parts in their original positions and were given a definite amount of preliminary drop test work to condition them for the regular tests.

Westinghouse D-3

Gears No. 1, 2 and 3

These gears as received were in good average condition and conformed very closely to the dimensions as given on the manufacturer's drawings. The gears had not been built up of maximum dimension parts to produce unusual capacity. The customary practice of machining and grinding certain parts had been followed and the gears had been worked in the bulldozer as is the regular practice in their manufacture. They showed also slight indication of drop test work but not an excessive amount. The results obtained in the tests agree very well with results obtained in other tests of the same gear, particularly in routine acceptance tests of gears for United States Railroad Administration cars.

Westinghouse NA-1

Gears No. 4, 5, 6, 7 and 8

These gears do not have as much machine work done on them as in the case with the Westinghouse D-3 gear, but are carefully fitted and assembled. The gears as received appeared to be in average condition and for the rougher character of the work, agreed very closely with the drawings. The gears had been bulldozed and had undoubtedly been under the drop testing machine. The bulldozing, it is understood, is a regular process in their manufacture and the drop test work had not been extensive. The gear parts were not over size and the results of the tests in general are believed to be representative of the action of the average product.

Sessions K

Gears No. 9, 10, 11 and 12

These gears are furnished commercially with but little finishing, it being the manufacturer's practice to gage the parts and grind the friction blocks when necessary to bring them to gage or to smooth up the bearing surfaces. The gears as received represented average workmanship and conditions and showed evidence of having been under the drop machine for a few movements. The results of the tests in general are comparable with previous tests of the same gear, particularly in routine acceptance tests of gears for United States Railroad Administration cars.

Sessions Jumbo

Gears No. 13, 14 and 15

These gears as received represented average workmanship and condition. They showed slight evidence of having been under the drop test machine for a few movements at some previous time, although the friction surfaces had a light coating of rust on them when received. The results of the tests are believed to be representative of the commercial gear.

Cardwell G-25-A

Gears No. 16, 17 and 18

These gears as received were in average condition as to workmanship and showed indications of having been under the drop machine. The springs furnished with the test gears were of excessive length, the average free length being 10-1/16 in., whereas the drawing dimension is but 9½ in. With all the parts properly assembled on the spring rod, the springs from the drawings should be under 3/16 in. compression while with the gears as finished the springs were under ⅝ in. compression. When assembling the gears in the frame for testing, with a pocket of the same length as in the car, it required the extreme efforts of two men working on an eight-foot wrench to screw up the spring nuts. It is noted also that the average drop test results obtained from these gears are greater by slightly more than 4 in. than the average results obtained in routine acceptance tests of the same gears for United States Railroad Administration cars, whereas with all other gears used on United States Railroad Administration cars the average of the test gears was lower than the average of the commercial gears. The lowest capacity gear of this type in the present tests was more than 3 in. greater than the highest capacity gear of the same type found in the United States Railroad Administration acceptance tests. It is therefore believed that the results obtained for these test gears are not representative of what may be expected from the regular product as furnished commercially.

Cardwell G-18-A

Gears No. 19, 20 and 21

These gears were received in average condition as to workmanship, and the parts conformed more closely to the drawings than in the case of gears number 16, 17 and 18, although they averaged above the drawings. The individual variations, however, would probably be accepted as within manufacturing limits. The averages are believed to more nearly represent the true value of the commercial gear than those obtained from test gears number 16, 17 and 18. These gears were submitted near the close of the test program.

Miner A-18-S

Gears No. 22, 23 and 24

These gears as received were in good average condition as to workmanship and material and the parts conformed closely to the dimensions as given on the drawings. They showed evidence of having been given some slight work, at least in the bulldozer, this being a part of the regular process of manufacture. The results obtained in these tests are in harmony with those of other tests and the gears as tested are believed to be representative of the commercial product.

Miner A-2-S

Gears No. 25, 26 and 27

The condition of these gears as received corresponds with that of the Miner A-18-S and the test gears are believed to be representative of the commercial gears of this type.

National H-1

Gears No. 28, 29 and 30

These gears as received conformed closely to the drawings' dimensions, and the results obtained are comparable with results obtained in other tests of this gear. They showed evidence of having been worked under a drop machine or in a bulldozer, the latter being a regular operation in the manufacture of the gear. The results of the test are believed to be representative of the gears as furnished commercially.

National M-1

Gears No. 31, 32 and 33

These gears as received were in the same general condition as those of the National H-1 type and the results, which conform to results of other tests, are believed to be representive of the commercial product.

National M-4

Gears No. 34, 35 and 36

The condition of these gears as received corresponds with that of the other National gears and is believed to be representative of the commercial product.

National M-4

Gears No. 34, 35 and 36

The condition of these gears as received corresponds with that of the other National gears and is believed to be representative of the commercial product. These gears were submitted near the close of the test program.

Murray H-25

Gears No. 37, 38 and 39

These gears as received were in average condition except that they had been given considerable work under the drop machine. In one case the friction surfaces were badly galled and scored. While the Murray gear is furnished commercially of rough castings and while these test gears had probably been given more conditioning than any other gears in the test, yet the results are not believed to have been influenced by it, especially as they are just slightly below the average of routine acceptance tests of the same type of gears for United States Railroad Administration cars.

Gould 175

Gears No. 40, 41 and 42

These gears as received conformed closely to the manufacturer's drawings and appeared to be in good average condition except that a coating of grease was found in the interior of the gears, upon the top surfaces of the wrought steel follower plates that rest upon the main coil springs. The bottom ends of the friction wedges, as well as the lower ends of the leaf springs, bear upon the top surface of this plate and have a lateral motion thereupon. The main friction surfaces were free from grease. This condition was reported to the manufacturers, who disclaimed all knowledge of the presence of the grease, and at their direction the parts were cleaned and the gears placed in a condition satisfactory to their representative, who inspected them upon invitation. These test gears had been given some slight preliminary work but not immediately before shipment, as one of the gears had a light deposit of rust upon the friction surfaces. The results of the tests are believed to be representative of the action of the commercial product.

Bradford K

Gears No. 43, 44, 45, 46 and 47

The undeveloped state of this gear makes it impossible to compare the test gears with the commercial product. The housings showed porosity and contained numerous small checks. A. R. A. Class F springs were sent in mistake for the Class G springs called for in the drawings. The gears were accordingly set up with Class G springs drawn from regular railroad stock. Several variations from the drawings were found. These gears are to be furnished commercially of rough castings, without any bulldozing or other working, and the test gears as received were in this condition, never having been operated before shipment.

Altogether, the test results from these gears are not satisfactory. It is felt that avoidable defects in workmanship and design are responsible, at least in part, for the breakage of gear parts that will be noted as the report proceeds.

Christy

Gears No. 51, 52 and 53

This is an undeveloped gear which has never been furnished commercially, so that comparisons are impossible. It is understood that the gears are designed to be furnished regularly of rough castings. The test gears, however, had all of the friction surfaces machined and almost the entire external surface of the barrel had been shaped off to give true surfaces and correct dimensions. The springs averaged $\frac{1}{2}$ in. less in length than called for on the drawings and the gears themselves averaged approximately $\frac{3}{8}$ in. less in length than the drawing dimensions, so that $\frac{3}{4}$ in. of free slack would have been present in each car with these new gears applied. The gears also had $\frac{3}{8}$ in. less of travel per gear than called for on the drawings. The drawing dimensions for the roller for the center wedge block are 1 in. in diameter by $6\frac{3}{4}$ in. long. In the three gears as received, the rollers were found to be of the following diameters:

Gear No. 51—$\frac{3}{4}$ in. diameter.
Gear No. 52—$\frac{27}{32}$ in. diameter.
Gear No. 53—$1\frac{1}{16}$ in. and $1\frac{1}{8}$ in. diameter (tapered).

This finding at once raises the question as to whether in repairs the correct size of roller would be used and whether, in fact, it would not be frequently omitted entirely. The condition of the gears of this type indicated that this company was not in a position to furnish commercial gears.

Harvey Friction Springs 8 in. x 8 in.

Gears No. 54, 55 and 56

The spring groups constituting these gears conformed reasonably close to the drawing dimensions except for the plain, inner coil centering springs which averaged $7\frac{15}{16}$ in. in free height instead of $7\frac{1}{2}$ in. as shown on the drawings. The results of the tests are believed to be representative of the commercial product.

A. R. A. Class G Springs

Gears No. 57, 58 and 59

The G springs used for the test were of ordinary carbon steel, oil tempered, drawn from regular railroad stock. The following tabulation will give the comparison of the test springs with the specification requirements of the American Railroad Association:

	Outer Coil Spring			Inner Coil Spring		
	Free Height	*Outside Diameter*	*Diameter of Bar*	*Free Height*	*Outside Diameter*	*Diameter of Bar*
Average of Test Spring.............	$7\frac{25}{32}$ in.	$7\frac{15}{16}$ in.	$1\frac{17}{32}$ in.	$7\frac{17}{32}$ in.	$4\frac{5}{8}$ in.	$\frac{31}{32}$ in.
Specified Dimension...............	$7\frac{7}{8}$ in.	8 in.	$1\frac{9}{16}$ in.	$7\frac{1}{2}$ in.	$4\frac{5}{8}$ in.	1 in.

9,000 LB. DROP TESTS

After measuring the test gears and reassembling them with their parts in their original positions, the 9,000 lb. drop tests were made. Except for a few gears that were added at a later date, the original series of drop tests was made at the Mt. Clare shops of the Baltimore & Ohio Railroad. After the car-impact tests at Rochester, the same gears were submitted to a second series of drop tests under the Pennsylvania Railroad machine at Altoona for check purposes, at which place the last few gears also were given their original drop tests.

The drop tests were in all instances made with the gears supported upon a solid anvil, a heavy plate casting being inserted instead of the springs regularly used beneath the anvil of the Baltimore & Ohio machine. Before beginning the drop tests of either of the above series each gear was given a certain amount of preliminary work to insure the proper seating of the parts. The uniform practice was followed of first determining the drop test value of each gear, by dropping the weight from 1 in. free fall and then increasing the fall by 1 in. increments, until the closing point was reached. The gear was then given 10 blows from 1 in. below the solid height, which usually resulted in building up the capacity of the gear slightly. After this preliminary work the regular drop tests were made, the tup being again dropped through heights increasing by 1 in. increments until the closing point was reached, as evidenced by flattening or shearing of lead records. In the case of gears such as the Harvey springs the solid point was previously determined from a preliminary static test and this point worked to in the drop test.

Two drop test diagrams have been reproduced for each type of gear to show the amount of gear closure at successive drops. These are shown in Figs. 18 to 35 inclusive, at the end of the chapter on static tests, along with the static diagrams for the same gears. The information for plotting the drop test diagrams was obtained during the first series of drop tests by causing the tup to drive a nail into the end of a wooden post, the penetration of the nail denoting gear closure for each successive drop. The diagrams have been plotted to the exact points recorded, with no averaging or smoothing up of the curves. The regularity of gear action can thus be seen and in such a test this is of as much, if not more interest than the general trend of the line.

Some of the drop test figures obtained in these tests are higher than usually reported for gears of the same type. The care taken to have all surfaces in good condition and the uniformity of testing conditions insures that the present results are comparable with each other. In general throughout this report the drop tests are reported in terms of "total fall," this being the free fall plus the penetration or actual travel of the gear. Some confusion has existed heretofore in this respect but it is proper to express these results in total fall rather than free fall if the true drop test capacities are to be compared.

The recoil of the 9,000 lb. weight was also measured by means of a special slide on the side of the drop machine. The quantities as tabulated are for the total recoil of the weight above the lowest point reached by it in closing the gear. The drop test capacity, foot pounds of work done, is accordingly represented by the potential energy in the weight at a height corresponding to the total fall required to close the gear. The energy given out by the gear upon release is denoted by the amount of recoil of the weight. The work absorbed is found by subtracting the energy of recoil from the "work done," or the total energy required to close the gear.

A discussion of the individual performance of the gears in the drop test follows:

Westinghouse D-3

Gears No. 1, 2 and 3

The action of these gears under the drop was entirely satisfactory. The initial flatness of the curves shows the result of the preliminary spring action and the curves as a whole indicate that the gear action is reasonably consistent throughout the entire range. The average total fall of the 9,000 lb. tup required to close a new gear of this type, when in good condition, is taken at 19.8 in., and the total recoil of the weight at 3.8 in. These figures are arrived at by averaging all of the drop test results for these gears, the same practice having been followed for each gear unless a statement to the contrary appears.

Westinghouse NA-1

Gears No. 4, 5, 6, 7 and 8

The drop test results on these gears are not quite so regular as on the older Westinghouse D-3 gear, but while the diagrams are more irregular, the action in general is good. The results also are considerably higher, hence it cannot be expected to find as regular action as in the lighter gear. Gear No. 8 showed slightly less in capacity than any of the others of this type. No breakage or failure of any kind occurred during these drop tests. The average total fall required to close a new gear of this type, when in good condition, is taken as 26.0 in., this being the average value of the three gears taken through the test. The total recoil is taken at 3.4 in.

Sessions K

Gears No. 9, 10, 11 and 12

The drop test diagrams for these gears, while not so smooth, are yet good for a gear of such short travel. In gears No. 9 and No. 10 the spring barrels began to scale before the gears went solid; in the case of gear No. 9 this began at 13 in. free fall, and in the case of gear No. 10 at 12 in. free fall. Failure of the gears had therefore begun before closure and hence the tests are not satisfactory. The average total fall required to close a new gear of this type, when in good condition, is taken as 18.8 in., this being the average value of the three gears taken through the test, and the total recoil at 4.3 in.

Sessions Jumbo

Gears No. 13, 14 and 15

This gear showed considerably more capacity and at the same time more uniform action under the drop test than the previous Sessions K gear. The spring barrel of gear No. 13 developed a crack during this test. The average total fall required to close a new gear of this type, in good condition, is taken at 28.1 in. and the total recoil at 5.2 in.

Cardwell G-25-A

Gears No. 16, 17 and 18

The action of these gears under the drop was good, the diagrams being especially smooth and regular. The cast iron friction blocks formed decided depressions in the malleable iron heads, however, and a crack developed at one corner of one of the friction blocks, while in the final drop tests at Altoona one of the side friction members was broken in halves. The average drop for the test gears of this type is 21.1 in., but as heretofore explained, it is believed that these test gears are not representative, the average drop test results obtained in United States Railroad Administration acceptance tests being 16.6 in. The gear is therefore credited with a value midway between these figures, or 18.9 in. total fall required to close an average new gear when in good condition. The average total recoil to be expected is taken at 2.8 in.

Cardwell G-18-A

Gears No. 19, 20 and 21

This gear showed smooth and regular action under the drop, and the diagrams are entirely satisfactory. The springs of gear No. 20 took a slight set during the drop tests. The average total fall required to close a new gear of this type, in good condition, is taken at 19.6 in. and the total recoil at 1.5 in.

It is interesting to note that whereas from the mechanics of the two types of Cardwell gear, the G-18-A should be of higher capacity than the G-25-A, yet the average results obtained from the test gears show 1.5 in. more fall required for the G-25-A than for the G-18-A. This shows further warrant for the action taken in allowing a reduced drop test value for the G-25-A gear.

Miner A-18-S

Gears No. 22, 23 and 24

The drop tests of these gears were satisfactory and the diagrams denote especially uniform gear action for all ranges. This is particularly noticeable because of the fact that the gear has a travel of but 2½ in. The average total fall required to close a new gear of this type, in good condition, is taken at 19.9 in. and the total recoil at 4.6 in.

Miner A-2-S

Gears No. 25, 26 and 27

These gears did not show so regular under the drop as the previous Miner gears but the diagrams are good. The drop capacity, however, is low, the average total fall required to close a new gear of this type, in good condition, being 13.2 in. The total recoil is taken at 3.8 in. In gear No. 25 the main spring went solid during this test.

National H-1

Gears No. 28, 29 and 30

This gear developed an unusually high capacity under the drop and while the diagrams are not entirely smooth, yet, considering the amount of fall and the short travel of 2½ in., the gear action is good. The average total fall required to close a new gear of this type, in good condition, is taken at 31.2 in., and the total recoil at 4.6 in.

National M-1

Gears No. 31, 32 and 33

The drop tests of these gears did not produce diagrams proportionally as smooth as those of the previous National gears, considering their lower capacity. The diagrams, however, show reasonably uniform gear action. The average total fall required to close a new gear of this type, in good condition, is taken at 19.2 in., and the total recoil at 3.4 in.

National M-4

Gears No. 34, 35 and 36

The action of this gear under the drop was very similar to that of the National M-1 just described. The average total fall required to close a new gear of this type, in good condition, is taken at 21.5 in., and the total recoil at 2.4 in.

Murray H-25

Gears No. 37, 38 and 39

These gears, while not of high capacity, showed the most regular action of any friction gear tested. The diagrams are unusually smooth and indicate consistent action throughout the full range of the gear. Considerable chafing and wear occurred during the closures under the drop. Upon removing one of the heads a cloud of dust could be blown from the friction surfaces. Unquestionably, this wear would soon deteriorate the gear. The average total fall required to close a new gear of this type, in good condition, is taken at 17 in., and the recoil at 3.3 in.

Gould 175

Gears No. 40, 41 and 42

These gears showed good action under the drop except for the fact that in each instance the plates of the friction spring took a slight permanent set. The gears showed high recoil and because of this feature it was difficult to keep them in position on the anvil. The average total fall required to close a new gear of this type, in good condition, is taken at 18.1 in. and the total recoil at 7.1 in.

Bradford K

Gears No. 43, 44, 45, 46 and 47

The drop testing of these gears was difficult and unsatisfactory. The springs went solid before the heads of the gears came together and gears No. 43 and No. 44 failed by splitting the heads. The failures were undoubtedly due to this spring condition, as extremely high forces are set up in this, as in most friction gears, if the springs go solid before the gear is closed. Gear No. 45 also developed a cracked head during the drop test. It is noticeable that the portion of the head immediately back of the coupler butt, in buffing, is not properly supported. Another serious point is that in several instances the heads pinched and stuck in the frame on release. These gears showed low capacity and high recoil under the drop, their action being very little different from that of a spring gear. The average total fall required to close a new gear of this type, in good condition, is taken at 10.8 in. and the total recoil at 5.3 in.

Waugh Plate Type

Gears No 48, 49 and 50

These gears gave reasonably smooth diagrams in the drop test but in each instance the plates took a permanent set. The drop capacity is low and the recoil high. The gear is of especially easy movement at the beginning of its travel. The average total fall required to close a new gear of this type, in good condition, is taken at 13.9 in. and the total recoil at 7.6 in.

Christy

Gears No. 51, 52 and 53

This gear was very erratic under the drop, and the action is not at all satisfactory. The gears as tested were shorter than the pocket dimension and this clearance allowed the wedge roller to get out of position upon recoil. The total fall required to close the test gears ranges from 14.3 in. to 26.3 in. It is therefore difficult to set an average value, but in the absence of better uniformity the three results have been averaged and the total fall set at 19.6 in. for this gear. The total recoil is taken at 5.1 in.

Harvey 8 in. x 8 in. Springs

Gears No. 54, 55 and 56

Each of these gears as tested consisted of two Harvey 8 in. x 8 in. springs, set side by side upon the anvil. The gears showed but little capacity under the drop, although the action was regular. In the case of gear No. 55 the springs took a slight permanent set. A total fall of 9.5 in. has been set as the drop test value of this gear (two complete springs) and the total recoil is taken at 4.2 in.

A. R. A. Class G Springs

Gears No. 57, 58 and 59

Each of these gears as tested consisted of two A.R.A. Class G springs, set side by side upon the anvil. The springs showed low capacity in the drop test, but the action was smooth throughout the range of the springs. A total fall of 5.8 in. has been set as the drop test value of two Class G springs, working either in twin or tandem fashion, and the total recoil is taken as 4.1 in.

Summary of 9,000 Lb. Drop Tests

The table, Fig. 16, has been prepared to show a summary of the drop tests, and the following paragraphs will explain the several columns of this table:

Column 1 is self-explanatory.

Column 2 gives the nominal travel as called for on the manufacturer's drawings.

Column 3 identifies the test gears by number.

Column 4 gives the actual travel obtained from the gears in the drop tests. In cases where the free length of the gear is less than the standard pocket dimension the actual travel has been given and an explanatory note made in Column 14.

Column 5 gives the actual free fall of the 9,000 lb. weight required to just close the new test gears. These figures do not include the travel of the gear.

Column 6 gives the actual total fall required to just close the new test gears and is obtained by adding the quantity in Column 5 to the actual travel as given in Column 4.

Column 7 gives the actual recoil of the 9,000 lb. weight from the fall indicated in Column 6. The recoil is from the lowest point reached by the weight when the gear was just closed.

Column 8 indicates the work done by the 9,000 lb. weight falling through the heights given in Column 6.

Column 9 represents the energy absorbed by the gear, based on the work done as given in Column 8, less the energy of recoil (Column 7).

Columns 5 to 9 give the individual results actually obtained with the test gears.

Columns 10 to 13 give average or modified results of a similar character, such as may be expected from gears of the same type, as they are manufactured and furnished commercially, with no selection for test purposes.

MAKE AND TYPE OF GEAR	NOMINAL TRAVEL	TEST GEAR NUMBER	ACTUAL RESULTS FROM TEST GEARS						AVERAGE RESULTS TO BE EXPECTED FROM COMMERCIAL GEARS				REMARKS
			TRAVEL OBTAINED	9000 LBS. WEIGHT.					9000 LBS. WEIGHT.				
				FREE FALL TO CLOSE GEAR.	TOTAL FALL TO CLOSE GEAR.	RECOIL ABOVE CLOSING POINT.	WORK DONE FT. LBS.	WORK ABSORBED FT. LBS.	TOTAL FALL TO CLOSE GEAR.	RECOIL ABOVE CLOSING POINT	WORK DONE	WORK ABSORBED	
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)
WESTINGHOUSE D-3	$2\frac{7}{16}$"	1	2.50"	16"	18.50"	3.70"	13875	11100	19.8"	3.8"	14850	12000	
		2	2.50"	17"	19.50"	3.75"	14625	11813					
		3	2.50"	19"	21.50"	3.83"	16125	13253					
WESTINGHOUSE NA-1	3"	4	3.00"	25"	28.00"		21000		26.0"	3.4"	18750	16200	
		5	3.00"	27"	30.00"		22500						
		6	3.06"	24"	27.06"	3.52"	20295	17655					
		7	3.00"	23"	26.00"	3.50"	19500	16875					
		8	3.00"	22"	25.00"	3.27"	15000	12548					
SESSIONS K	$2\frac{1}{16}$"	9	2.00"	19"	21.00"		15750		18.8"	4.3"	14100	10875	
		10	2.17"	16"	18.17"	3.78"	13628	10793					
		11	2.06"	18"	20.06"	4.43"	15045	11723					
		12	2.06"	16"	18.06"	3.92"	13545	10605					
SESSIONS JUMBO	3"	13	3.10"	23"	26.10"	5.03"	19575	15803	28.1"	5.2"	21075	17175	
		14	3.06"	24"	27.06"	5.06"	20295	16500					
		15	3.00"	28"	31.00"	6.44"	23250	18420					
CARDWELL G-25-A	$2\frac{3}{4}$"	16	2.75"	19"	21.75"	3.61"	16313	13605	* 18.9"	2.8"	14175	12075	* See Text.
		17	2.75"	18"	20.75"	2.92"	15563	13373					
		18	2.75"	18"	20.75"	3.12"	15563	13223					
CARDWELL G-18-A	$3\frac{3}{16}$"	19	3.25"	19"	20.25"	1.82"	15188	13823	19.6"	1.5"	14800	13575	
		20	3.29"	15"	18.29"	1.00"	13718	12968					
		21	3.25"	17"	20.25"	1.54"	15188	14033					
MINER A-18-S	$2\frac{1}{2}$"	22	2.51"	15"	17.51"	4.18"	13133	9998	19.9"	4.6"	14925	11475	
		23	2.52	17"	19.52"	4.53"	14640	11243					
		24	2.53"	20"	22.53"	4.87"	16898	13245					
MINER A-2-S	$2\frac{1}{2}$"	25	2.55"	11"	13.55"	3.83"	10163	7290	13.2"	3.8"	9900	7050	
		26	2.53"	11"	13.53"	3.82"	10148	7283					
		27	2.50"	10"	12.50"	3.79"	9375	6533					

Fig. 16—Comparative Performance of Gears in Drop Tests
(See also facing page)

NATIONAL H-1	2 1/2"	28	2.53"	29"	31.53"	4.61"	23648	20190	31.2"	4.6"	23400	19950	
		29	2.50"	30"	32.50"	4.80"	24375	20775					
		30	2.50"	27"	29.50"	3.97"	22125	19148					
NATIONAL M-1	2 1/2"	31	2.52"	16"	18.52"	3.14"	13890	11535	19.2"	3.4"	14400	11850	
		32	2.53"	16"	18.53"	3.22"	13898	11483					
		33	2.53"	18"	20.53"	3.61"	15398	12690					
NATIONAL M-4	2 1/2"	34	2.54"	17"	19.54"	2.13"	14655	13058	21.5"	2.4"	16125	14325	
		35	2.46"	21"	23.46"	4.44"	17595	14265					
		36	2.53"	19"	21.53"	2.32"	16148	14408					
MURRAY H-25	2 3/4"	37	2.65"	15"	17.65"	3.68"	13238	10478	17.0"	3.3"	12750	10275	
		38	2.47"	14"	16.47"	3.41"	12323	9795					
		39	2.81"	14"	16.81"	3.20"	12608	10208					
GOULD 175	2 1/2"	40	2.38"	16"	18.38"	7.68"	13785	8025	18.1"	7.1"	13575	8250	*Gould Gear No. 41 was 1/16" Shorter Than The Pocket Length.
		41	*2.44"	15"	17.44"	6.94"	13080	7875					
		42	2.50"	16"	18.50"	7.29"	13875	8408					
BRADFORD K	2 1/2"	43	2.38"	FAILED					10.8"	5.3"	8100	4125	
		44	2.44"	FAILED									
		45	2.45"	9"	11.45"	5.41"	8588	4530					
		46	2.44"	6"	8.44"	5.19"	6330	2438					
		47	2.44"	10"	12.44"	5.87"	9330	6242					
WAUGH PLATE	2 1/4"	48	2.17"	12"	14.17"	8.04"	10628	4598	13.9"	7.6"	10425	4725	
		49	2.25"	11"	13.25"	7.25"	9938	4500					
		50	2.25"	12"	14.25"	8.22"	10688	4523					
CHRISTY	2 1/2"	*51	2.32"	12"	14.32"	3.38"	10740	8228	19.6"	5.1"	14700	10875	*Christy Gear No. 51 was .22" Shorter Than The Pocket Length. Gear No. 52 was .38" Shorter Than The Pocket Length; and Gear No. 53 was 3/16" Shorter Than The Pocket Length.
		*52	2.21"	16"	18.21"	4.96"	13658	9938					
		53	2.28"	24"	26.28"	5.87"	19710	15308					
HARVEY 2-8"x8" SPGS.	1 7/8"	54	1.88"	6"	7.88"	3.84"	5910	3130	9.5"	4.2"	7025	3975	
		55	1.76"	9"	10.76"	4.51"	8070	4688					
		56	1.83"	8"	9.83"	4.22"	7373	4208					
COIL SPRINGS 2-8"x8"-CLASS G	1 7/8"	57	1.83"	4"	5.83"	4.16"	4373	1253	5.8"	4.1"	4350	1275	
		58	1.70	4"	5.70"	4.08"	4275	1215					
		59	1.71	4"	5.71"	4.11"	4283	1200					

Fig. 16—Comparative Performance of Gears in Drop Tests

(See also facing page)

STATIC TESTS

Immediately upon the completion of the drop tests the same two gears of each type were closed in a testing machine at a speed of ⅛ in. per minute. Readings were taken for each 1/10 in. of compression, the closure being continuous, with no stops except where sudden changes in load occurred.

In many gears, when being closed at a slow speed, the resistance will build up at an abnormal rate, and shortly, from some reason such as the elasticity of the parts, a sudden readjustment will occur. In many instances this is accompanied by a loud report that may be best described by use of the word "bombardment." Invariably such readjustment results in a sudden reduction of the load. When a bombardment occurred during the tests, the machine was stopped until a full record of the conditions could be obtained. In plotting the static test diagrams the actual results have been used, all bombardment actions being shown, and no curves having been averaged or smoothed out, as is frequently done in plotting such diagrams. No gear should be condemned, however, solely because of the presence of bombardments or irregular action in the static tests, for some gears, while showing very irregular static diagrams, and even total failure in this slow test will yet show excellent results in both the drop test and the car-impact tests. Bombardments are conceded to be a normal action of many types of gears in slow static testing.

The car-impact tests show that when cars are coupled at a velocity of four miles per hour, each of the two opposing draft gears begins to close at a rate of 2112 in. per minute (176 ft. per min.) and that the closing rate gradually falls off until it is zero at the point of maximum gear closure, this corresponding with the point where both cars are of equal velocity. The average rate of closure at four miles per hour impact is approximately 1400 in. per minute; the static test rate of ⅛ in. per minute exists for less than 1/100 in. of gear movement. Results of slow static tests cannot therefore be compared with service or impact tests. The static is the simplest and the easiest draft gear test to make, and it is probably understood by the average mechanical man better than any other. It is unfortunate, therefore, that it is not more reliable, but as will be seen as the various tests are discussed, it is usually misleading and cannot be employed for comparing the merits of different draft gears.

The practice of rating gears upon a supposed ultimate resistance such as, for example, "a 200,000 lb. gear" or "a 350,000 lb. gear," is to be discouraged. Due to the limited travel of draft gears it is necessary that the ultimate resistance of the gear be high if cars are to be handled at switching speeds above two miles per hour. The manner in which this ultimate resistance is reached is of great importance. It will be seen in some of the static cards that while the ultimate resistance is high, yet at the beginning of the diagram it is extremely low and becomes really effective only during the last quarter of the diagram. This means not only that the gear is of low capacity for its ultimate resistance, but that the final rate of building up the force is too high and will set up undesirable vibrations in the car structure. The ultimate resistance of a gear cannot, therefore, be wholly indicative either of capacity or of cushioning value, capacity being a product of the average force and travel, and cushioning being dependent upon the rate of building up of the force as well as the magnitude of the force itself.

Static test diagrams have been plotted for closures at the rate of ⅛ in. per minute and are reproduced in Figs. 18 to 35, inclusive, at the end of this chapter, on the same sheets with the drop test diagrams. The same gears were also partially closed at the rate of ¾ in. per minute and at an average rate of 3 in. per minute, for comparison. These three closures were made in immediate succession so that the condition of the gear had not changed.

A discussion of the individual performance of the gears in the static tests follows:

Westinghouse D-3

Gears No. 1 and 2

The action of these gears during compression was smooth and regular, with an occasional clicking, but no noticeable falling off in load. On the release, in each instance some slight tendency to stick was observed. No failure of any parts of the gears occurred in these tests. The effect of the preliminary spring action is noticeable in both diagrams, but in neither instance is there shown the customary flattened top to these curves, which ordinarily results from the functioning of the pressure limiting feature.

Westinghouse NA-1

Gears No. 4 and 5

The static test was attempted on two gears only of the above type, gears No. 4 and No. 5. This was made at a speed of ⅛ in. per minute and in each instance the resistance built up to such an extent at this slow speed that the gears were destroyed. The failures occurred by bulging and shortening of the barrel and by breaking the supporting ends off the stationary friction plates or spacers. In each instance the gears failed to such an extent that they were eliminated and gears No. 6 and No. 7 substituted for the further tests. No attempts were made to conduct static tests on gears No. 6 and No. 7.

These failures in the slow static test at a speed of ⅛ in. per minute do not necessarily disqualify the type of gear, as in the later car-impact tests the average rate of gear closure was approximately 1450 in. per minute, or 11,500 times as rapid as in this static test. Furthermore, these high resistances were not found in the car-impact tests.

Sessions K

Gears No. 9 and 10

Two gears only of this type, gears No. 9 and No. 10, were subjected to the static test, as one of them, gear No. 9, failed in the test. This failure was similar to that of the previously discussed Westinghouse NA-1 gear and the same remarks apply. The failure was by bulging and shortening of the barrel and bending and elongation of the friction box. The friction faces after the test were found to be badly galled and drawn. This gear was eliminated and gear No. 11 substituted for the tests.

The action of these Sessions K gears in the static test was unusual. This gear usually bombards badly and often requires sledging to keep it moving when being subjected to static tests. In the present instance the action was smooth and without a single bombardment. On attempting to release gear No. 10, however, it was found to be stuck, and sledging was resorted to in order to start the release. From this on the release line was smooth. After this test the friction surfaces of the gear were found to be galled and drawn. Such action indicates that either the friction blocks or box were made of extremely soft material.

Sessions Jumbo

Gears No. 13 and 14

The performance of these gears in the static test was typical of static tests generally. Gear No. 13 developed many light bombardments. Gear No. 14 creaked continually while being compressed, but showed no falling off in load of any magnitude. The action of both gears on release was good. The fact that one of these gears showed so much greater ultimate resistance and capacity than the other in this test is undoubtedly due to the condition of the friction surfaces.

Cardwell G-25-A

Gears No. 16 and 17

These gears closed reasonably smoothly in the static test, there being a continual creaking but no falling off in load of any magnitude. In each gear a small corner was broken off of one of the triangular friction blocks during this test. After the test the friction surfaces were found to be somewhat galled. Gear No. 16 stuck at one point on the release, but snapped loose, and the remainder of the release was smooth. These gears each had a very heavy initial compression and this is shown in the beginning of the curves.

Cardwell G-18-A

Gears No. 19 and 20

The action of these gears in the static test was good, there being no bombardments, but a continual creaking. The release was steady and the friction faces after the tests were in good condition. Gear No. 19 had a heavier initial compression than gear No. 20, and this is reflected in the curves.

Miner A-18-S

Gears No. 22 and 23

These gears in the static test showed smooth action both on compression and release, there being neither bombardments nor creaking. The friction faces after the tests were in good condition. The heavy initial compression of these gears is evident in the diagrams.

Miner A-2-S

Gears No. 25 and 26

The action of the two gears of this type tested was quite different, as can be seen from the diagrams, gear No. 25 being of much less ultimate resistance and capacity than gear No. 26. In the drop tests immediately preceding, however, the two gears were of equal capacity. The action of both gears was smooth except near the closing point of gear No. 26, where the load became irregular. No bombardments or creaking accompanied this change in load. On release the action was smooth in both instances. Both gears had heavy initial compression.

The results obtained from the static tests of these two gears illustrate forcibly a fact frequently noticed: that a slight change in the condition of the friction surfaces may not materially affect the action of the gear in the drop test and yet entirely change the results in the static test.

National H-1

Gears No. 28 and 29

The static tests of these gears are especially interesting. Gear No. 28 showed a very low ultimate resistance when being closed at a speed of $\frac{1}{8}$ in. per minute and in the succeeding tests, as the closing speed was increased the resistance increased; because of this the three compression lines are shown on the static diagram. In the case of gear No. 29, the resistance at ⅛ in. per minute built up beyond the capacity of the 600,000 lb. testing machine. Yet in the drop tests the two gears were of practically equal capacity. This action was undoubtedly due to friction surface conditions that could not be detected, and the results further confirm the idea that a slight deposit upon the friction surfaces may materially influence the static tests, but that higher speed tests may not be affected to the same degree. This is of special importance when it is considered that in service the highest draft gear capacity is needed at the highest closing speeds.

Two bombardments occurred during the closing of gear No. 29. The release of both gears was smooth.

National M-1

Gears No. 31 and 32

The action of these gears in the static test was good. Two slight bombardments occurred during the closing of gear No. 31, but none in the case of gear No. 32. The release of both gears was smooth.

National M-4

Gears No. 34 and 35

These gears bombarded during the static tests and in each instance a bombardment occurred near the point of closure, so that the resistance had no opportunity to again build up before the end of gear travel was reached. The figures given in Column 5 of the table, Fig. 17, are for the maximum resistances, namely, 358,000 lb. and 349,000 lb. The results in Columns 3 and 4, however, are for resistances at the end of gear closure. It will be understood that in another test bombardments might occur at other points or even disappear entirely.

Murray H-25

Gears No. 37 and 38

The action of these gears in the static test was regular and smooth, there being no bombardment or creaking. The release also was smooth and positive. These static cards are among the smoothest and most regular obtained from friction gears.

Gould 175

Gears No. 40 and 41

These gears in the static test closed by means of a continued series of light bombardments, as reproduced in the diagrams. At the beginning of the release a tendency to stick occurred in each gear, this being reflected in the release curves.

Bradford K

Gears No. 45 and 46

The static tests of these gears, while smooth and regular, show but little resistance for a friction gear, and practically no friction. The release is positive, but of such a character as to indicate very little absorption of energy.

Waugh Plate Type

Gears No. 48 and 49

The action of these gears was smooth and regular. Gear No. 49 showed practically no resistance during the first ½ in. of compression. The release was positive and smooth. Each of these gears took a permanent set of approximately ⅛ in. during the static test.

Christy

Gears No. 51 and 52

These were the most erratic gears in the static tests. Gear No. 51 gave one violent bombardment after another, and also stuck at one point on the release. Gear No. 52 stuck on the compression and could not be closed in the 600,000 lb. machine. The barrel of the gear started to fail and the final yield shown in the curves is not due entirely to movement of the friction parts, but partly to shortening of the barrel. The friction surfaces were found to be badly galled and drawn after the test.

Harvey 8 in. x 8 in. Springs

Gears No. 54, 55 and 56

Each of these gears as tested consisted of two 8 in. x 8 in. Harvey springs set side by side in the testing machine. It will be noticed that most of the resistance is in the later portion of the travel and increases abruptly. One heavy bombardment occurred during the test of gear No. 55, this incidentally showing that the Harvey springs are actually friction mechanisms. The release lines, which are smooth and regular, also show that friction is present. The absorption, however, is low.

A. R. A. Class G Springs

Gears No. 57, 58 and 59

Each of these gears as tested consisted of two 8 in. x 8 in. Class G springs set side by side in the testing machine. The diagrams obtained show typical coil spring action, there being a slight amount of absorption due to hysteresis and to the friction of the free ends of the spring against the faces of the testing machine.

Summary of Static Tests

The results of the static tests are shown in the table, Fig. 17. For comparison there are shown also the results obtained in the drop test and the later car-impact tests for the same gears. It will be noticed in general that the resistances as given are in excess of figures commonly reported for gears of the same types. This may be due to the fact that the friction surfaces in these tests were in good condition and that all gears had identical treatment. The following description of the several columns of this table will serve to explain it more fully:

Column 1 is self-explanatory.

Column 2 identifies the test gears by number.

Column 3 gives, for a closing speed averaging 3 in. per minute, the ultimate or maximum resistance of the gear at the point where the gears just closed or where normal gear action ceased. Columns 4 and 5 give the ultimate resistance, at closing speeds of ¾ in. and ⅛ in. per minute. An asterisk (*) in any of these columns denotes that the maximum resistance as tabled was developed before the gear was closed, a bombardment or other cause reducing the resistance at the point of closure. It will be understood that the capacity of the testing machine would not admit of complete closures at the 3 in. and ¾ in. speeds. The results have been extended proportionately, however, so that the tabulated results represent complete gear closures.

Columns 6 and 7 give the work done and the work absorbed by the gears in the static test at the ⅛ in. per minute closure.

Columns 8 and 9 give for comparison the work done and work absorbed by the same gears in the drop test.

Column 10 gives a computed resistance for the drop test. This figure has been obtained by proportioning the resistances to the foot-pounds of work done in these two tests. Thus in gear No. 1, the static test at ⅛ in. per minute gave an ultimate resistance of 190,000 lb. with 18,434 ft. lb. of work done. In the drop test the work done was 13,875 ft. lb.; hence on the same basis the ultimate gear resistance is 143,000 lb. The figures in this column therefore, although purely hypothetical, are of interest. If the static card is indicative of the dynamic force curve, then the results in Column 10 are approximately correct, for inasmuch as the one leg (gear travel) is the same in both diagrams, the other leg (resistance) should be roughly proportional to the area, or work done.

Columns 11 and 12 give the work done and work absorbed by the same gears in the later car-impact tests at Rochester.

Column 13 gives the ultimate resistance figures obtained for these gears in the car-impact tests.

The resistance figures given in this table thus represent a variety of speeds and conditions of gear closure, the static closures being at a constant speed and the drop and car-impact closures beginning at a high speed, which gradually reduces to zero at the point of maximum closure. In the car-impact tests, with a gear in each car, the gears begin to close at a speed not less than 1056 in. per minute at 2 miles per hour impact, or 2112 in. per minute at 4 M.P.H. impact. In the drop test, with a 15 in. free fall, the gears begin to close at a speed of 6458 in. per minute, or 9130 in. per minute at 30 in. free fall.

Make and Type of Gear.	Test Gear Number	Static Test — Speed of Testing Machine. 3" per min. Ultimate Resistance	3/4" per min. Ultimate Resistance	1/8" per minute. Ultimate Resistance	1/8" per minute. Work Done	1/8" per minute. Work Absorbed	Drop Test — Work Done	Drop Test — Work Absorbed	Drop Test — Computed Ultimate Resistance	Car-Impact Test. Work Done	Car-Impact Test. Work Absorbed	Car-Impact Test. Ultimate Dynamic Resistance
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
Westinghouse D-3	1	189000#	196000#	190000#	18434 FT. LBS	16400 FT. LBS	13875 FT. LBS	11100 FT. LBS	143000#			
	2	207000	212000	200000	18667	16534	14625	11813	157000	13800 FT. LBS	11433 FT. LBS	195000#
	3						16125	13253		15533	12900	240000
Westinghouse NA-1	4			FAILED			21000					
	5			FAILED			22500					
	6						20295	17655				
	7						19500	16875		21417	19883	158000
	8						15000	12548		16917	13550	187000
Sessions K	9			FAILED			15750					
	10	401000	443000	395000	34700	32400	13628	10793	155000			
	11						15045	11723		23733	20317	260000
	12						13545	10605		15000	12433	165000
Sessions Jumbo	13	318000	318000	318000	32300	28450	19575	15803	163000			
	14	504000	560000	457000	61970	57000	20295	16500	150000	17650	13216	137000
	15						23250	18420		20400	15417	250000
Cardwell G-25-A	16	327000	381000	410000	52500	51234	16313	13605	128000			
	17	332000	337000	356000	46600	43800	15563	13373	120000	16233	13100	295000
	18						15563	13223		19600	17967	315000
Cardwell G-18-A	19	234000	225000	240000	32400	31300	15188	13823	112000			
	20	141000	135000	146000	20100	18700	13718	12968	100000	14033	12417	110000
	21						15188	14033		20200	18733	186000
Miner A-18-S	22	367000	383000	290000	36067	33233	13133	9998	106000			
	23	419000	424000	364000	46100	43600	14640	11243	116000	22800	19167	390000
	24						16898	13245		14633	7500	390000
Miner A-2-S	25	173000	152000	146000	19500	18200	10163	7290	76000			
	26	344000	346000	289000	49833	48033	10148	7283	59000	11283	9733	105000
	27						9375	6533		8767	7100	68000

Fig. 17—Comparative Ultimate Resistance of Gears

(*See also facing page*)

NATIONAL H-1	28	472000	330000	215000	10333	7000	23648	20190	492000				
	29	541000	528000	600000			24375	20775		25867	20500	390000	
	30						22125	19148		28500	21000	550000	
NATIONAL M-1	31	487000	493000	518000	43933	42067	13890	11535	164000				
	32	461000	414000	550000	62267	58467	13898	11483	122000	23033	20333	400000	
	33						15398	12690		16967	13234	218000	
NATIONAL M-4	34	442000	435000	*358000	34600	32167	14655	13058					
	35	412000	387000	*349000	28330	26300	17595	14265		20633	19283	159000	
	36			* SEE TEXT.			16148	14408		16300	12350	138000	
MURRAY H-25	37	241000	242000	246000	18767	16600	13238	10478	174000				
	38	207000	204000	206000	17500	15900	12323	9795	145000	14800	12867	210000	
	39						12608	10208		13000	10300	130000	
GOULD 175	40	267000	280000	253000	20700	17167	13785	8025	169000				
	41	261000	258000	250000	19667	16934	13080	7875	166000	12567	8067	260000	
	42						13875	8408		14967	12133	230000	
BRADFORD K	43												
	44												
	45	104000	101000	103000	6450	2083	8588	4530	137000				
	46	82000	81000	85000	6367	1333	6330	2438	84000	7333	2883	270000	
	47						9330	6242		6333	1417	220000	
WAUGH PLATE	48	187000	181000	200000	9900	4967	10628	4598	214000				
	49	148000	144000	150000	7300	3867	9938	4500	204000	7767	2733	335000	
	50						10688	4523		10433	5500	285000	
CHRISTY	51	194000	180000	*163000	16300	15033	10740	8228					
	52			600000			13658	9938		12300	10317	194000	
	53			* SEE TEXT.			19710	15308		13567	11500	150000	
HARVEY 2-8"x 8" SPGS.	54	364000	380000	430000	10400	5400	5910	3130	244000				
	55	377000	361000	382000	7667	4134	8070	4688	402000	5900	4317	245000	
	56	393000	409000	458000			7373	4208		4083	1100	300000	
COIL SPRINGS 2-8"x8" CLASS G	57			59000			4373	1253					
	58			56000	3567	434	4275	1215	67000	3900	600	60000	
	59			60000	4034	434	4283	1200	64000	4333	300	60000	

*Denotes that maximum resistance was developed before the gear was closed, a bombardment or other cause reducing the resistance at the point of closure.

FIG. 17—COMPARATIVE ULTIMATE RESISTANCE OF GEARS

(*See also facing page*)

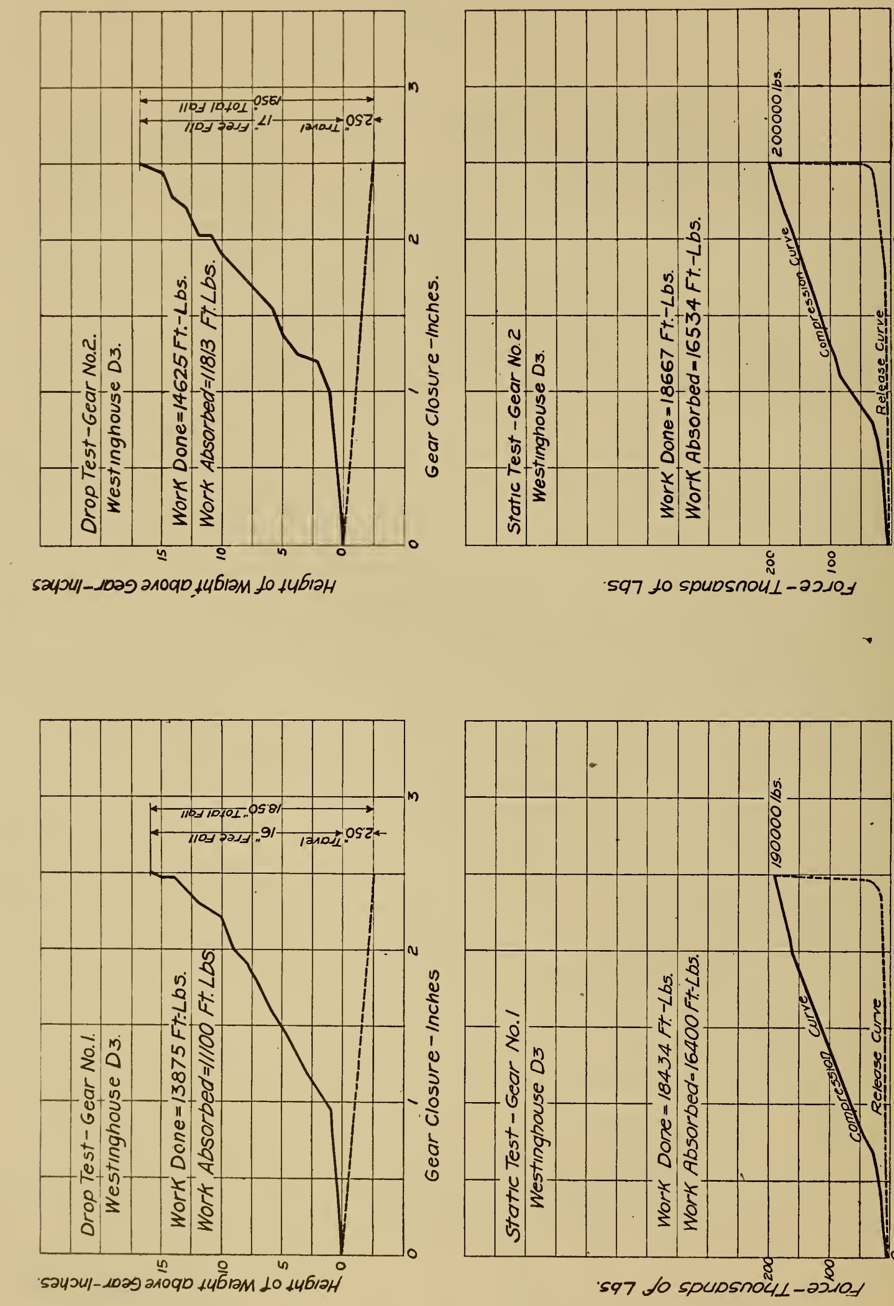


Fig. 18—Drop Test and Static Test Diagrams, Westinghouse Type D-3

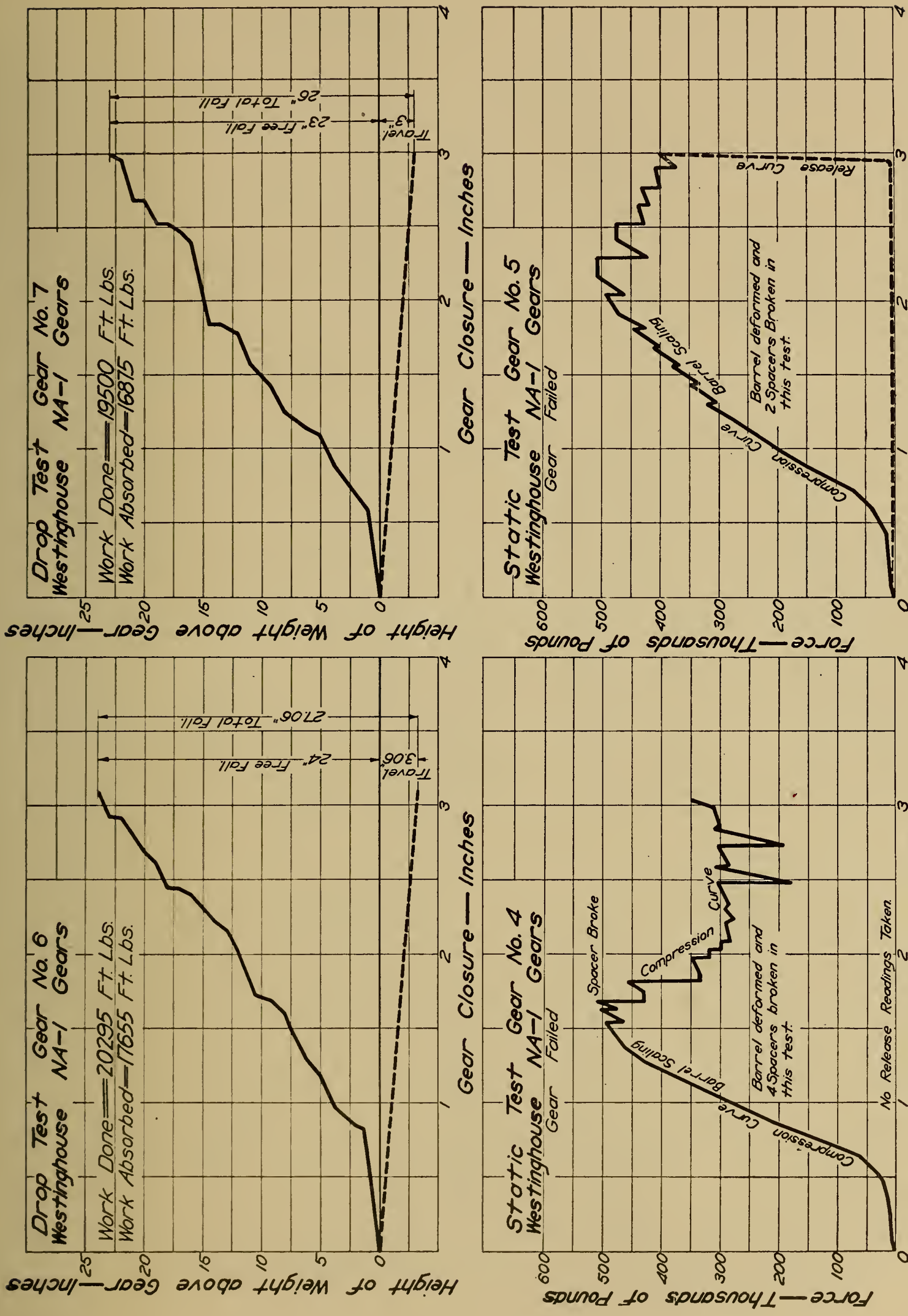


Fig. 19—Drop Test and Static Test Diagrams, Westinghouse Type NA-1

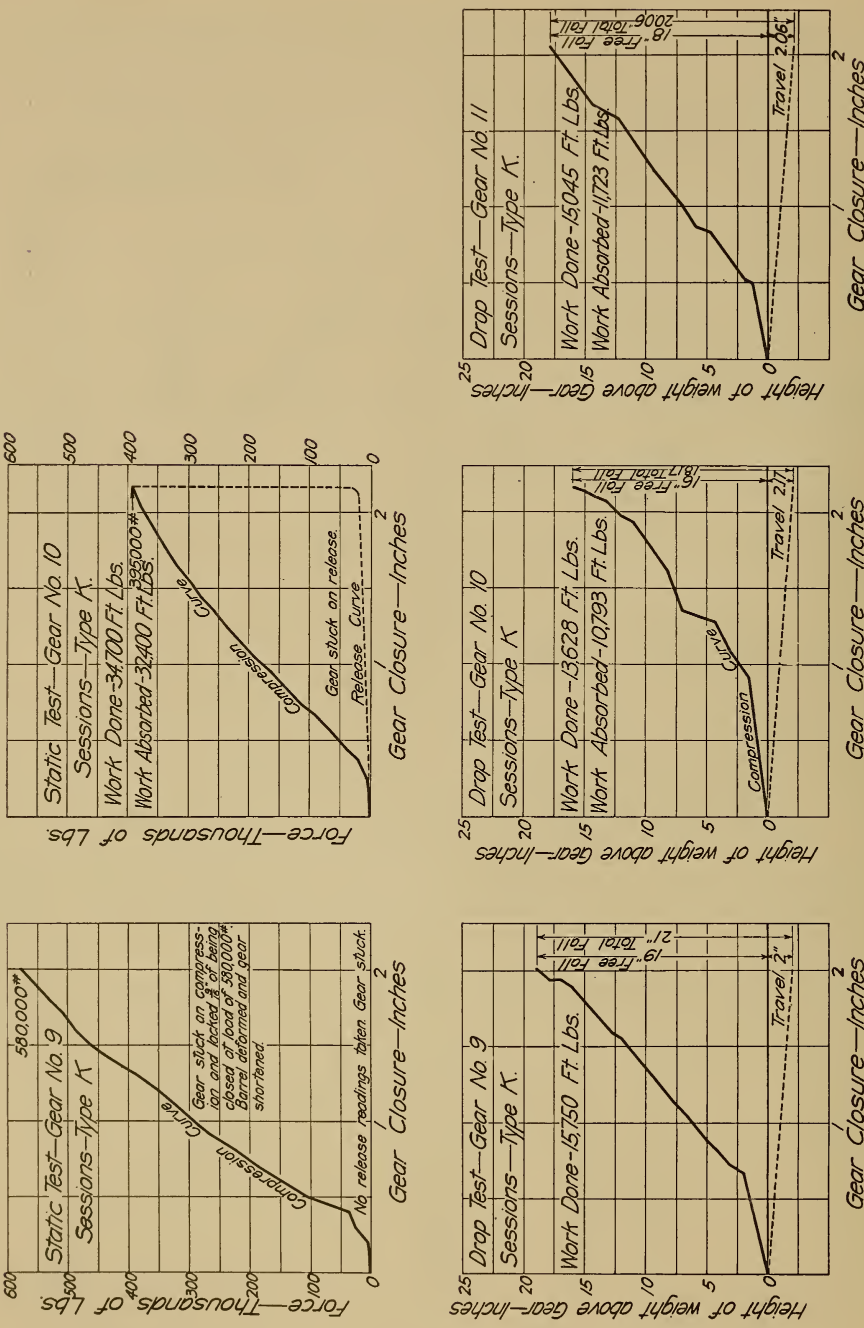


Fig. 20—Drop Test and Static Test Diagrams, Sessions Type K

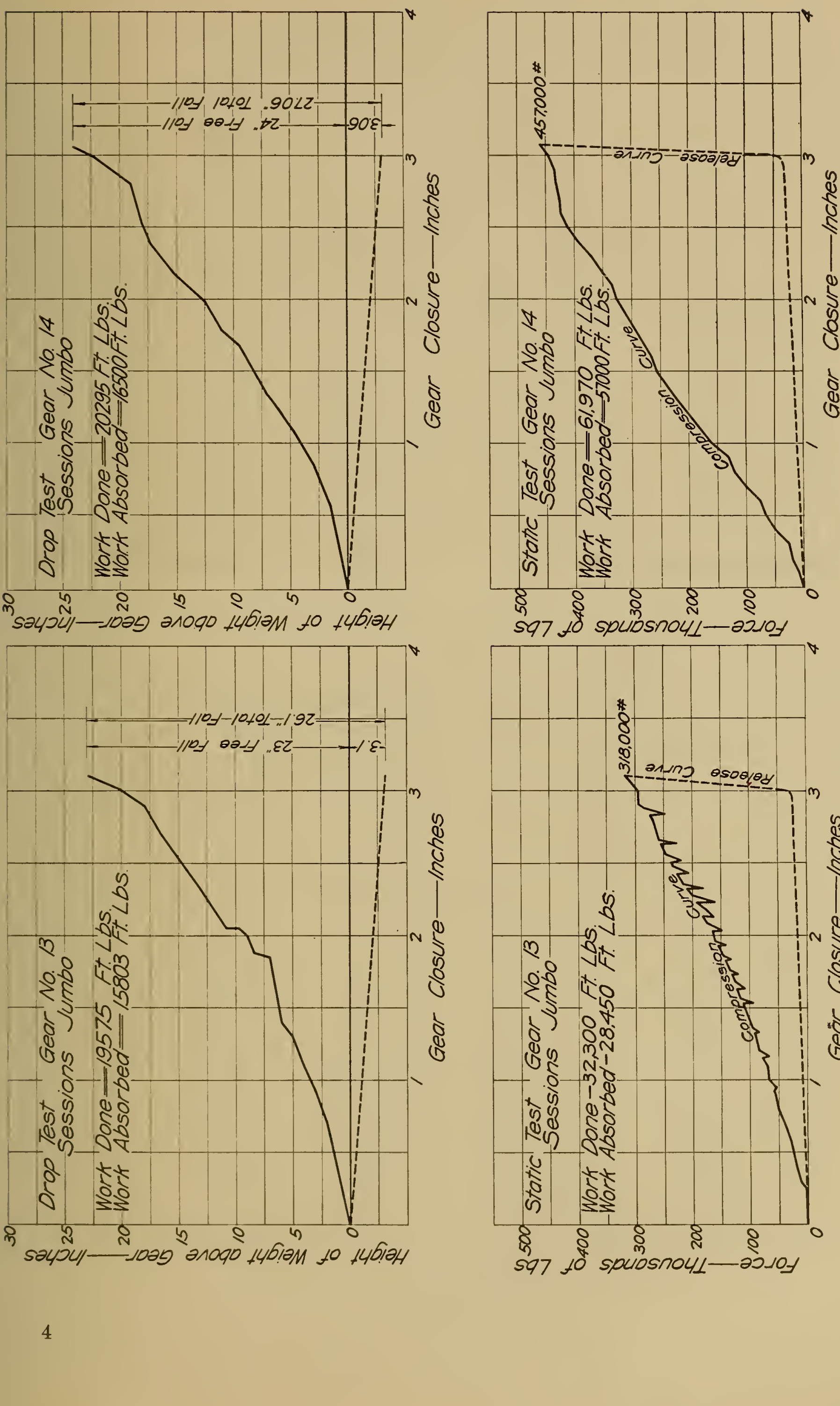


Fig. 21—Drop Test and Static Test Diagrams, Sessions Jumbo

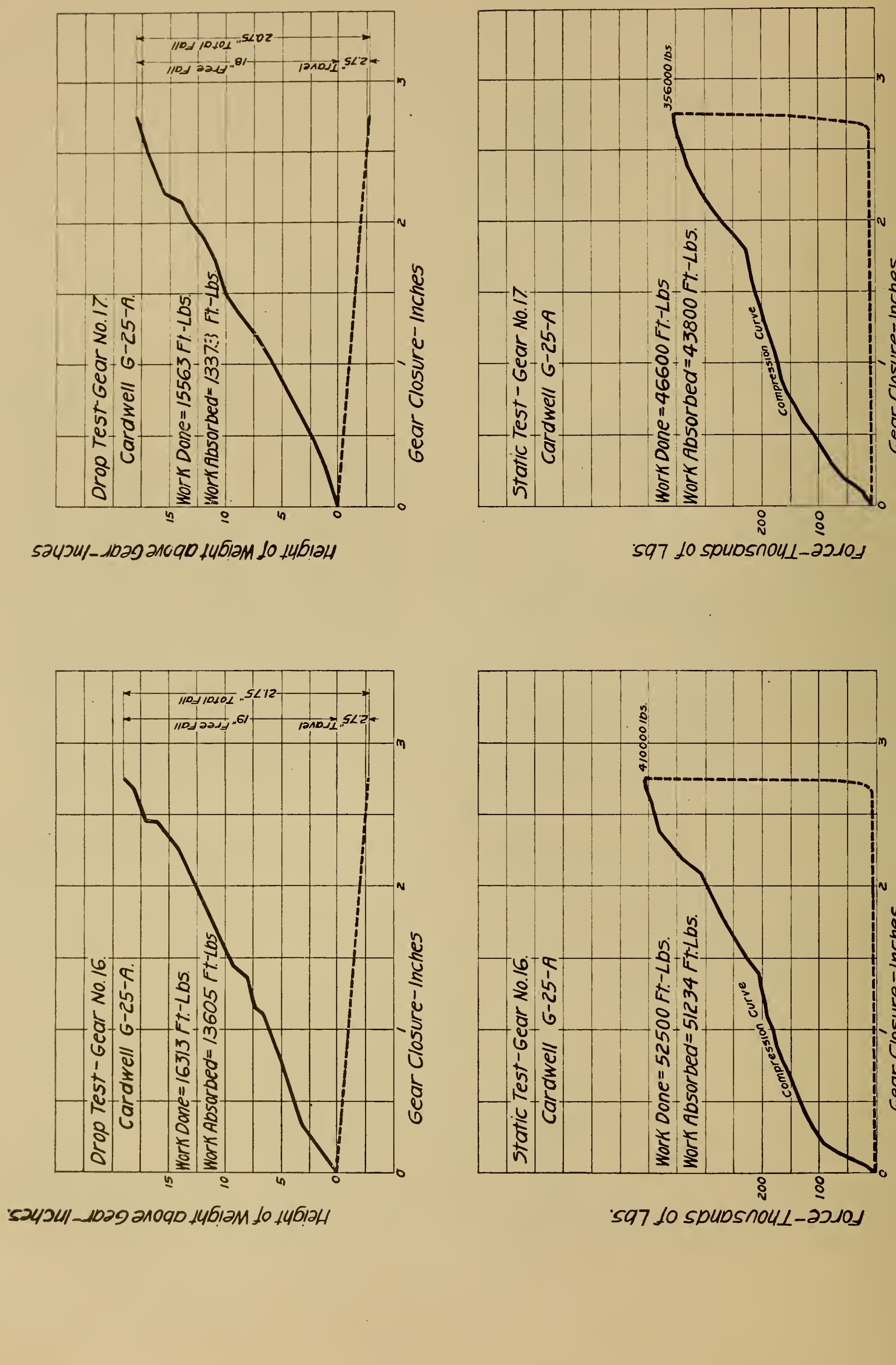


Fig. 22.- Drop Test and Static Test Diagrams, Cardwell Type G-25-A

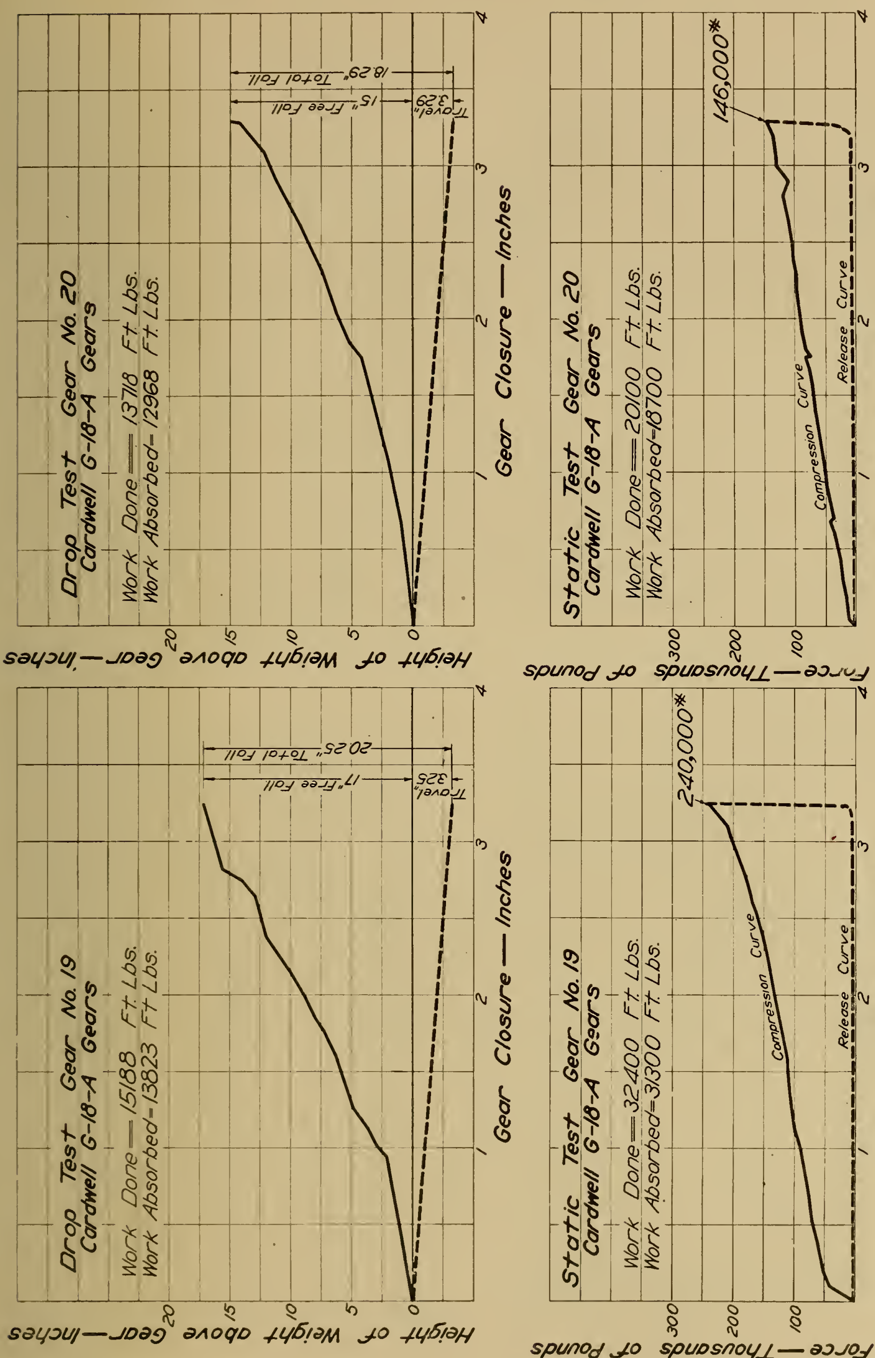


Fig. 23—Drop Test and Static Test Diagrams, Cardwell Type G-18-A

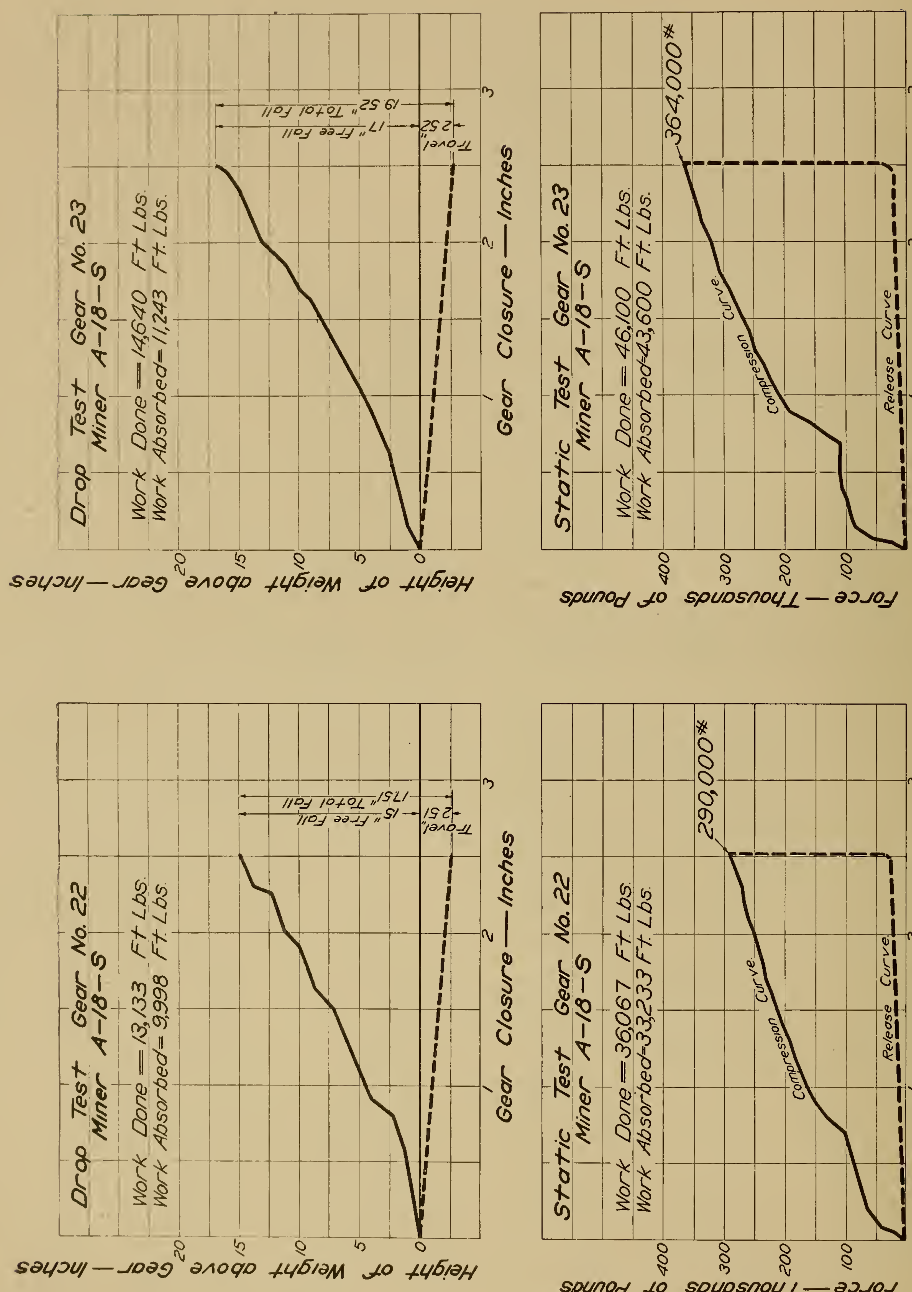


Fig. 24—Drop Test and Static Test Diagrams, Miner Type A-18-S

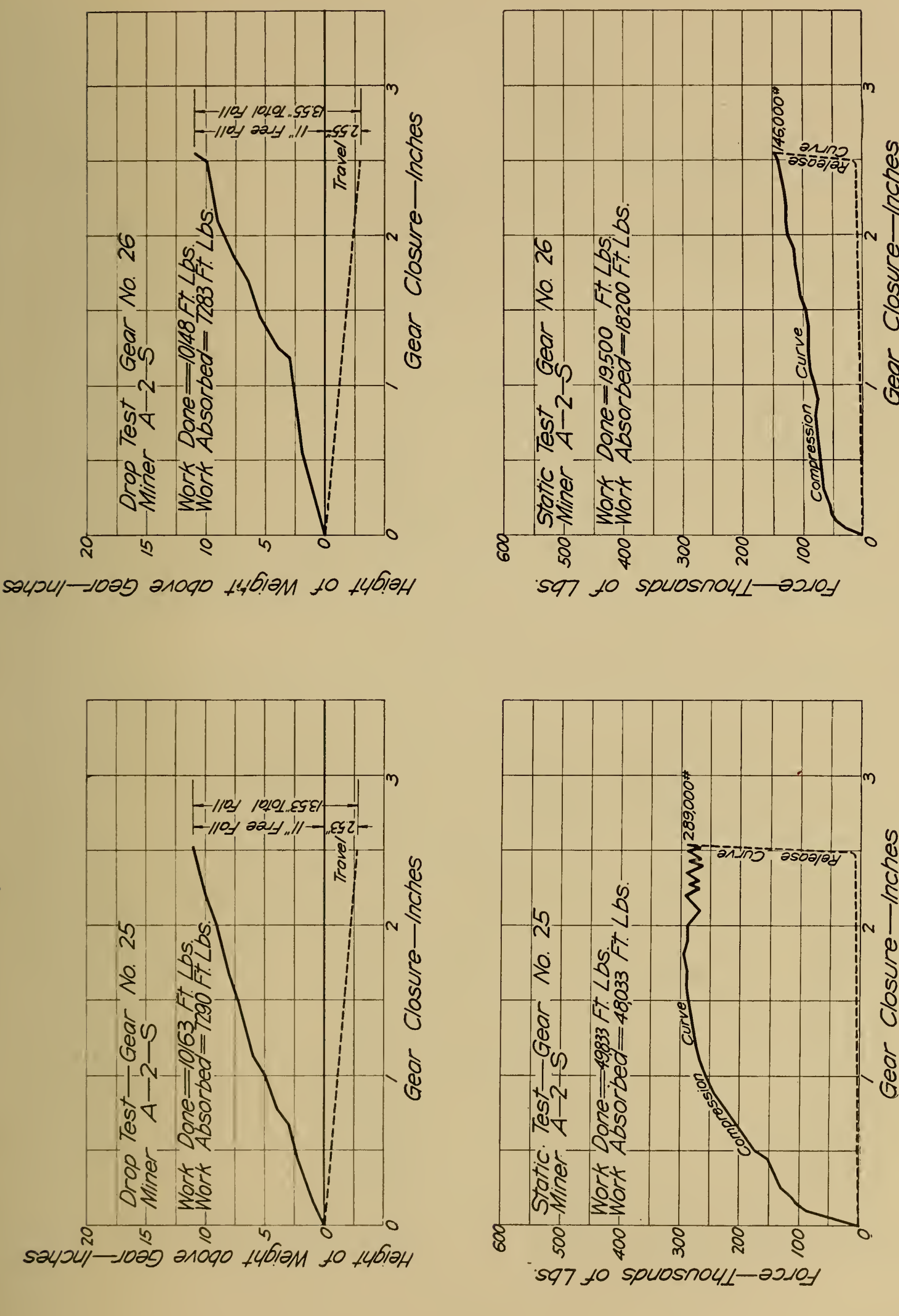


Fig. 25—Drop Test and Static Test Diagrams, Miner Type A-2-S

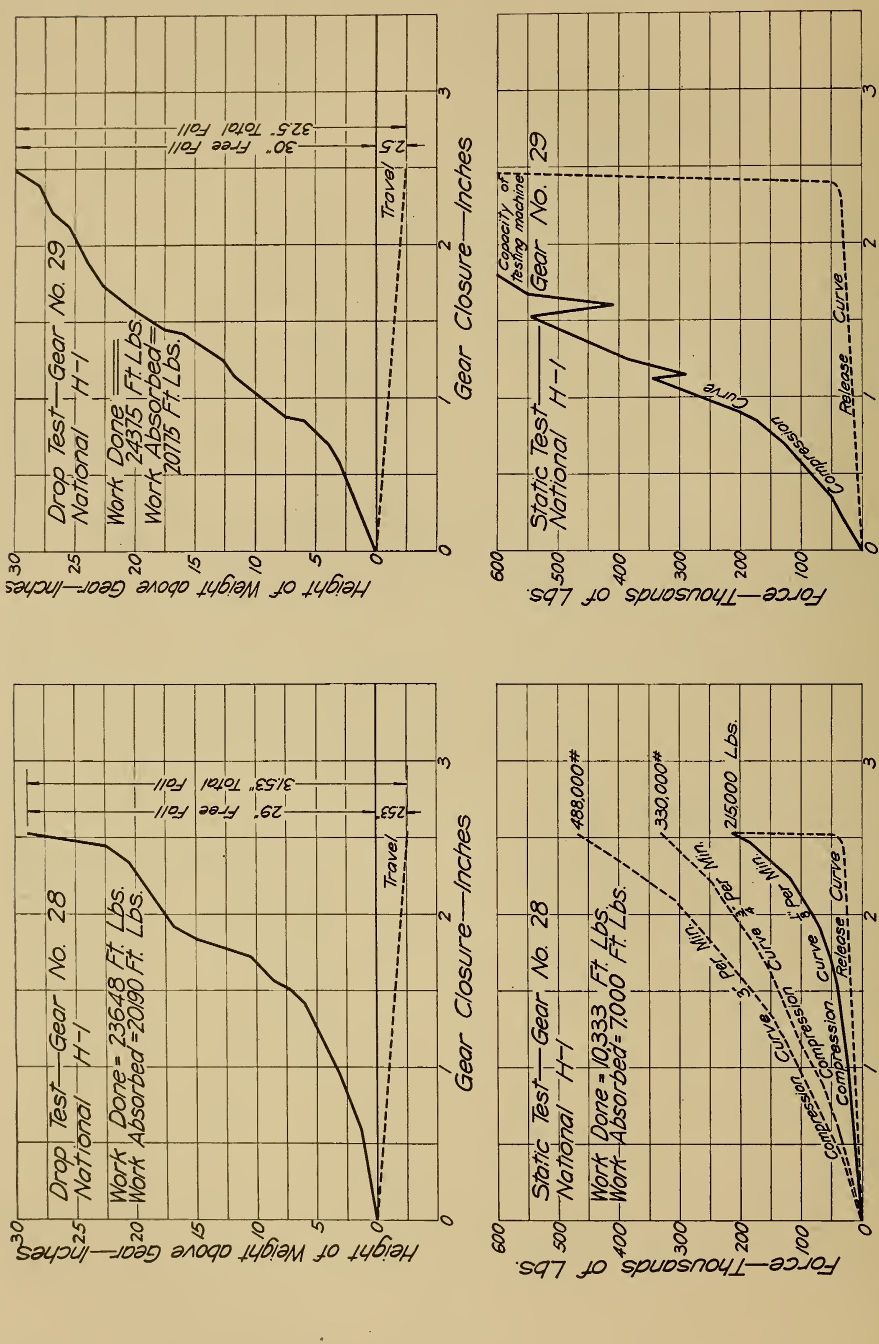


Fig. 26—Drop Test and Static Test Diagrams, National Type H-1

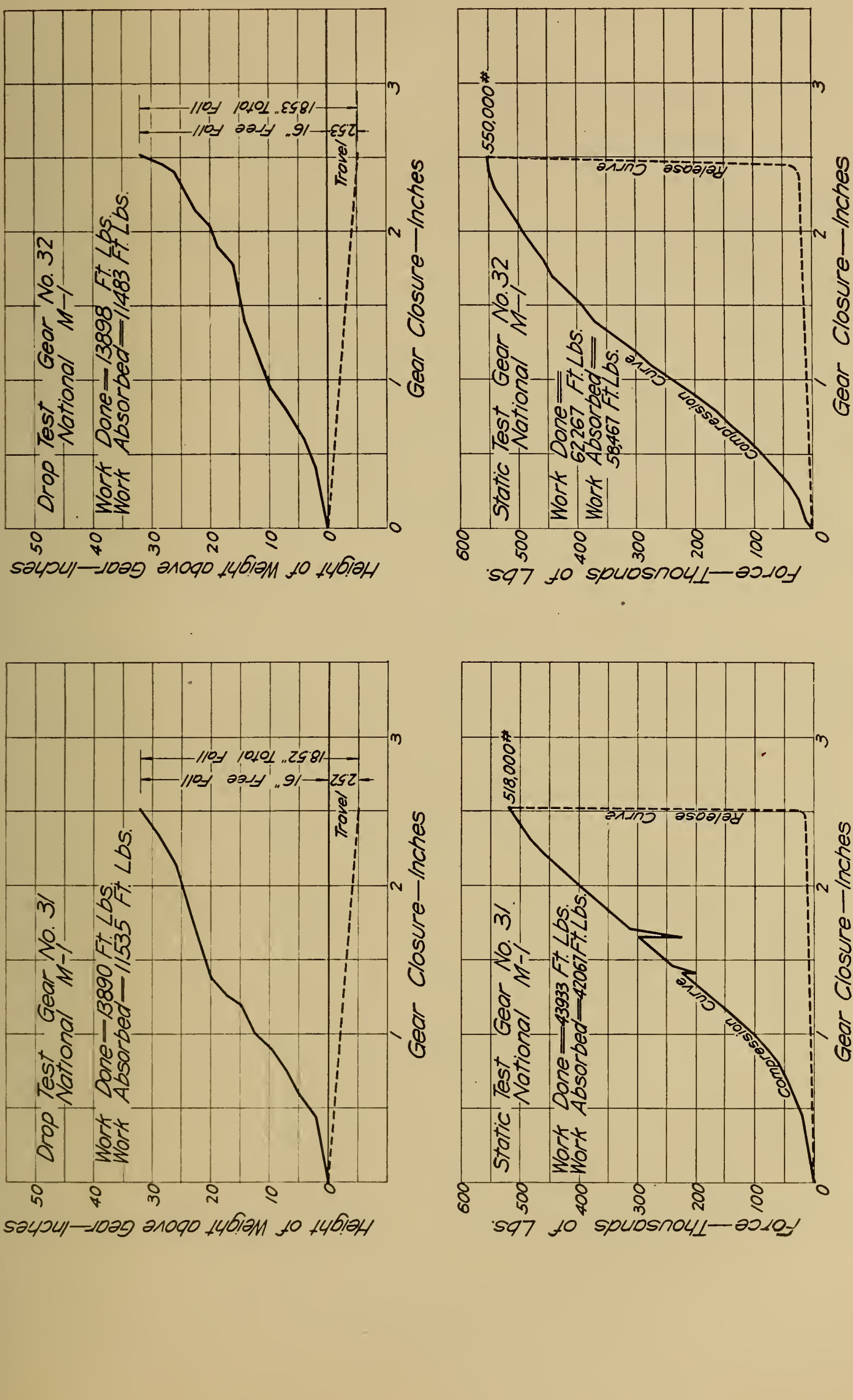


Fig. 27—Drop Test and Static Test Diagrams, National Type M-1

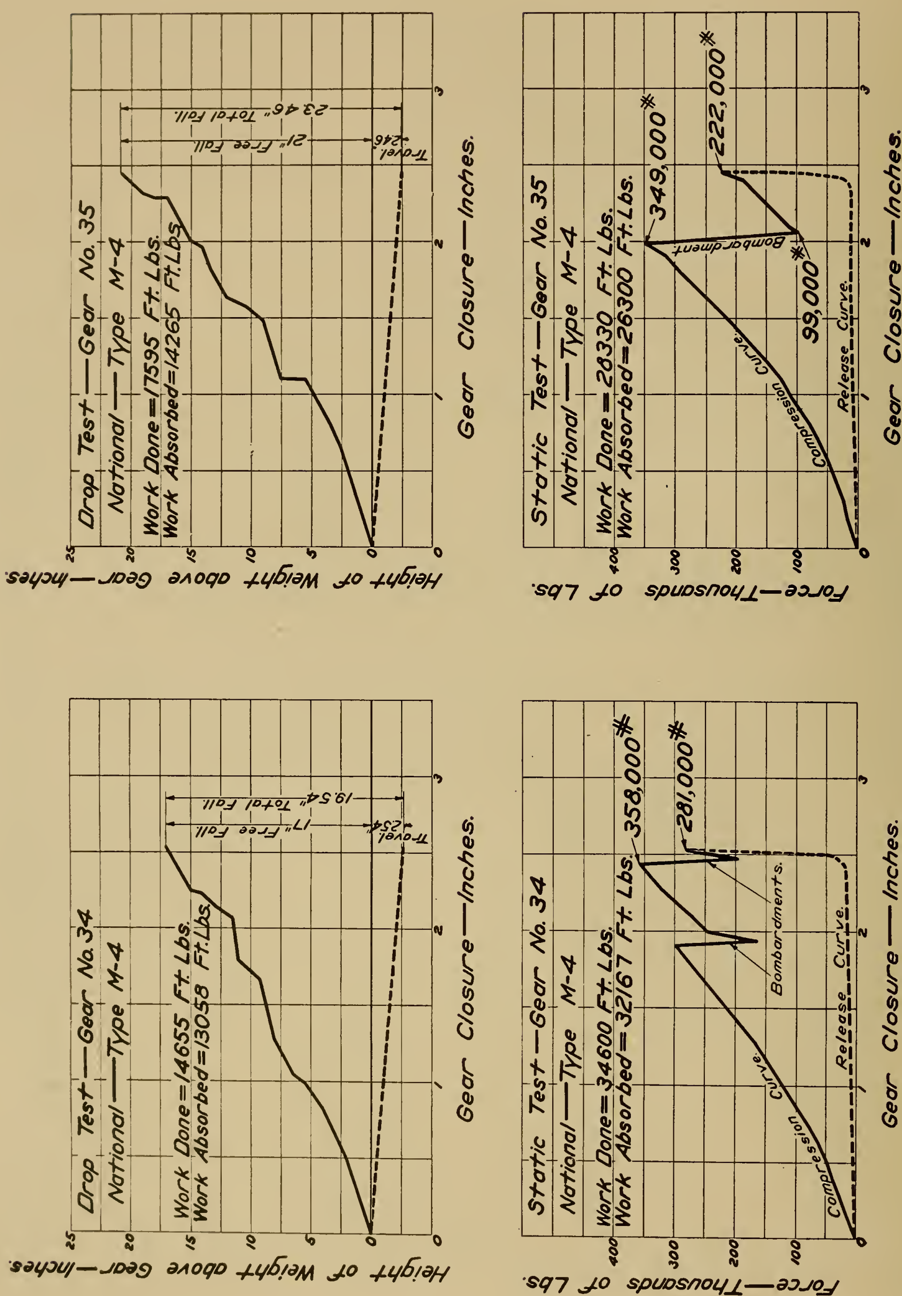


Fig. 28—Drop Test and Static Test Diagrams, National Type M-4

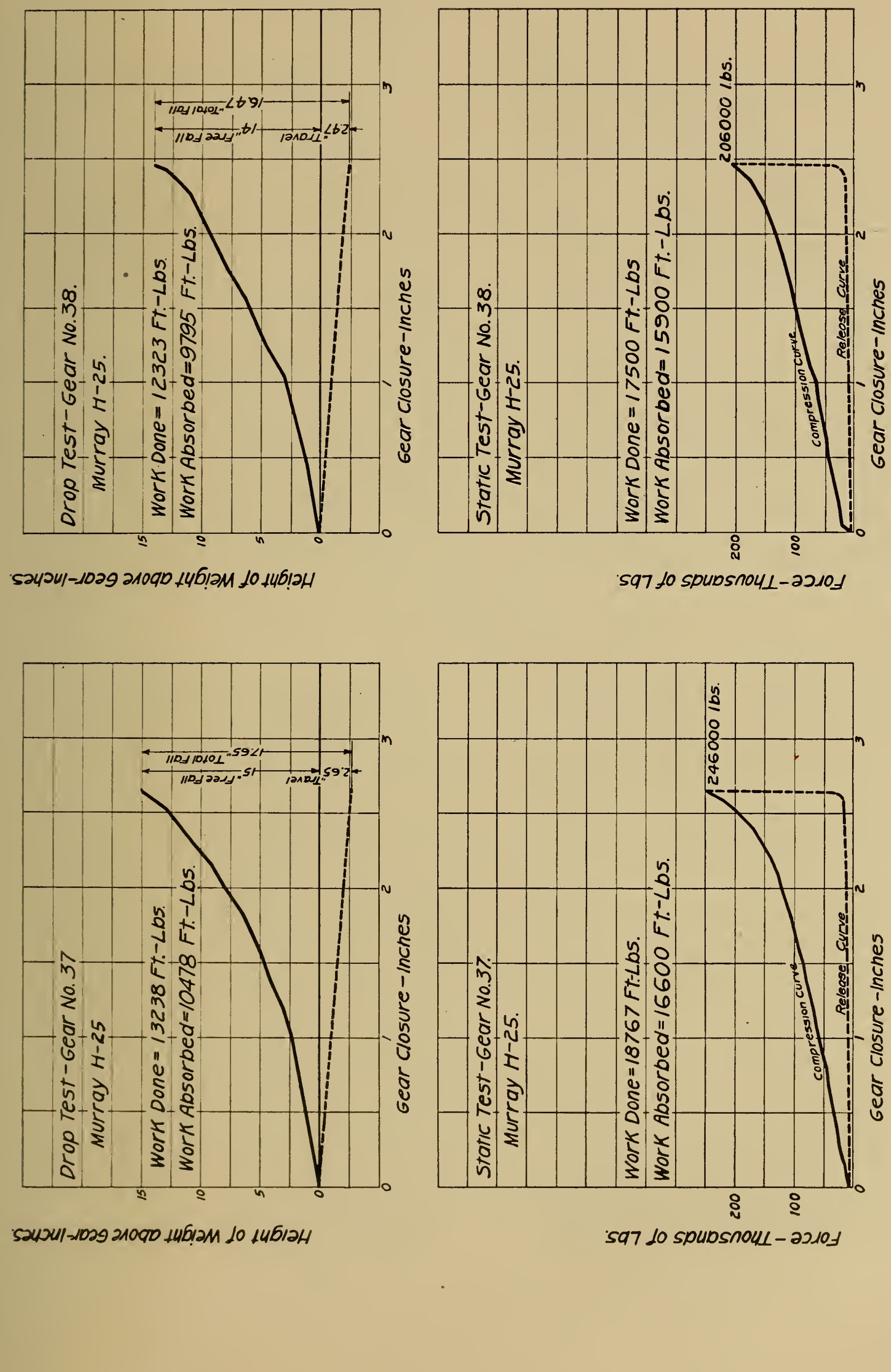


Fig. 29—Drop Test and Static Test Diagrams, Murray Type H-25

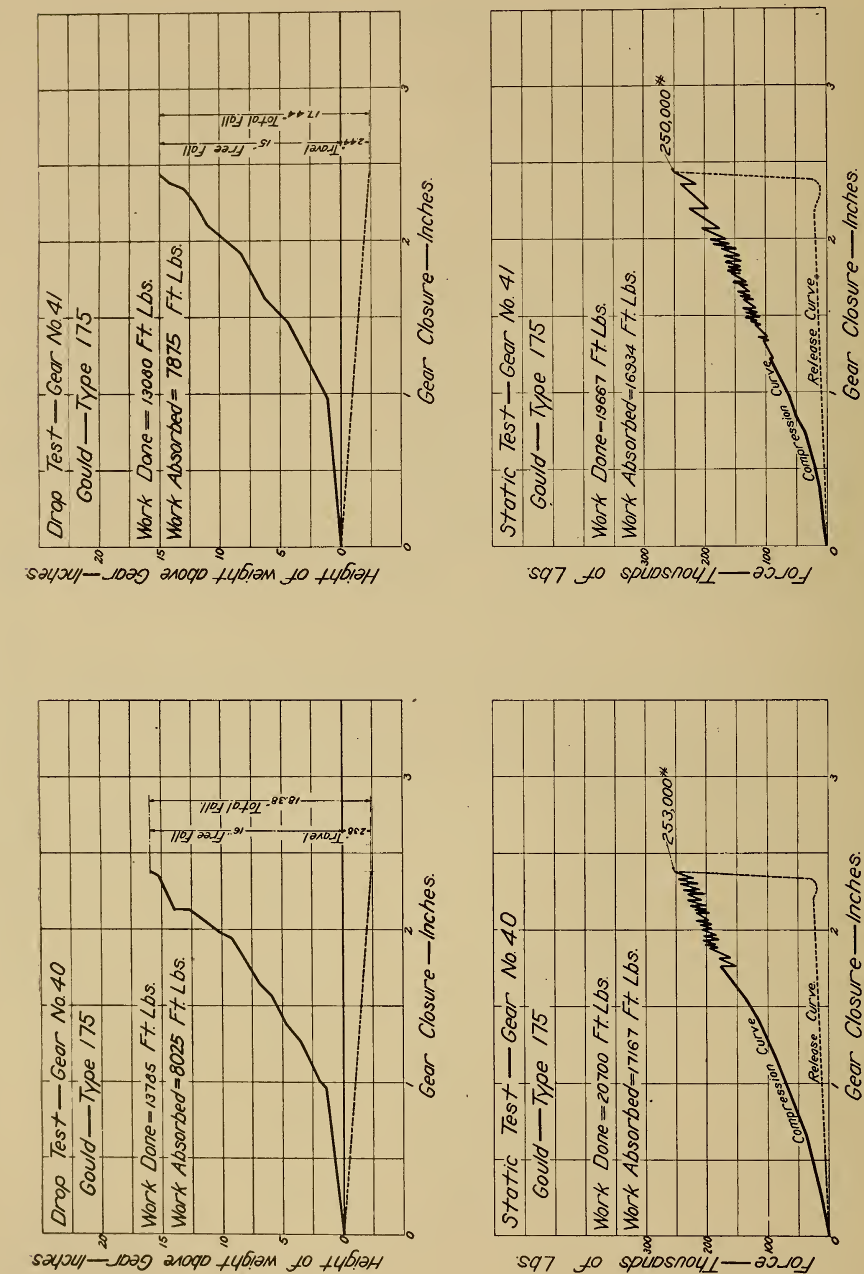


Fig. 30—Drop Test and Static Test Diagrams, Gould Type 175

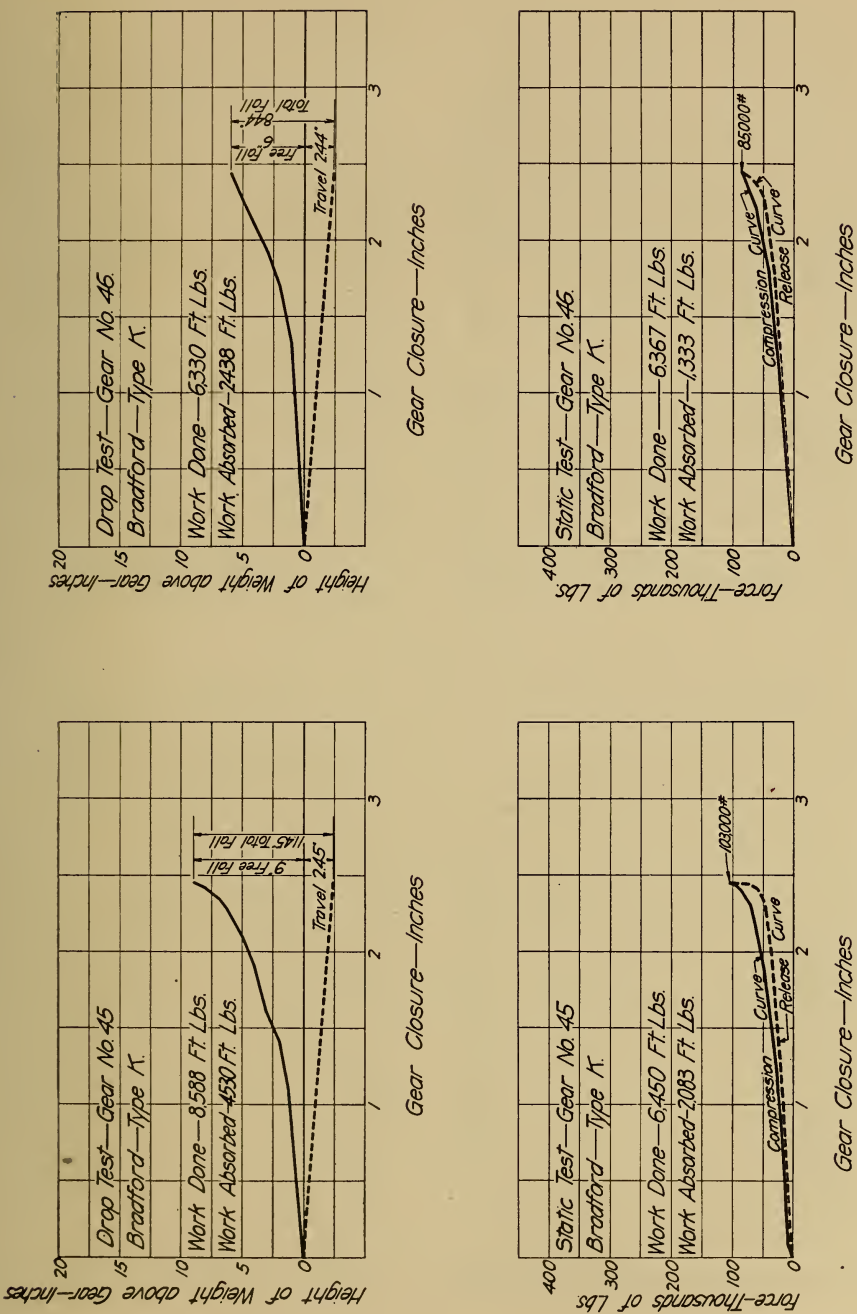


Fig. 31—Drop Test and Static Test Diagrams, Bradford Type K

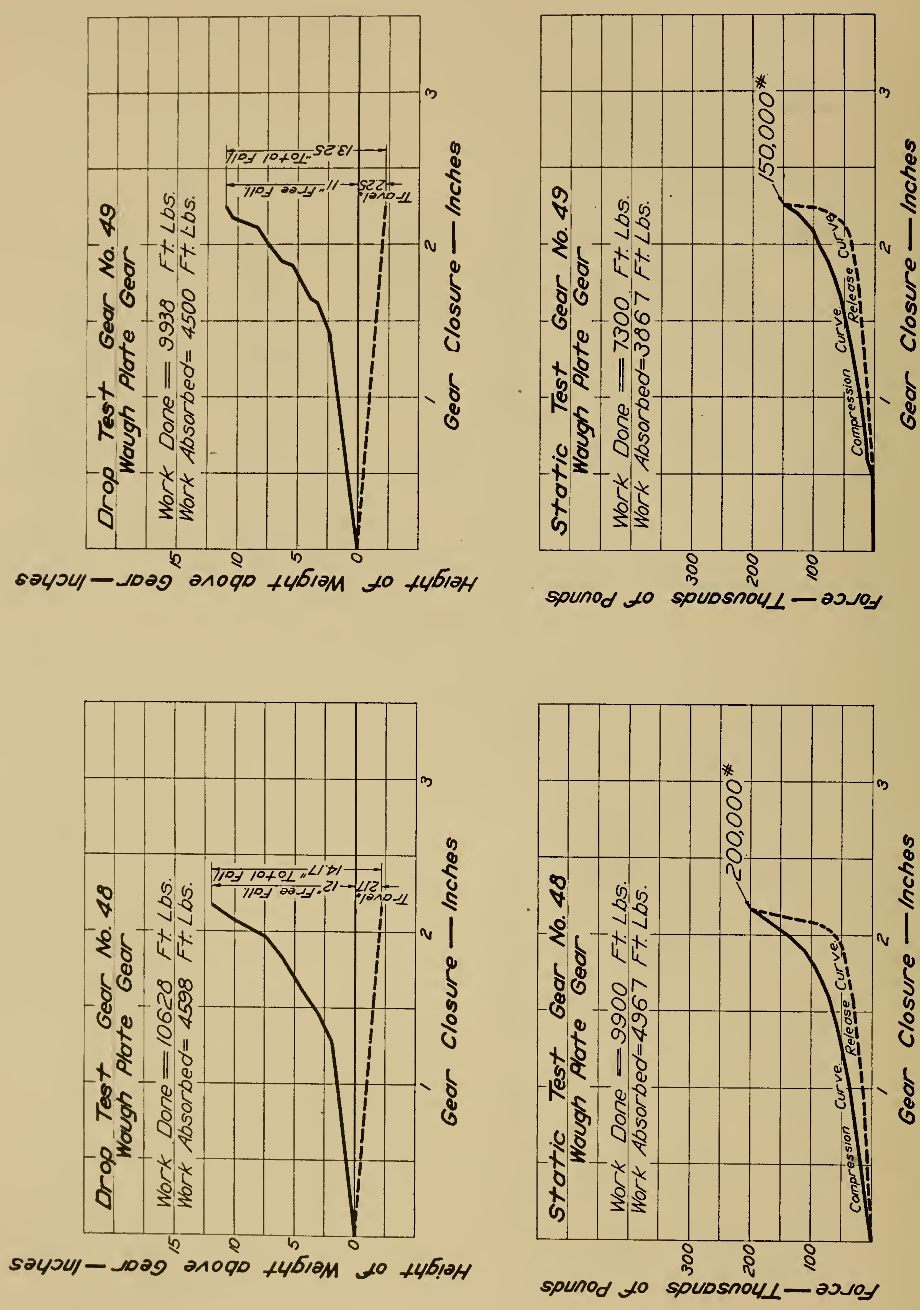


Fig. 32—Drop Test and Static Test Diagrams, Waugh Plate Gear

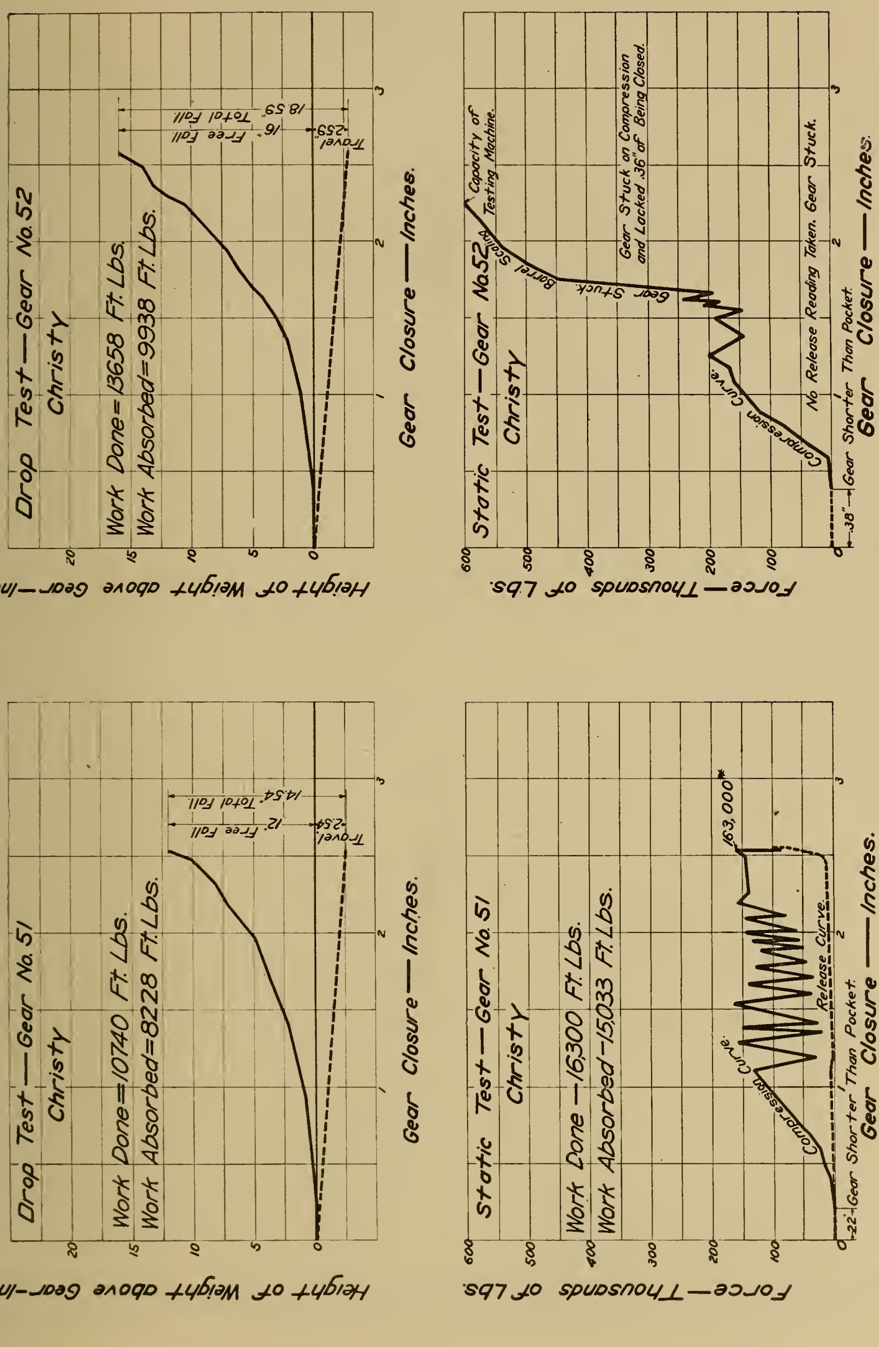


Fig. 33—Drop Test and Static Test Diagrams, Christy Draft Gear

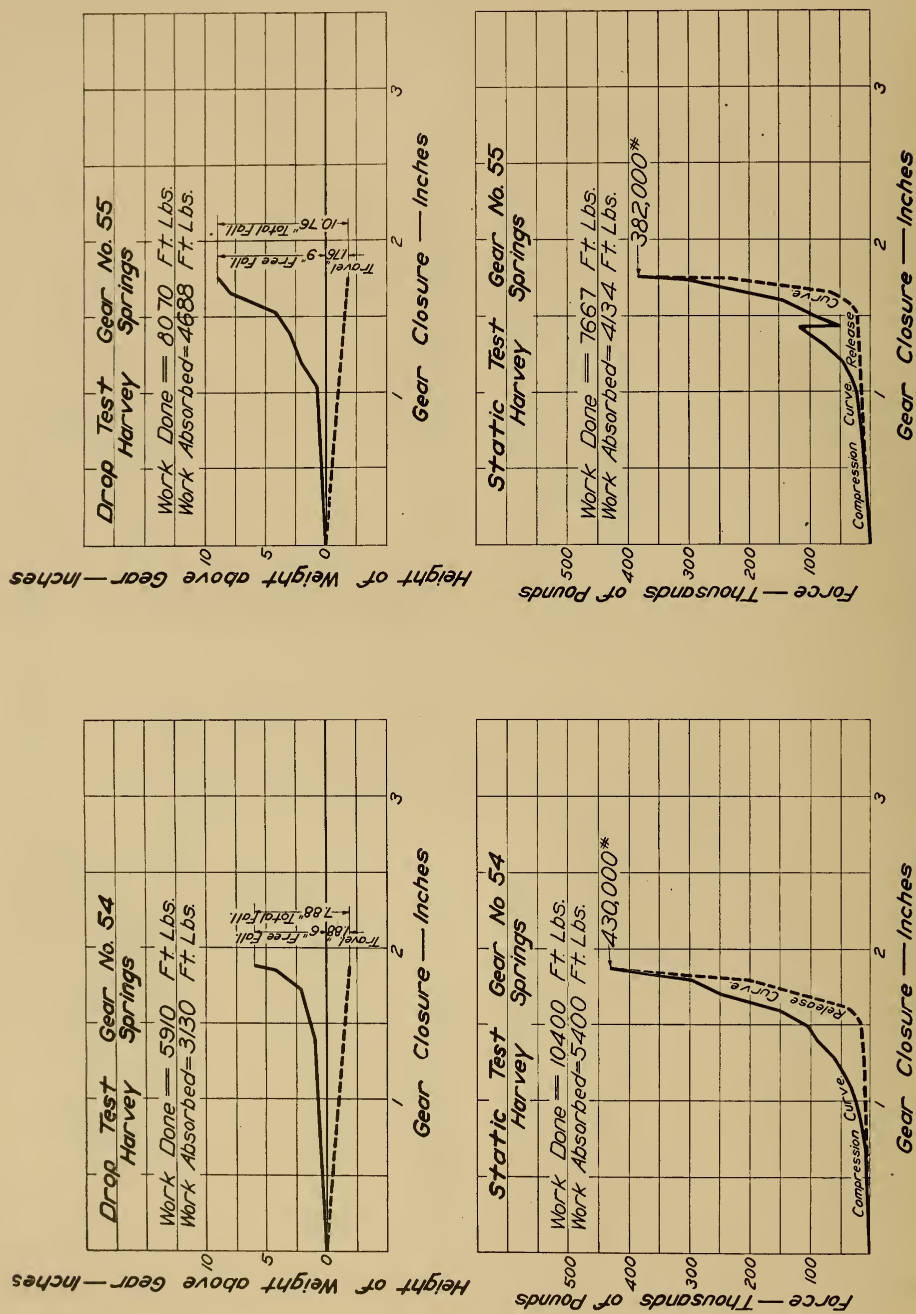


Fig. 34—Drop Test and Static Test Diagrams, Harvey Friction Springs

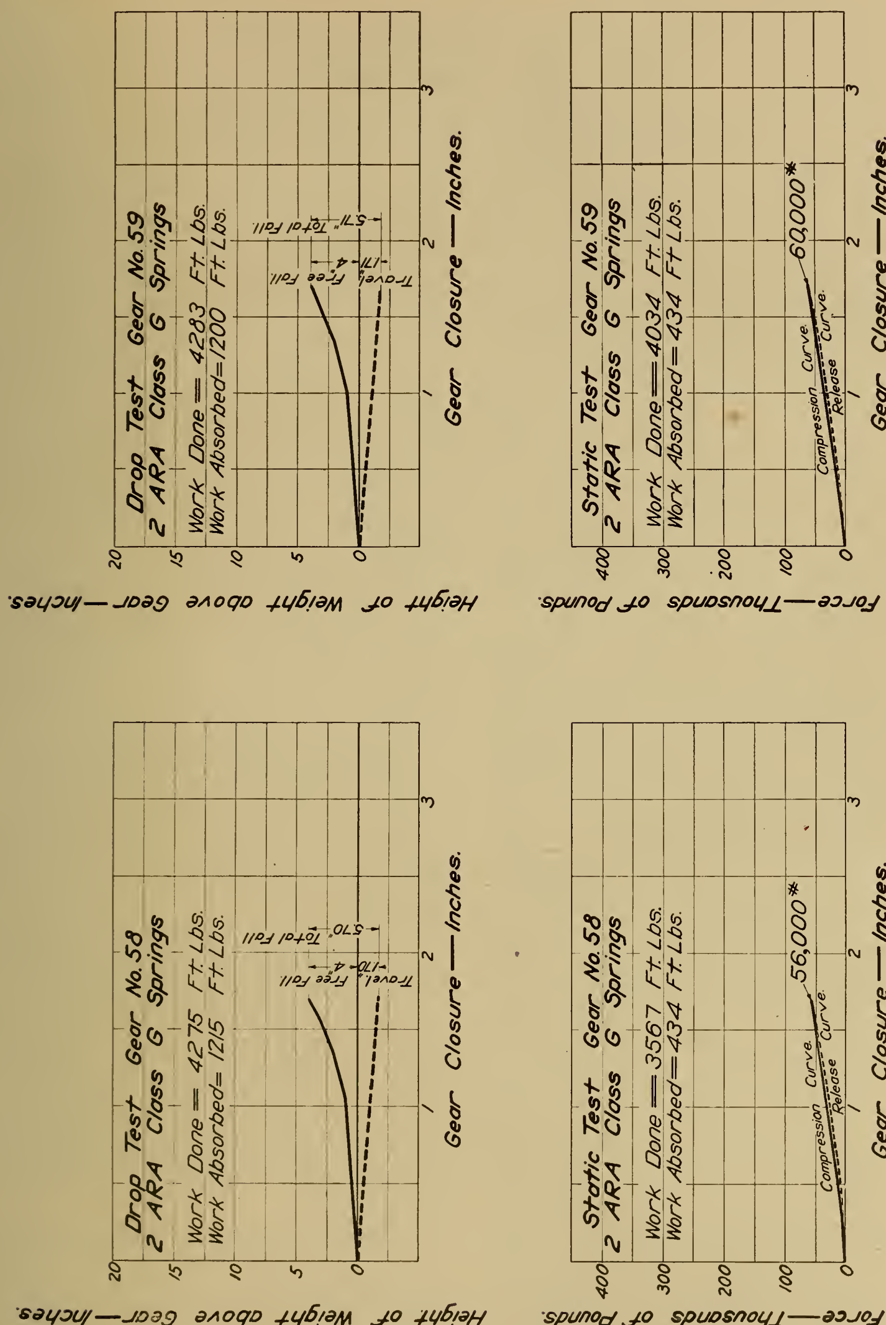


Fig. 35—Drop Test and Static Test Diagrams, A. R. A. Class G Springs

9,000 LB. DROP TESTS

Friction Surfaces Coated with Foreign Material

It has been repeatedly noticed when taking down gears in car repair yards that the friction surfaces, while usually worn to a good bearing contact, are not in the same clean and perfect condition as that of protected test gears. On the contrary, there is frequently found an actual coating or glazing of hard, black material that can sometimes be scraped off with a knife. This is probably an accumulation of particles of metal, coal dust and rust.

In order to obtain some knowledge of the effect of foreign material upon the friction surfaces, one of each type of gear was taken apart, after completing the original drop and static tests, and the friction surfaces were dampened and sprinkled with a mixture of pulverized coal and iron rust. The gears were then reassembled with the parts in their original positions and the dampness allowed to dry out. Each gear was then put under the 9,000 lb. drop and the closing point determined in as few blows as possible, and the gear then given 12 blows at or just below the closing point.

The gear was then taken apart and the free material wiped off with clean waste. In almost every instance the friction faces were now found to be covered with a hard, glazed coating similar to that found in service. This was removed with clean, sharp sandpaper and the surfaces again wiped off with clean waste. This in every instance left the friction surfaces in as perfect looking condition as could be desired. The action of the gears immediately after this is therefore especially interesting, as in almost every case the careful cleaning of the surfaces did not increase the capacity, and quite a number of blows were required to restore the gears to their original capacities. In several instances it was impossible to entirely restore the gears. It will be seen from a study of the results of these tests that any gear might by this method be made to show an extremely low capacity, even though all parts of the gear were of full size and to gage and the friction surfaces apparently in perfect condition. At the same time, an inferior gear could be in apparently no better or more favored condition and yet show decidedly higher results.

The table, Fig. 36, has been prepared to show the results of this test, and the following paragraphs will more completely explain the values given in the several columns.

Column 1 is self-explanatory.

Column 2 identifies the single gear of each type used for this test.

Column 3 gives for ready comparison the original total fall required to close this same gear.

Column 4 gives the total fall required to close the gears when first operated with the mixture upon the friction surfaces.

Column 5 gives the total fall required to close the gears with coated friction surfaces, but after receiving 12 blows.

Column 6 gives the total fall required to close the gears immediately after sandpapering and thoroughly cleaning the friction surfaces.

Column 7 gives the number of blows that had to be given each gear to restore it to its original capacity as given in Column 3. In this work of restoration each gear was operated until the original capacity

MAKE AND TYPE OF GEAR	TEST GEAR NUMBER	TOTAL FALL OF 9000 LBS. WEIGHT TO CLOSE: IN ORIGINAL CONDITION	WITH COATED SURFACES: FIRST CLOSURE	WITH COATED SURFACES: AFTER TWELVE BLOWS	FIRST CLOSURE AFTER CLEANING SURFACES	NUMBER OF BLOWS REQUIRED TO RESTORE GEAR TO ORIGINAL CAPACITY AFTER CLEANING SURFACES	REMARKS
①	②	③	④	⑤	⑥	⑦	⑧
WESTINGHOUSE D-3	1	18.50"	10.50"	10.5"	11.5"	18	
WESTINGHOUSE NA-1	6	27.06"	13.06"	14.06"	14.1"	41	
SESSIONS K	10	18.17"	8.2"	9.2"	13.2"	15.2" After 30 blows	Gear could not be fully restored.
SESSIONS JUMBO	13	26.10"	16.1"	16.1"	21.1"	25.1" After 35 blows	Gear could not be fully restored.
CARDWELL G-25-A	16	21.75"	15.75"	12.75"	18.8"	28	
CARDWELL G-18-A	19	20.25"	13.25"	17.00"	17.75"	30	
MINER A-18-S	22	17.51"	11.51"	11.51"	12.5"	40	
MINER A-2-S	25	13.55"	7.55"	7.55"	10.6"	27	
NATIONAL H-1	28	31.53"	15.5"	17.5"	17.5"	34	
NATIONAL M-1	31	18.52"	9.52"	13.52"	13.5"	17	
NATIONAL M-4	34	19.54"	8.54"	14.50"	18.00"	4	
MURRAY H-25	37	17.65"	9.7"	10.7"	11.7"	54	No hard coating on surfaces as in other gears, but considerable wear.
GOULD 175	40	18.38"	11.4"	11.4"	13.4"	52	
BRADFORD K	45	11.45"	9.5"	9.5"	10.5"	11	
WAUGH PLATE	48	14.17"	11.2"	11.2"	11.2"	12.2" After 15 blows	Gear could not be fully restored.
CHRISTY	51	14.32"	9.3"	9.3"	10.3"	30	
HARVEY 2-8" x 8" SPGS.	54	7.88"	5.9"	5.9"	6.9"	13	
COIL SPRINGS 2-8"x8" CLASS G	A.R.A. Springs not included in this test.						

Fig. 36—Performance of Gears with Coated Friction Surfaces (Drop Test)

was restored or until three successive blows resulted in no increase of capacity.

It will be understood that the blows given in these tests were kept within the capacities of the gears, the general practice being to work the gears slightly below the solid point rather than above it.

The results of these tests indicate that while new gears in laboratory tests may show acceptable capacities, depreciation may occur in service, not only from eventual wear, but from an immediate coating of the friction surfaces. Iron rust and coal dust, with or without sweated friction surfaces or rain, will undoubtedly greatly reduce the capacity of a gear in service as in these tests. It has not been possible to investigate this as thoroughly as desired, but some little work of this character has been done in conjunction with the Engineer of Tests of the Norfolk & Western Railway. Experiments were made upon five different types of gears which had been in service for approximately three years each on as possible, after which a building up or restoration test was made by giving the gear additional blows until no further increase in capacity could be obtained. No attempt was made to clean the surfaces of these gears prior to the building up tests, as the test was intended to develop what recovery might be effected by simply working the gear. The gears were all in good condition as to wear, and would in every instance have been so declared upon surface inspection. The results of these tests are shown in the table, Fig. 37, a description of the columns of which follows:

MAKE AND TYPE OF GEAR	NUMBER OF GEARS TESTED	AVERAGE RESULTS—9000 LBS. DROP			
		DROP TEST VALUE OF NEW GEAR TOTAL FALL	TOTAL FALL REQD. TO CLOSE GEARS WHEN FIRST REMOVED	GEAR RESTORED TO—TOTAL FALL	NUMBER OF BLOWS NECESSARY FOR RESTORATION
1	2	3	4	5	6
MINER A-18-A	10	19.9"	16.4"	17.5"	18
MINER A-59	2	27.0"	18.0"	20.5"	32
SESSIONS K	2	19.3"	8.6"	9.0"	14
SESSIONS JUMBO	2	28.1"	15.0"	16.5"	21
NATIONAL H-1	10	31.2"	19.4"	26.9"	32

Fig. 37—Drop Tests of Friction Gears Which Were Taken Out of Service, Norfolk & Western Railway

N. & W. 100-ton coal cars. The gears were carefully removed from the cars so as not to disturb the deposit and glazing on the friction surfaces and were put in tight, individual boxes and carried immediately to the drop test machine. The actual fall required to close the gears in their service condition was determined in as few blows

Column 1—In this column the gear types are identified, and in explanation the Miner A-18-A gear is the same as the Miner A-18-S of the U.S.R.A. tests, except that the A-18-A has 3 in. travel and the A-18-S has 2½ in. travel. The Miner A-59 gear is an especially long gear, not usable in the standard pocket and hence not included in the U.S.R.A. tests. The Sessions K, the Sessions Jumbo, and the National H-1 are identical with the same types in the U.S. R.A. tests.

Column 2—This column indicates the number of gears carried through the N. & W. tests.

Column 3—This column gives for ready reference the total average drop test value of the several types, when new and in good condition, as found in the U.S.R.A. tests. The Miner A-18-A is taken the same as the A-18-S. For the Miner A-59 a value is taken from previous tests of these gears.

Column 4—This column gives the average total fall, including the travel, required to just close the gears when first tested after removal from service. These figures therefore represent the value of the gear as in actual service, after a period of three years' use, as heretofore explained.

Column 5—This column gives the average total fall to which it was possible to build up or restore the gears.

Column 6—This column gives the average number of blows necessary to restore the gears to the falls given in Column 5.

In this test the Sessions gears, which have the friction elements of unhardened cast iron working against unhardened forged steel, showed the greatest percentage of depreciation and the least restoration. The National gear, which has hardened steel friction elements working together, showed the next greatest percentage of depreciation and the greatest restoration. The Miner gears, which have hardened steel friction shoes working against a malleable iron barrel, showed the least percentage of depreciation and necessarily the least percentage of restoration. It does not thus appear that the character, and particularly the hardness of the friction surfaces, influenced this depreciation. On the other hand, the Miner gears were under a heavy initial compression in the cars and the Sessions and National under practically none. It is therefore probable that the tightness of the friction parts may have prevented the entry of the foreign material in the case of the Miner gears. In the case of these N. & W. gears, the friction shoes in the Miner gears were in every instance tight with the gears in position in the cars, while the friction members were loose in every one of the Sessions and National gears. As no measurable wear had occurred in the National gears the manufacturers offer in explanation of this loose condition of the friction members that an inferior lot of springs had been used, with consequent set and loosening of the friction parts in the car. In the case of the Sessions gears the designs provide for loose friction blocks in the car. Further investigation along the lines of gear depreciation, due to foreign material on the friction surfaces in service, should be made.

DESTRUCTIVE TESTS

Immediately after the tests with coated friction surfaces, the same gears, one of each of the types included in the program, were tested to at least partial destruction under the 9,000 lb. drop, the gears being supported on a solid anvil. In each instance successive blows were given from heights beginning at 1 in. free fall of the weight and increasing by 1 in. increments, a record being made of the point at which each gear went solid and of the point at which destruction began, as evidenced by scaling, fracture, bending or shortening of some part of the gear. These tests are of the kind best suited to show the ability of a gear to receive over-solid blows and are designed to penalize weakness instead of putting a premium upon it, as set forth in a preceding chapter of this report. It will be noticed that some of the gears begin to show evidence of distress at a fall of just a few inches above the solid point.

A discussion of the individual performance of the gears in the destructive tests follows:

Westinghouse D-3

Gear No. 1

This gear in the destructive test went solid at 16 in. free fall, and at 20 in. free fall a number of fine cracks were observed at the tops of the convolutions that occur near the lower end of the barrel. The gear was given seven more drops, the last one being a free fall of 27 in. The cracks in the barrel had now opened up and the barrel had bulged at the point of the cracks to $9\frac{3}{4}$ in. diameter, whereas the diameter here before the test was 9 in. The barrel had shortened $\frac{5}{8}$ in. Neither the free height nor the friction height of the gear, however, was reduced, as neither the friction spring nor the preliminary spring had taken permanent set of any consequence. The travel of the gear had increased from $2\frac{1}{2}$ in. to $3\frac{1}{8}$ in., due to the shortening of the barrel. This increased travel, it should be noted, is accompanied under these extraordinary circumstances by what would undoubtedly prove, upon repetition, to be a disastrous deflection of the friction springs. A destructive value of 23.8 in. has been given this gear, this figure being determined by the general rule outlined at the close of this chapter.

Westinghouse NA-1

Gear No. 6

This gear was carried up to a final blow of 36 in. free fall, the gear going solid at 24 in. free fall. At 28 in. the barrel started to scale on the ends just opposite the slots in the sides of the ends of the friction spacers. At 34 in. the barrel showed a crack at the bottom of one of the key slots. After the test the free length of the gear, which is also the friction length, was found to have been reduced $\frac{3}{8}$ in., being now $\frac{7}{16}$ in. less than the pocket dimension. The barrel had shortened $\frac{3}{8}$ in., the gear travel remaining the same as originally. The slots for the spacer ends had been reduced $\frac{5}{16}$ in., making the spacers bind and causing the gear to stick at lighter blows. There was no evidence of spacer failures or of the barrel deforming beneath the spacer ends, as occurred in the static tests of gears No. 1 and No. 2 of this same type. The release spring had taken a set of $\frac{3}{8}$ in. To this gear has been given a destructive value of 30 in.

Sessions K

Gear No. 10

This gear was subjected to a maximum blow of 33 in. free fall. During the test the gear stuck and failed to release on a number of the lighter blows. The gear went solid at 13 in. free fall and at 15 in. free fall the barrel started to scale and to bulge. After this test the barrel was found to have shortened $\frac{11}{16}$ in. and the friction box opening to have elongated $\frac{9}{32}$ in. The outer coil spring had taken a set of $\frac{5}{16}$ in. and the free length of the gear had been reduced by ½ in., being now ⅜ in. less than the pocket length and the friction length 1⅜ in. less than the pocket length. Because of the fact that this test gear, along with others of the same type, heretofore began to scale in the regular drop tests before the closing point was reached, the destructive value has been reduced below that denoted by this test, the destructive value being placed at 1 in. over the average solid value, or at 19.8 in.

Sessions Jumbo

Gear No. 13

This gear was carried up to a final free fall of 30 in, going solid at 21 in. The barrel of this gear was slightly cracked through one of the rivet holes in the preceding drop test and attention was therefore particularly directed to this point during the destructive test. At 25 in. the crack started to widen. At 28 in. the friction box began scaling. At 29 in. the crack in the corner of the barrel had opened ⅛ in. and at 30 in. the weight recoiled and the gear jumped enough to allow the recurring fall of the weight to land upon the side of the gear, necessitating a discontinuance of the test. Upon measurement the gear was found to have shortened $\frac{1}{16}$ in. in free length and the friction box to have elongated $\frac{1}{16}$ in., the gear now being $\frac{1}{16}$ in. less in free length than the standard draft gear pocket. In view of the questionable crack developing in this gear prior to this test, the benefit of all doubt has been given it and its destructive value of 32.1 in. is based upon the point at which this crack first started to widen.

Cardwell G-25-A

Gear No. 16

This gear was given drops up to and including a free fall of 32 in., the gear going solid at 18 in. free fall. At 20 in. six cracks had developed in the heads and at 22 in. ten cracks had appeared and the heads were deforming. After the test the gear was measured and it was found that the free length had been reduced ½ in. and the solid length ⅜ in. The free length, however, was still 1 in. greater than the standard pocket dimension, this gear being nominally under a heavy initial compression in the car, as heretofore explained. The heads had been badly deformed and cracked, and had each shortened an average amount of $\frac{7}{32}$ in. The spring rod had bent $\frac{5}{16}$ in., due to the inertia of the springs and spring washers, and had elongated ⅛ in. The outer coil springs had taken an average set of ⅜ in. The friction blocks were not injured. To this gear has been given a destructive value of 20.9 in.

Cardwell G-18-A

Gear No. 19

This gear was given successive drops up to and including a free fall of 32 in., the gear going solid at 17 in. free fall. At 20 in. the top head began to fail and at 23 in. the top surface was depressed. At 26 in. three cracks had developed in the heads. This gear was in somewhat better condition at the completion of this test than the Cardwell G-25-A gear. A destructive value of 22.6 in. has been given this gear.

Miner A-18-S
Gear No. 22

This gear was given successive drops up to and including a free fall of 30 in. The gear in this test went solid at 14 in. free fall. At 19 in. free fall the springs went solid and at 21 in. free fall the barrel began scaling. At 23 in. the barrel began to bend out of line and at 27 in. a crack appeared. After the test the free length of the gear was found to have decreased 3/8 in. and the barrel to have shortened 5/16 in., the free length of the gear being now 1/4 in. less than the standard pocket dimension. There was no breakage of center wedge, friction shoes or rollers. To this gear has been given a destructive value of 26.9 in.

Miner A-2-S
Gear No. 25

This gear was carried up to a final blow of 36 in. free fall, the gear going solid at 10 in. free fall. At 19 in. one friction shoe flaked and showed a slight crack. At 20 in. the barrel began scaling and at 24 in. bulging of the barrel could be detected. At 29 in. one crack developed in the friction end of the barrel and at 30 in. a second crack developed here. After the test the free length of the gear was found to have been reduced by 1 1/16 in., being now 15/16 in. less than the standard pocket length. The friction length was 1 in. less than the pocket length. The barrel had bulged and shortened 3/4 in. and the outer coil spring had taken a set of 13/16 in. The friction end of the barrel had opened slightly and the two cracks mentioned had developed in this portion of the barrel. The friction shoes were each cracked in the roller seats and were cracked and flaked at the ends. The rollers had hammered and seated into the shoes and the center wedge, but the rollers were not injured. To this gear has been given a destructive value of 20.2 in.

National H-1
Gear No. 28

This gear was given successive drops up to and including a free fall of 60 in. in an unsuccessful effort to fracture or deform some part essential to the operation of the gear. It went solid at a free fall of 31 in. At 48 in. two of the columns showed bending and at 52 in. all four columns were bent. At 49 in. the friction spring went solid and the center post of the gear came into action. At 54 in. the friction blocks had become loose. Upon measurement after the test the center post of the gear was found to have shortened 3/16 in. and the friction spring had taken a set of 3/16 in. The free length of the gear had been reduced 1/8 in. and the length when the friction shoes tightened had also been reduced 1/8 in. The gear length at this latter point, however, was still 1/8 in. in excess of the standard pocket dimension and the free length 1/4 in. in excess of it, so that this gear after the test would have been under 1/4 in. total compression and 1/8 in. friction compression in the car. The corner posts had shortened 1/4 in. and the travel of the gear had consequently been increased by this amount. The gear was not damaged in this test except for the set of the spring and the shortened and bent corner posts, which, incidentally, are simply round steel bars of 1 5/8 in. diameter by 19 1/8 in. long and could be readily straightened. The gear after this test was entirely serviceable. In view of the fact, however, that the columns bent at a point 17 in. above the solid point of the gear a destructive value of 48.2 in. has been given this gear. The ability of this gear to withstand punishment is very remarkable.

National M-1
Gear No. 31

This gear was tested up to a final free fall of 48 in., going solid at 17 in. free fall. At 27 in. one of the columns started to bend out of line and at 35 in. three columns were bent, the fourth one bending at 39 in. At 42 in. the center post came into action and at 44 in. the spring went solid and the friction shoes loosened. After the test the columns were found to have bent 1 in. out of line and shortened $\frac{7}{16}$ in. The friction spring had taken a set of $\frac{1}{8}$ in. and the center post had shortened $\frac{5}{16}$ in. The free length of the gear had not been reduced, but the length at which friction starts had been reduced $\frac{3}{16}$ in., this length being now the same dimension ($24\frac{5}{8}$ in.) as the standard draft gear pocket. Except for the bent corner posts, the gear was suitable for service after this test. Inasmuch as the first of these started to bend at a drop of 10 in. above the solid point, this gear has been given a destructive value of 29.2 in. This gear, like the National H-1, shows exceptional ability to withstand severe punishment.

National M-4
Gear No. 34

This gear was given successive blows up to and including a free fall of 48 in., the gear going solid in the test at 17 in. At 23 in. three columns were bending. After the completion of the test, all four columns were bent approximately $\frac{13}{16}$ in. out of line and one of the heads had a small crack in the column guide, due to the bent column. The center column had not come into bearing to assist in taking the solid blow and the spring had not gone solid, although it showed a set of $\frac{7}{8}$ in.

The absolute free height of the gear had been reduced by $\frac{3}{8}$ in., but would still have been under compression in the car. The gear, after this test, would have been entirely serviceable except for the bent corner posts.

In view of the fact that the corner posts began to show bending at 6 in. above the solid point, the destructive value of this gear has been set at 27.5 in.

Murray H-25
Gear No. 37

This gear was given successive blows up to and including a free fall of 26 in., the gear going solid at 15 in. At 20 in. the side members began to scale and at 21 in. bulging could be detected. Also at 21 in. the spring went solid. Upon measurement it was found that the free length of the gear had been reduced $\frac{1}{2}$ in., being now $\frac{1}{8}$ in. less than the standard pocket length. The shouldered side members had shortened $\frac{7}{16}$ in. and had bent and bulged. The wedge-shaped openings in the heads had spread an average of $\frac{1}{8}$ in. To this gear has been given a destructive value of 22 in.

Gould 175
Gear No. 40

This gear in the destructive test was carried up to 32 in. free fall, going solid at 15 in. free fall. At 19 in. free fall the barrel began scaling in the reduced lower portion and at 20 in. bulging of the barrel could be seen. At 26 in. the mouth of the barrel cracked slightly. After this test the barrel was found to have shortened $\frac{3}{8}$ in. and the barrel mouth to have spread $\frac{3}{16}$ in. The free length of the gear was reduced $\frac{5}{8}$ in. and the friction length $\frac{7}{8}$ in., the gear travel having been reduced from $2\frac{7}{16}$ in. to $2\frac{5}{16}$ in. and the friction members having become loose, the free length being $\frac{9}{16}$ in. less than the standard draft gear pocket length and the friction length $\frac{13}{16}$ in.

less. The outer coil spring had taken a set of $\frac{9}{16}$ in. To this gear has been given a destructive value of 22.1 in.

Bradford K

Gear No. 45

This gear in the destructive tests was carried up to a final free fall of 24 in. The gear during the drop test immediately preceding it had been given free falls up to and including 10 in. and at this point in the previous test the top head had cracked and had been deformed $\frac{3}{16}$ in. All of the springs had been solid and had taken permanent set. As the destructive test proceeded the gear showed increasing failure and deformation. At the conclusion of the test the springs had taken a set of ¾ in. and the gear had been shortened ⅝ in. The heads were badly cracked and deformed. To this gear has been given a destructive value of 11.8 in.

Waugh Plate Type

Gear No. 48

This gear was given blows up to a final free fall of 32 in., the gear going solid at 10 in. free fall. Some set of the plates had taken place at 12 in. and at 14 in. the gear was loose in the standard pocket by $\frac{3}{32}$ in. No parts were broken, but the free height of the gear was reduced ½ in. and a number of the plates were given a noticeable camber. The gear, however, even though loose in the pocket, was serviceable. A destructive value of 15.9 in. has been set for this gear.

Christy

Gear No. 51

This gear was given blows up to a final free fall of 42 in., going solid at 12 in. The barrel started scaling at 20 in. At 24 in. bulging of the barrel could be detected. After the test the barrel was found to have shortened ¾ in., the free length of the gear having been reduced ⅝ in., being now $\frac{27}{32}$ in. less than the pocket length and the friction length $1\frac{3}{32}$ in. less than the pocket length. The barrel was bulged 1⅛ in. at points in its sides where the metal is cut away to provide space for the spring, seven cracks having developed in the barrel at these points. The outer coil spring had taken a set of ¼ in. To this gear has been given a destructive value of 27.6 in.

Harvey Springs

Gear No. 54

Two Harvey 8 in. x 8 in. friction spring units were set side by side, in twin fashion, and were given successive blows up to and including 40 in. free fall in an effort to break a spring. Except that at 32 in. a small corner of no consequence broke off the end of one coil, no breakage occurred. Set, however, was noticed much earlier. The friction coils when received were each 8⅛ in. in height, and at the beginning of this test were 8 in. and $8\frac{1}{16}$ in. free height. After the 11 in. drop the friction coils both stood at $7\frac{15}{16}$ in. height. After the 18 in. drop they stood at 7⅝ in. and after the test at 7⅛ in., each having taken 1 in. set during the test and being ⅞ in. less than the pocket length. To this gear (two 8 in. x 8 in. Harvey springs) has been given a destructive value of 14.5 in.

Two A. R. A. Class G Springs

Gear No. 57

These springs were set up side by side, in twin fashion, upon the solid anvil of the 9,000 lb. drop machine. During the regular drop tests the springs took an average set of $\frac{1}{16}$ in. Upon further testing more pronounced set occurred, at 6 in. free fall, the average being $\frac{3}{16}$ in. per spring. They

MAKE AND TYPE OF GEAR	TEST GEAR NO.	9000# WEIGHT: TOTAL FALL REQ'D. TO CLOSE GEAR IN THIS TEST	9000# WEIGHT: ADDITIONAL FALL BEYOND CLOSING POINT REQUIRED TO START FAILURE	DEVELOPEMENT OF FAILURE	AVG. TOTAL FALL 9000# WEIGHT REQUIRED TO CLOSE ONE COMMERCIAL GEAR OF THIS TYPE.	DESTRUCTIVE VALUE ASSIGNED TO THIS TYPE OF GEAR
1	2	3	4	5	6	7
WESTINGHOUSE D-3	1	18.5"	4"	RAPID	19.8"	23.8"
WESTINGHOUSE NA-1	6	27.0"	4"	RAPID	26.0"	30.0"
SESSIONS K	10	15.2"	2"	RAPID	18.8"	19.8"
SESSIONS JUMBO	13	24.1"	4"	SLOW	28.1	32.1
CARDWELL G-25-A	16	20.8"	2"	RAPID	18.9"	20.9
CARDWELL G-18-A	19	20.3"	3"	RAPID	19.6	22.6
MINER A-18-S	22	16.5"	7"	MEDIUM	19.9"	26.9
MINER A-2-S	25	12.5"	7"	MEDIUM	13.2"	20.2"
NATIONAL H-1	28	33.5"	17"	SLOW	31.2"	48.2"
NATIONAL M-1	31	19.5"	10"	SLOW	19.2"	29.2"
NATIONAL M-4	34	19.5"	6"	SLOW	21.5"	27.5"
MURRAY H-25	37	17.7"	5"	MEDIUM	17.0"	22.0"
GOULD 175	40	17.4"	4"	MEDIUM	18.1"	22.1"
BRADFORD K	45	11.5"	1"	RAPID	10.8"	11.8"
WAUGH PLATE	48	12.2"	2"	SLOW	13.9"	15.9"
CHRISTY	51	14.3"	8"	SLOW	19.6"	27.6"
HARVEY 2-8"x8" SPGS.	54	7.9"	5"	MEDIUM	9.5"	14.5"
COIL SPRINGS 2-8"x8"-CLASS G	57	5.8"	2"	RAPID	5.8"	7.8"

Fig. 38—Performance of Gears in Destructive Tests

were carried up to a final fall of 12 in. and at this point the average set was ¾ in. No breakage occurred.

A destructive value of 7.8 in. has been given for the two springs, but it should be noted that this is for conditions where the springs are not protected, but the coils are allowed to go solid.

Summary of Destructive Tests

The table, Fig. 38, has been prepared to show the results of these tests and to grade the gears as to destructive value. It is quite possible that a repetition of lighter blows would in each instance have produced failure, but it is believed that no great error is made by this comparative grading.

The several columns of the table are described as follows:

Column 1 is self-explanatory.

Column 2 identifies by number the gears that were subjected to this test.

Column 3 gives the total fall required to close the gears during this particular test. In some instances this varies slightly from the figure obtained in the original drop tests, due usually to the fact that some of the gears could not be fully restored in the immediately preceding tests with coated friction surfaces.

Column 4 gives the additional height from which the 9,000 lb. weight was dropped, reaching this by increments of 1 in. from the solid point, before visible distress of the gears began.

Column 5 has been inserted to denote whether the failure from this point on developed slowly or rapidly, under constantly increasing falls.

Column 6 gives for reference the figure accepted as the average total fall required to close a new commercial gear of this type, when in good condition, this column being the same as Column 10, Fig. 16.

Column 7 gives the comparative destructive values assigned the several types. This figure is obtained by adding to the average drop test value of the type of gear as given in Column 6, the over-solid values in Column 4. Thus in the case of the Westinghouse D-3, gear No. 1 of this type was subjected to this test. During the test gear No. 1 went solid at a total fall of 18½ in. and at a total fall of 22½ in., or 4 in. above the solid point, the barrel started to fail. Accordingly the destructive value of this gear has been set by adding 4 in. to the average total fall figure 19.8 in., giving a destructive value of 23.8 in. The same practice has been followed for all gears.

RIVET SHEARING TESTS

The draft gears for the U.S.R.A. (United States Railroad Administration) cars were purchased on the requirement that they be of "150,000 lb. capacity" and the Mechanical Committee later defined a 150,000 lb. capacity draft gear in the following words:

"A 150,000 lb. draft gear should be defined as one that will sustain a drop of 16 in. (including travel of gear) of a 9,000 lb. weight, without shearing the rivets of one or both lugs, which are to be secured to suitable supporting members by nine $\frac{1}{2}$ in. rivets of .15 carbon or under, driven in $\frac{9}{16}$ in. holes."

A representative number of gears of each type used on U.S.R.A. cars were selected at random and subjected to the above test. The average of the results for each type was used to determine whether or not that type of gear met the terms of the specifications. In these tests the gears were supported upon a solid anvil and the weight was dropped from successive heights, increasing by 1 in. increments until the rivets sheared.

In testing the Sessions K gears, the highest capacity gears sheared the rivets at a lower drop than the lower capacity gears. In five instances the rivets sheared before the solid point of the gear was reached. In three instances the rivets sheared at the point of gear closure; and in but two instances did it require a blow from above the solid point to shear the rivets. In three instances the rivets were sheared at a point below the specification requirement when the successive blows by 1 in. increments were given the rivets. In each of the cases, however, when the gear was again set up and a single blow given from a height sufficient to produce a total fall of 16 in. the rivets did not shear. Thus, one of these gears, when given blows increasing by 1 in. increments, sheared the rivets at a total fall of 11 in., and when it was immediately thereafter given a single blow from a total fall of 16 in. the rivets were not sheared. The rivets in this re-test sheared at the next blow, or at a total fall of 17.2 in.

For the Sessions gears it required on the average 2.7 in. less fall to shear the rivets than to close the gears. In the Westinghouse D-3 gears the rivets usually sheared 1 in. above the solid point, although in a few instances it required an over-solid blow of 2 in. to produce shear. In the Cardwell G-25-A gears it required on the average an over-solid blow of 3.2 in. to shear the rivets. In one instance a 4 in. over-solid blow was necessary. In the Murray H-25 gears it required an average over-solid blow of 2.4 in. to shear the rivets.

Considering the Westinghouse, Cardwell and Murray gears, it will be noted that the number of over-solid blows required to shear the rivets is in inverse relation to the over-solid sturdiness or destructive value of the gears as given in the table, Fig. 38. The short travel of the Sessions K gear necessitates a higher ultimate resistance, so that the elastic limit of the $\frac{1}{2}$ in. rivets is passed before the gear goes solid. In considering this test it should be remembered that the eighteen $\frac{1}{2}$ in. rivets ($\frac{9}{16}$ in. when driven) have a shearing area of 4.47 sq. in., giving an ultimate shearing value of approximately 189,000 lb., with an elastic limit in shear of approximately 135,000 lb. In practice the rear draft lugs each have twelve $\frac{7}{8}$ in. rivets in $\frac{15}{16}$ in. diameter holes, or a shearing area of 16.57 sq. in., with an ultimate shearing value of 700,000 lb., or an elastic shearing limit of 500,-000 lb.

The table, Fig. 39, has been prepared to show the results of these ½ in. rivets shearing tests made for the acceptance of gears for U.S.R.A. cars.

In order to show for comparison how all of the gears perform in this test, and in order to study the specifications in the light of a full knowledge of each particular gear, one of each type of gear in the tests was subjected to the ½ in. rivet shearing test. This was done after the car-impact tests and was the final test given the gears.

RESULTS OF ½" RIVET SHEARING TESTS. GEARS FOR U.S.R.A. CARS.- 9000LB. DROP.							
MAKE AND TYPE OF GEAR	NUMBER OF GEARS TESTED	TOTAL FALL TO CLOSE GEAR			TOTAL FALL TO SHEAR RIVETS		
		MAXIMUM	MINIMUM	AVERAGE	MAXIMUM	MINIMUM	AVERAGE
1	2	3	4	5	6	7	8
WESTINGHOUSE D-3	18	21.6"	18.6"	19.9'	22.6"	19.6"	21.0
SESSIONS K	10	23.1"	18.1"	20.5"	22.1"	11.0"	17.8"
CARDWELL G-25-A	5	17.6"	15.6"	16.6"	20.6"	19.5"	19.8"
MURRAY H-25	7	17.8"	16.8"	17.6"	20.8"	17.8"	20.0"

Fig. 39—Results of ½-in. Rivet Shearing Tests. Draft Gears for U. S. R. A. Cars 9,000-lb. Drop

The results are given in the table, Fig. 40, the several columns of which are described as follows:

Column 1 is self-explanatory.

Column 2 identifies by number the gears that were subjected to this test.

Column 3 gives the original total fall required to close each gear. During this test care was taken to see that all gears were up to this original capacity.

Column 4 gives the free fall required to shear the rivets of one or both lugs. This height of fall was reached through successive blows increasing by 1 in. increments.

Column 5 gives the actual amount of gear closure obtained in this test with the free fall of Column 4.

Column 6 gives the total fall required to shear the rivets, this being the sum of the quantities in Columns 4 and 5.

Column 7 denotes whether one or both lugs sheared, this, however, being of secondary interest.

In each of these tests the rivet samples were analyzed and the carbon content in no instance exceeded .15, the usual average ranging from .09 to .12.

Some of these results will at first thought appear inconsistent, but a more careful study will show that the results in general are approximately what should be expected when gears of different travels and capacities are tested upon undersized rivets. Thus the gears generally may be divided into two classes: those closing at four miles per hour or more in the car-impact tests and those closing at less than four miles per hour. Seven types as follows fall in the higher class:

MAKE AND TYPE OF GEAR	TEST GEAR NUMBER	TOTAL FALL REQUIRED TO CLOSE THIS GEAR SOLID ANVIL:	FREE FALL REQUIRED TO SHEAR RIVETS	TRAVEL OF GEAR WHEN RIVETS SHEARED	TOTAL FALL REQUIRED TO SHEAR RIVETS	ONE OR BOTH LUGS SHEARED
(1)	(2)	(3)	(4)	(5)	(6)	(7)
WESTINGHOUSE D-3	2	19.50"	17"	2.47"	19.5"	ONE
WESTINGHOUSE NA-1	7	26.00"	21"	2.66"	23.7"	BOTH
SESSIONS K	11	20.06"	11"	1.45"	12.5"	ONE
SESSIONS JUMBO	14	27.06"	14"	2.10"	16.1"	BOTH
CARDWELL G-25-A	17	20.75"	20"	2.75"	22.8"	ONE
CARDWELL G-18-A	20	18.29"	18"	3.29"	21.3"	ONE
MINER A-18-S	23	19.52"	17"	2.47"	19.5"	ONE
MINER A-2-S	26	13.53"	11"	2.53"	13.5"	ONE
NATIONAL H-1	29	32.50"	9"	1.00"	10.0"	ONE
NATIONAL M-1	32	18.53"	17"	2.53"	19.5"	ONE
NATIONAL M-4	35	23.46"	17"	2.30"	19.3"	BOTH
MURRAY H-25	38	16.47"	16"	2.47"	18.5"	BOTH
GOULD 175	41	17.44"	16"	2.44"	18.4"	BOTH
BRADFORD K	46	8.44"	8"	2.44"	10.4"	ONE
WAUGH PLATE	49	13.25"	11"	2.25"	13.3"	ONE
CHRISTY	52	18.21"	12"	1.95"	14.0"	ONE
HARVEY 2-8"x8 SPGS	55	10.76"	8"	1.76"	9.8"	ONE
COIL SPRINGS 2-8x8"-CLASS G	58	5.70"	7"	1.70"	8.7"	ONE
SOLID STEEL BLOCK			4"		4.0"	

Fig. 40—Performance of Test Gears in ½-in. Rivet Shearing Tests. 9,000-lb. Drop

National H-1.
National M-1.
National M-4.
Miner A-18-S.
Westinghouse NA-1.
Sessions Jumbo.
Sessions K.

It will be noted that in no case did a gear of this class require an over-solid blow to shear the ½ in. rivets. In six cases the rivets sheared before the gears went solid and in the remaining case the rivets sheared at the solid point. Furthermore, as might be expected, the gears of short travel usually sheared the rivets earlier than those of equal capacities and longer travel.

A more clear understanding of this action will be had from the diagrams of Fig. 41. At the top of this figure is shown a straight line diagram of a gear of 34.2 in. drop capacity and of 3¼ in. travel. The ultimate resistance of this gear is 189,000 lb. and the eighteen ½ in. rivets should shear at the same value, or just when the gear goes solid. Next there is shown a diagram of a 2 in. travel gear of the same ultimate resistance, 189,000 lb., and with this gear also the rivets should just shear at the solid point, but which in this case would be at 21 in. drop instead of 34.2 in. Next is shown the diagram of a gear of 2 in. travel but of the same capacity (34.2 in. drop) as the 3¼ in. travel gear at the top of the figure. The ultimate resistance of the gear would in this case be 307,000 lb. and the rivets, which have the value of 189,000 lb., should shear at 1.23 in. gear travel or at a drop of but 12.9 in. This gear, therefore, which is of the identical capacity as the 3¼ in. travel gear, will shear the rivets at 12.9 in. drop, whereas the 3¼ in. travel gear will require 34.2 in. drop. This increase in drop is due solely to the increased travel of the gear with consequent decrease in ultimate resistance. At the bottom of Fig. 41 are shown superimposed diagrams of two gears, each of 2½ in. travel, but the one of 26.2 in. drop capacity and the other 39.3 in. drop capacity, or just 50 per cent increase. In the case of the lighter capacity gear the rivets do not shear until the gear goes solid or at 26.2 in. fall. In the case of the larger capacity gear the rivets would shear at 17.6 in. drop. Here is shown how by simply increasing the capacity of the gear 50 per cent the ½ in. rivets are caused to shear 13.1 in. lower than with the lighter capacity gear. These conditions are for straight line gears and for shearing the rivets at a single blow. When a succession of blows is given from varying heights the difference becomes even more marked, as a heavier capacity gear begins to punish and permanently deform the light rivets early in the test.

The above principles are reflected in the test results. Thus the Sessions K gear and the Cardwell G-25-A were of practically equal drop test capacity but different travels, namely, $2\frac{1}{16}$ in. and 2¾ in. respectively. The Sessions K gear ($2\frac{1}{16}$ in. travel) sheared the rivets at 12.5 in., while the Cardwell (2¾ in. travel) sheared them at 22.8 in. Again, the National H-1 gear sheared the rivets at 10 in. fall, while the National M-1 gear sheared them at 19.5 in. fall. Here the travels of the gears are the same, but one gear had a drop test capacity of 32.5 in., while the other had 18.5 in.

An effort was also made to test all the gears on full-sized rivets, a total of twenty-four ⅞ in. rivets (16.57 sq. in.) being used for the two lugs. It was hoped that by using full-sized rivets an idea could be obtained as to the relative merits of the gears, based upon the shearing point or upon the destruction of the gear prior to shearing. Three types only were attempted on these rivets, as in each instance the gears failed before shearing occurred. Further tests

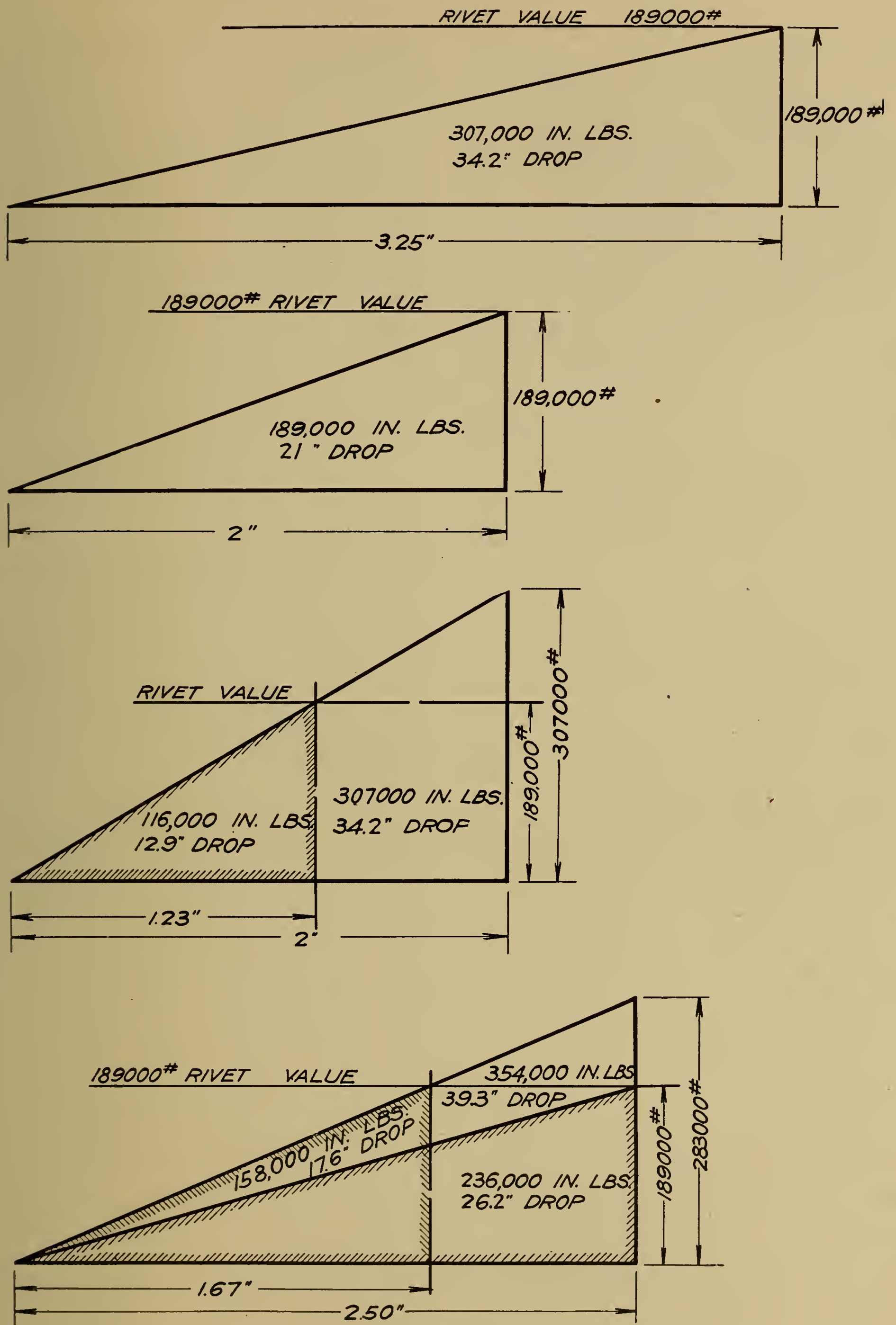


Fig. 41—Diagrams of Rivet Shearing Action of Draft Gears

were prevented by breaking of the set-up, but this limited experience showed the futility of attempting to use full-sized rivets for testing all gears. From this experience and from careful study, it is believed the present ½ in. rivet shearing test is not a fair method of grading gears. Car sills are designed for a load of 500,000 lb. and it therefore should not be expected to hold the draft gear to the limits required by this light test. It is believed possible, however, to develop a rivet shear test to grade gears of all capacities, and investigations have been outlined to develop a test of this character. In this work the following points will be established or disproved:

1. That rivet shearing tests should be designed to show smoothness of action and the ultimate dynamic resistance of the gears.

2. That such tests should not be carried beyond the solid point of the gear, because of the fact that all gears are not of equal rigidity when solid.

3. That the rivets should be of such area that a single blow may be given from a stated height, within the capacity of the gear, and the rivets not be sheared.

4. That the rivets should shear at a blow from not more than a stated height above the solid point; this in order to penalize over-solid weakness of construction.

5. That the tup should not be dropped through successive heights increasing by 1 in. increments because of passing the elastic shearing limit of the rivets before the gear is solid, but that the gear should be set up on lugs or the equivalent and a single blow given from a specified height for that particular gear and the rivets not be sheared. Using a new set of lugs, another single blow should be given from a second specified height at which the rivets should shear.

6. That the number of rivets used and the heights of drop should not be the same for all gears but should be set for each type in accordance with its capacity and travel.

7. That the rivet area and height of fall should be determined by the drop test capacity and travel of the gear.

8. That the test rivets should be of high carbon steel or other material having a high elastic shearing limit; this in order to avoid the uncertainty of the exact shearing point.

9. That when all gears are constructed of equal travel the question of rivet shearing tests will be greatly simplified.

CAR-IMPACT TESTS

In order to obtain an exact knowledge of the action of gears in service, car-impact tests were made, using the same gears as in the foregoing laboratory tests. The results are therefore of especial interest as showing not only the action of the gears themselves under service conditions, but as demonstrating also, for the first time, how laboratory tests compare with service action. The gravity testing plant of the T. H. Symington Company at Rochester, N. Y., was used for these tests. In general, the use of private laboratories of interested companies was avoided, the preference being given to the testing facilities of railroads. This being the only plant of its kind in existence, however, and the Symington Company being interested in the manufacture of draft gear attachments rather than of gears, and having no gear in the tests, the Section availed itself of the opportunity to use the Symington testing facilities.

This plant was originally built for investigating the action of full-sized cars, either loaded or empty, when equipped with different draft gears. The Symington Company, who were practically pioneers in this work, constructed the plant with the prime idea of studying the action of the cars when equipped with different gears rather than investigating the action of the gear itself. As originally constructed and used, much valuable information was obtainable from this equipment although, as in any other impact testing, misleading conclusions as to the relative merits of draft gears could be unintentionally reached by subjecting gears to oversolid car velocities. The extended remarks heretofore made regarding over-solid laboratory testing apply equally as well to this service testing. Over-solid testing should never be done except for discovering weak gear construction. Some of the earlier tests have been of value, however, in showing what slightly over-solid speeds are necessary to produce gear injury and failure. After the owners had made certain changes and additions desired by the Section for a more exact investigation of both gear action and car action, at speeds within the ranges of the gears, the Symington plant was taken over and operated by the United States Railroad Administration for the purpose of the car-impact tests.

The Symington Test Plant

In general this test plant consists of a test track with two full-sized cars which can be caused to collide at any desired velocity, accurate means being provided to record the results. The first portion of the track is inclined in order to impart velocity to one of the cars. This section of track is 147 feet in length and is on a general grade of 12 per cent. An electric hoist is located in a small house at the top of the incline and this hoist through the medium of a puller car, is capable of drawing a loaded 50-ton car up the incline. At the foot of the incline is a 53 ft. section of approximately level track which terminates in a 196 ft. section of track on a 1 per cent ascending grade, this in turn terminating in a 287 ft. section of track on a 1½ per cent grade. The entire test track is thus 683 feet in length, beginning on a 12 per cent descending grade and ending on a 1½ per cent ascending grade. The track is straight throughout its length.

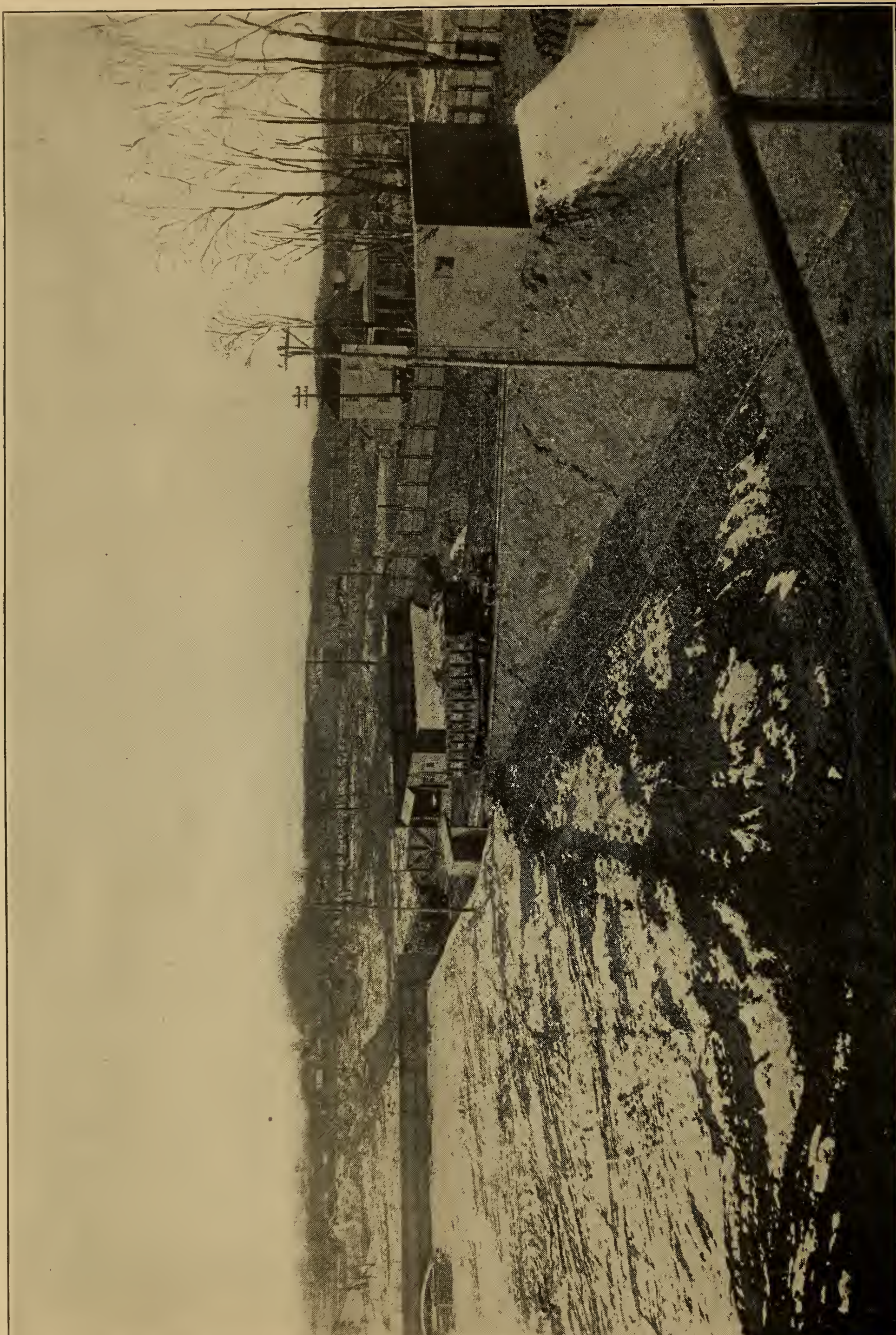


Fig. 42—General View of Symington Gravity Test Plant

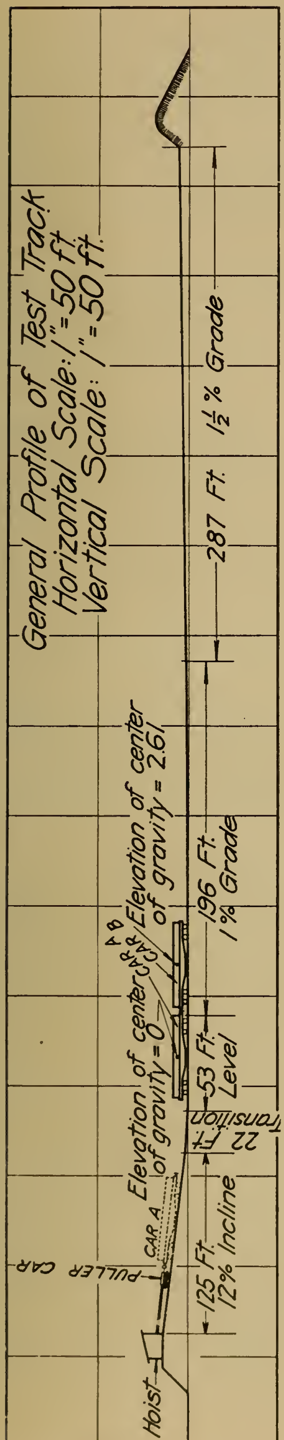


Fig. 43—General Profile of Test Track

Two composite gondola cars of 50 tons capacity are used, one of which, termed for reference car "B," is spotted at a certain point at the beginning of the 1 per cent grade. The other car, termed car "A," is drawn up the incline by means of the puller car and hoist. A movable trip block is clamped on a third rail located alongside the track. The puller car has a projecting trip-lever which strikes the trip block, releasing car "A" and allowing it to roll down the incline. When fully on the level portion of the track, car "A" collides with the standing car "B." Either or both cars may be equipped with draft gears of any type, and both cars are free to follow such movement during and after the draft gear cycle as may result from the use of the particular gear. A general photographic view of this test plant is shown in Fig. 42.

Fig. 43 shows the general profile of the test track with the test cars in positions as at the first instant of impact. While the general condition of the track is good, any local variations in elevation would influence the results of the tests unless considered in the calculations. Accordingly a minute check of the critical portion of the track was made, the levels being taken while moving the loaded cars over the track. Fig. 44 shows in magnified scale the true path of the center of gravity of one of the test cars as it moves along the portion of the track traversed by the cars during the tests. Figs. 45 and 46 show to a still greater magnification the exact paths of the centers of gravity of the two cars over the short portions of the track traversed during the draft gear cycle. With the aid of these profiles, Figs. 44 to 46, it is possible to make a careful and exact study of the energy transferrence from one car to the other, and of that absorbed by the draft gears and cars.

The test cars are 50-ton low side, composite gondolas, 46 ft. 0 in. inside length with fish-belly center and side sills and with a steel frame superstructure. The cars have 2¼ in. floor planking and 3¼ in. side and end planking. Each of the cars has four diagonal floor braces of 5 in. channels at the corners, and each has been supplied in addition with four diagonal braces at the center, extending from the side sills to the center sill, these latter braces being 6 in. by 4 in. by ⅜ in. angles. The light weight of each car is 47,800 lb., and they have been loaded with pig iron to give a total gross load of 143,000 lb. per car. Wood cribbing is arranged inside of the car to hold the lading against shifting. A general view of one of these cars (car B) with its lading is shown in Fig. 47. These cars were reweighed and the lading properly adjusted and distributed before beginning the tests. Care was taken to avoid testing after heavy rainstorms, as it was found that each car took up approximately 1,200 lb. of water during a prolonged rain. The brakes were removed from the cars to avoid any possibility of dragging shoes.

The test cars are equipped with Farlow Two-Key draft gear attachments as shown in the photographic reproduction, Fig. 48, and the test gears may thus be carefully adjusted in the draft gear pocket. In this arrangement the gear is positioned between the arms of the horizontal wrought steel yoke. In buffing, the gear seats against the rear follower, which latter bears against the rear of the yoke. The yoke in turn seats against the cast steel back-stop and bolster center casting. This casting bridges between the sills and ties them together, there being a total of seventy-four ¾ in. rivets supplied for transferring the buffing force from the backstop casting to the center sills in each of the test cars. The draft gear is held to compressed position by means of the second draft key, which in service forms also the front pulling stop for the gear. The regular practice in this form of attachment is to have this key protect the draft gear by allowing it to strike the ends of the slots in the check

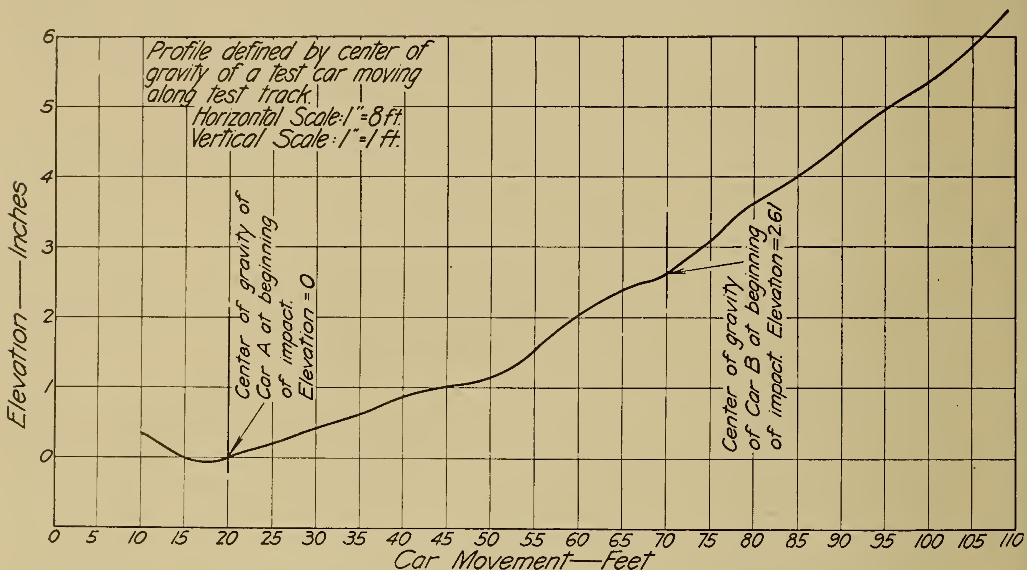


FIG. 44—ENLARGED PROFILE OF TEST TRACK FOR 90 FEET

plates and sills at the same time the gear goes solid. In the tests, however, the slots were lengthened at the rear to prevent this key ever going solid.

A dummy coupler having a flat buffing face and of 16 sq. in. cross sectional area, was used instead of a standard coupler. The artificial looseness or slack resulting between the coupler butt and the front follower was taken up by temporary wedges before each run so that all action, both on compression and release, could be restricted to the gears themselves and be definitely measured and recorded.

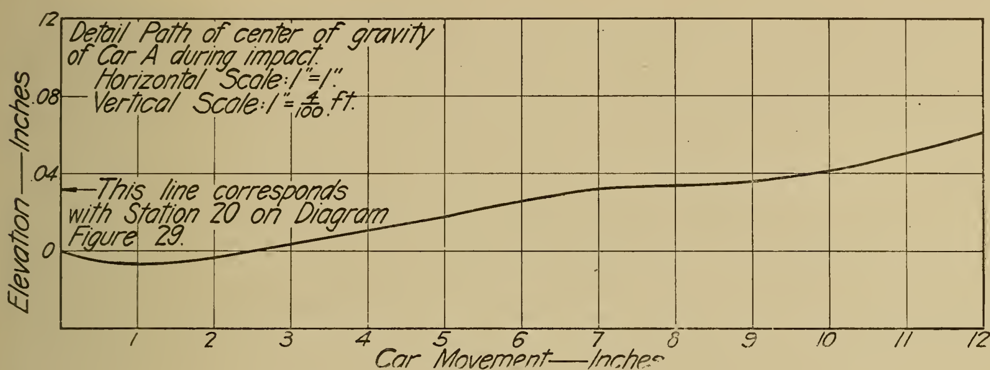


Fig. 45—Enlarged Profile for 12-in. Movement of Car A

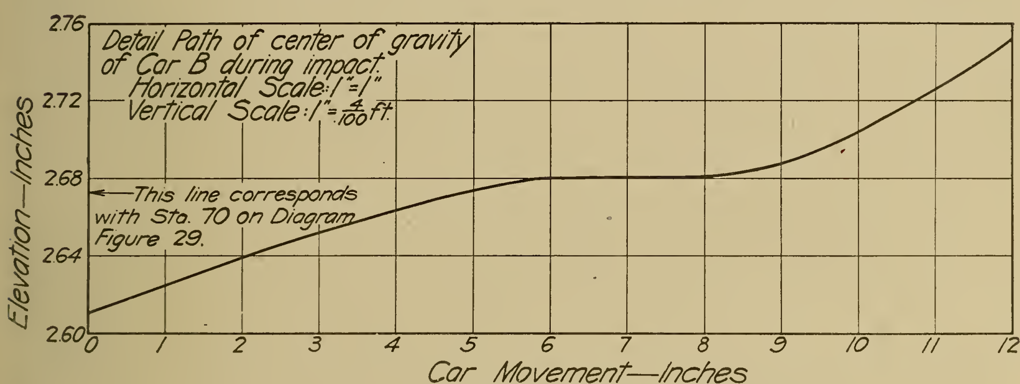


Fig. 46—Enlarged Profile for 12-in. Movement of Car B

The front key, which passes through the key slot of this coupler and through the front slots in the yoke, was used in these tests for supporting and aligning the parts only. The front end of the dummy coupler shank was guided both vertically and laterally. In all tests care was taken to see that the draft gears seated against the second key and not against the coupler key.

Action of Cars During Impact

In any case of car-impact, the first and prime effort is for the velocities of the two cars to equalize; that is, for car A to slow down and car B to speed up. This is caused by the effort of car A to push car B ahead, which continues as long as the velocity of car A is greater than that of

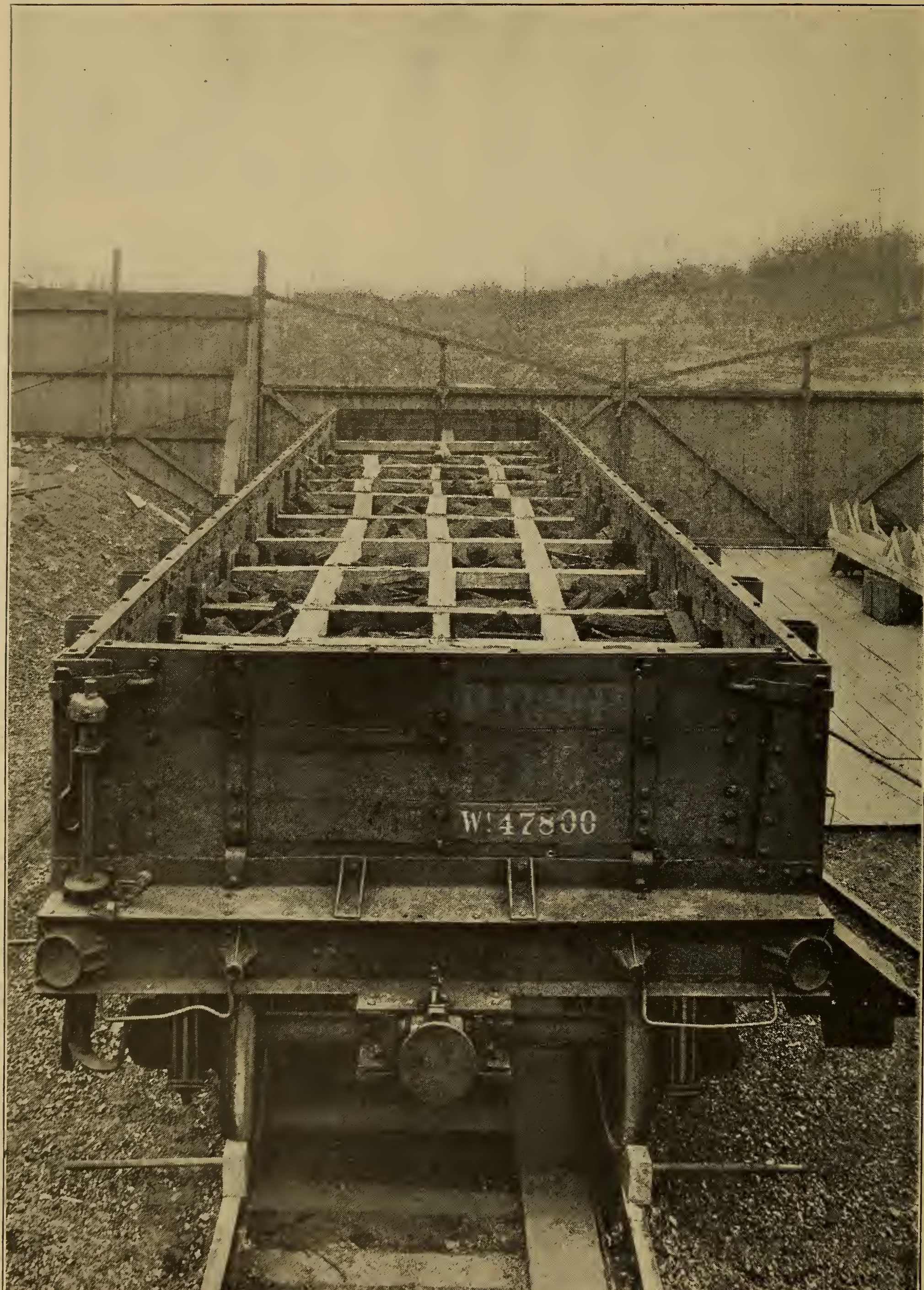


Fig. 47—General View of Car B and Its Lading

car B. This pushing or propelling effort must always result in the compression or yield of some part or parts of the car. The draft gears are supplied for the purpose of providing this yield and to reduce the amount of yield required from the car structure. But in every case of impact, some part or parts of the cars, either draft gears or other more rigid car parts, will continue to compress or yield until the very instant when the velocities are equal. For light impacts, the velocities are usually equalized without compressing the draft gears to their full amount. In the case of an over-solid velocity, the draft gears will first be fully compressed, their resistance slowing down car A and speeding up car B. But in this over-solid case, when the gears are fully compressed, car A has still a greater velocity than car B and it will be apparent that car A will continue to urge car B forward. This results in an impact directly upon the gear housing. The additional work to be done must be accomplished by certain forces working through a certain space represented by the additional yield of the solid parts, the force going directly through the housing to the sills. The couplers, gear housing, and sills must now continue to yield until the car velocities are finally equalized. The amount of yield and the magnitude of the forces are inversely proportional and depend entirely upon the sturdiness of, or the resistance offered by, the parts. The sturdier the parts, the more force will be required to deform them and the less will be the amount of penetration and yield and incidentally the lower will be the unit stress. On the other hand, the lighter the parts the greater will be the yield and the less the force, but the higher will be the unit stresses. Accordingly, although providing a temporary cushioning for the over-solid blow, the lighter parts will shortly be deformed or broken. Any weak link, be it coupler shank, draft gear housing, draft lugs or center sills, will reduce the force

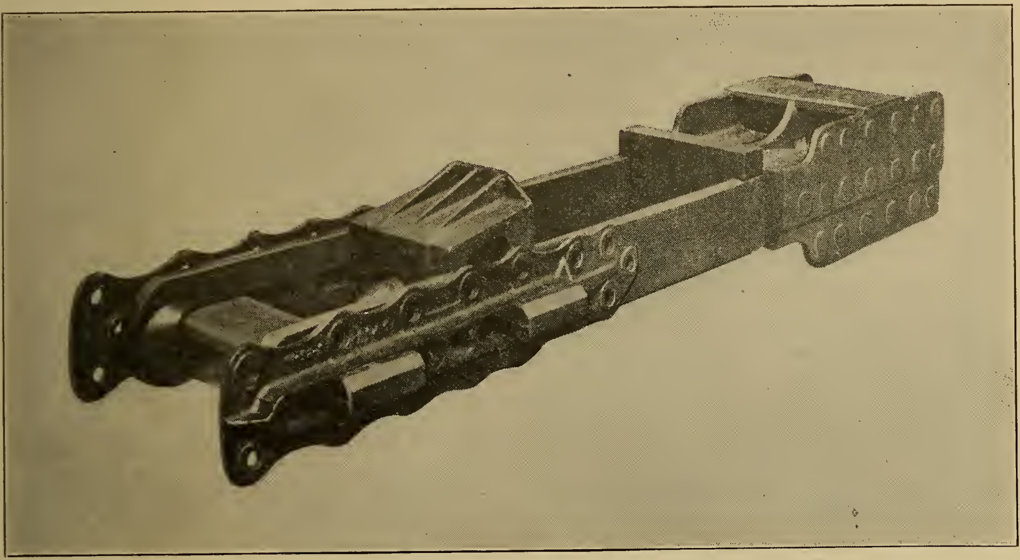


FIG. 48—FARLOW TWO-KEY DRAFT GEAR ATTACHMENTS USED ON TEST CARS

peak but only at the expense of its life.

The velocity of car A is thus being gradually decreased and that of car B increased until the velocities are equal, at which instant all parts have reached the maximum of compression. If all of the parts were perfectly inelastic, if in other words, there should be no tendency for the gears to release themselves or for the car structures to give back the energy of their elastic yield, it is evident that there would then be no force of recoil to separate the cars, and both cars would accordingly move off together at this equal velocity, each car having one-half of the original velocity of car A, neglecting a slight loss due to internal resistances. With equal rolling and grade resistance, the cars would continue together until both finally came to rest without separation. Except for the slight loss due to rolling resistance, the work done in compressing the gears and car structure is thus always equal to one-half of the original kinetic energy in car A, and it should be especially noted that this is the same whether there be no recoil of the gears and car parts, or full recoil.

The force exerted between the cars in compressing the gears and car structure is entirely independent of the question of absorption. Up to the point of maximum compression the matter of absorption of energy has not entered into or influenced the problem. It is entirely a question of force and yield and it should be remembered that frictional resistance, while truly absorbing energy (foot-pounds) does not in any manner whatsoever reduce or "absorb" force. The force required to close a friction draft gear, and consequently the force going through the gear to the sills, may be greater or less than a spring draft gear of equal capacity, depending solely upon its compression curve, and not in the slightest degree upon its percentage of absorption. The cushioning value of a gear therefore is not measured by absence of recoil, or energy absorption, but solely by its action during the closing period. Whether or not a gear has extensive recoil has nothing to do with its action on compression, or with the force delivered by the gear to the car during its compression.

In practice, the cars having reached a point in the draft gear cycle where their velocities are equal, and the compression period of the cycle completed, the release of the gears begins. All gears have more or less recoil and it is this force, together with the rebound of the car structure, that tends to part the cars and to cause one car to travel faster and farther than the other. It should be especially noted that the force of recoil has the same effect between the cars as the force of compression; namely, to reduce the velocity of car A and to increase the velocity of car B. During the period of gear compression the force between the cars, or the force tending to accelerate car B, results from the higher velocity of car A, or its direct tendency to push car B ahead. During the period of draft gear release, the force tending to further accelerate car B or to urge it forward, results from the recoil or return of stored energy in the two draft gears and both car structures.

The recoil of the gears and car parts thus giving to car B a greater velocity and to car A a lesser velocity, car B will begin to travel faster than car A, and the gears, following the resulting parting of the cars, will continue to release until final separation of the cars. It is evident that the greater the force of recoil, or release, the greater the pressure between the cars during release, and consequently the higher will be the velocity attained by car B and the greater the retardation of car A. A gear with 100 per cent recoil would actually bring car A to rest by the time the cars separate, while car B would be pushed ahead at a velocity practically equal to the original impact velocity of car A. On the other hand, a gear with no recoil, or 100 per cent absorption, would, as heretofore set forth, cause both cars to move off together, each at one-half the initial velocity of car A. Gear absorption is thus inversely proportional to the pressure exerted between the cars during the period of release, the effect of high absorption being to hold the two cars at nearly equal velocities after impact. This means, in effect, that with high absorption of energy, car B will not be propelled at so high a velocity and consequently will strike the next succeeding car at a reduced velocity, while with no absorption, car B will strike the next car at almost the same velocity as the original of car A. Absorption therefore is not primarily a means of reducing the force of impact between the first two cars, or of protecting these cars, but is a means of reducing the moments of the successive impacts between successive cars in a train.

The following may be accepted as general principles of draft gear action in impact:

1. That draft gears are compressed only because of differences of velocity between adjacent cars.

2. That the resistance offered by the gear during compression tends to overcome the difference of velocity of the cars and tends to bring both cars to the same velocity.

3. That gears continue to close, and at over-solid velocities the car structures continued to compress, until the car velocities are equalized.

4. That this action of a gear is independent of its ability to absorb energy, or in other words, is the same whether the resistance be obtained from friction or solely from spring action.

5. That the cushioning offered by the compression of a draft gear is not dependent upon its percentage of absorption.

6. That absorption does not in any manner reduce the force going through the draft gear to the car sills while the gear is being compressed, and does not lower the force exerted betwen the first two cars colliding. That it does act to lower the velocity with which the second car strikes the third car and consequently reduces the force between successive cars.

7. That the amount of "work-absorbed" by a gear, or the percentage of absorption does not regulate or reduce the force of first collision, but is important as determining whether shocks will run practically undiminished throughout the train or whether there will be successive reductions in their moments from car to car.

8. That the first measure of a draft gear is the amount of energy required to close the gear, this being the sole factor from which to determine for what switching speeds a gear is suitable. This is expressed as "work-done" and has no relation whatsoever to "work-absorbed."

9. That the next requirement is that a gear, either spring or friction, shall compress with such a rate of increase of resistance as will cause the lowest practical ultimate force and the least practical vibration of the car structure.

10. That the next measure is with respect to the action of the gear on release or the amount of the recoil, whether the energy of compression is returned, to go on to the next car, or whether it is partially absorbed as by friction. This property is expressed by the term "work-absorbed."

Records in Car-Impact Tests

In the car-impact tests the following records were taken:

Impact velocity of car A.
Travel of cars along track.
Draft gear travel and action.
Seismograph readings.
Graphs of car action.

From these prime records a complete study of the action of both the gears and cars can be made, the details of which will appear as the manner of making and interpreting the several records is discussed.

Impact Velocity

The first information needed in such tests is an accurate knowledge of the velocity of car A at the very instant of impact. It is not enough to simply release the car at a fixed point along the incline, for the same station will not always develop the same velocity. Nor is it satisfactory to establish five-foot or ten-foot stations near the point of impact and calculate the impact velocity by means of the average velocity between these stations, as very marked changes in velocity may occur over such periods. The kinetic energy of car A is determined from the impact velocity, and as it varies with the square of the velocity, and furthermore as all of the results of the tests are based upon this record, accuracy here is of utmost importance. In these tests car A was caused to draw a velocity line upon a revolving drum, so that the exact velocity at the very instant of impact is obtained within a possible error of less than 1 per cent. A more detailed description of the recording device will be given later under the heading of "Car-Movement Curves."

Travel of Cars Along Track

An interesting record, easily obtainable, is the distance each of the two cars travels along the track after the impact. Care must be taken in interpreting these figures, however, as a slight change in grade will offset a considerable track movement of the cars. Thus, if but one of the eight wheels of a car mounts an obstacle on the track $\frac{1}{16}$ in. in height, it is equivalent to six inches additional movement of the entire car along level track. An interesting point in connection with this record is that for equal impact velocities, the higher the recoil of the gear used, the greater the distance car B will travel. In general, the recoil of gears will be proportional to the distance between the cars after coming to rest; that is, the greater the recoil the farther apart will be the cars when they come to rest.

Draft Gear Travel and Action

Knowing the impact velocity, the next point of interest is the amount of and the nature of the travel or yield of the draft gears. The test cars are equipped with friction plunger gages to show the amount of coupler travel. This corresponds reasonably closely with the actual draft gear travel, but is not sufficiently accurate for analytical investigations. In order to obtain a more direct knowledge of the movement and action of the gears, car B is provided with a small revolving drum upon which is drawn a curve which shows not only the amount of draft gear movement for that car but the character of the movement; that is, whether the gear compresses and releases regularly or irregularly. A photographic view of this instrument is shown in Fig. 49, a case or bracket being secured to the side sill of car B in which is a small motor-driven drum which extends transversely of the car. A pencil is caused to move lengthwise of the drum in harmony with the movement of the front draft gear follower. For this purpose a piano wire extends from this draft gear follower to the pencil arm, the connections being arranged in such manner that tipping of the follower block will not produce false movement of the pencil. Relative movement between the side sill and the center sill is also compensated for. A 40 lb. coil spring and a 40 lb. friction drag prevent overtravel of the pencil, the spring alone serving to return the pencil during the release of the gear. The drum is covered with paper, and as the gear is compressed or released the pencil is moved correspondingly along the axis of the revolving drum, thus producing a time-closure diagram for the gear in car B.

Tests were in each instance made with gears in both cars and again with a gear in car B only, car A in the latter case being fitted up with a solid steel block instead of a draft gear. The action of the individual gear can be best studied under these latter conditions because it is definitely known that any irregularities recorded are due to the particular gear. In the former case the record does not determine which of the two opposing gears is responsible for the irregularities. Such irregular action is almost invariably recorded when both cars are equipped with gears. The specimen card reproduced in Fig. 50 was made from the gear in car B when each car was equipped with a friction draft gear. This card shows the typical action of friction gears in the double-gear tests, or when both cars are equipped with gears.

By means of these cards it is found that while some draft gears act smoothly and regularly, others, particularly those from friction gears of high capacity, are often closed by a succession of alternating movements or jerks. This will be shown as the individual cards are reproduced. The lower capacity gears naturally show smoother gear action than those of higher capacity. In fact, without exception, the high capacity gears show jerks and irregularities in the compression line of the time-closure diagrams. This, in the single gear tests, is believed to be due largely to the pulsations or periodic vibrations between the two cars resulting from the high forces incident to a high capacity gear with short travel. The cards show that with a gear in each car, the two gears do not work in harmony; that frequently on compression, and almost invariably on the release, one gear will work for a while and then the other one will operate. From this it is concluded that twin arrangement of friction draft gears is not permissible.

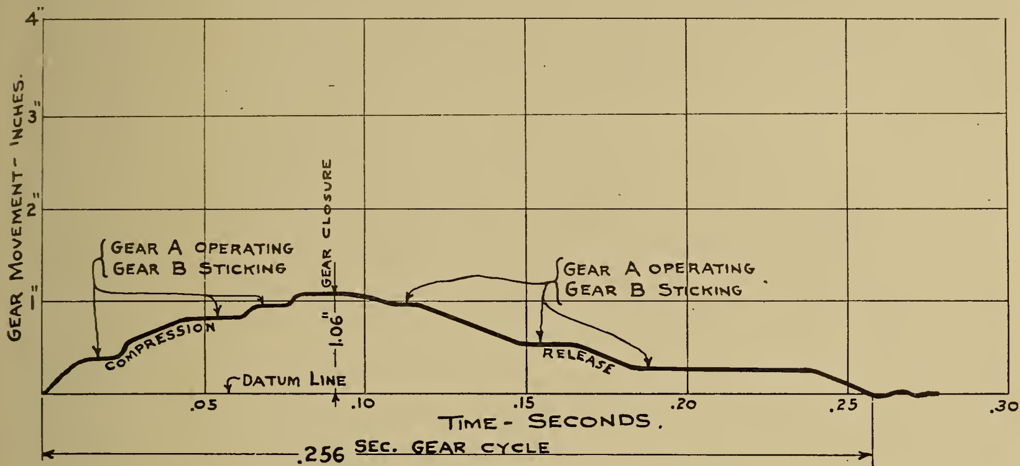


Fig. 50—Specimen Time-Closure Curve Produced on Small Drum of Car B

Seismograph Readings

Each of the test cars is equipped with a pendulum device, secured to the side of the car, and so arranged that the retarda-

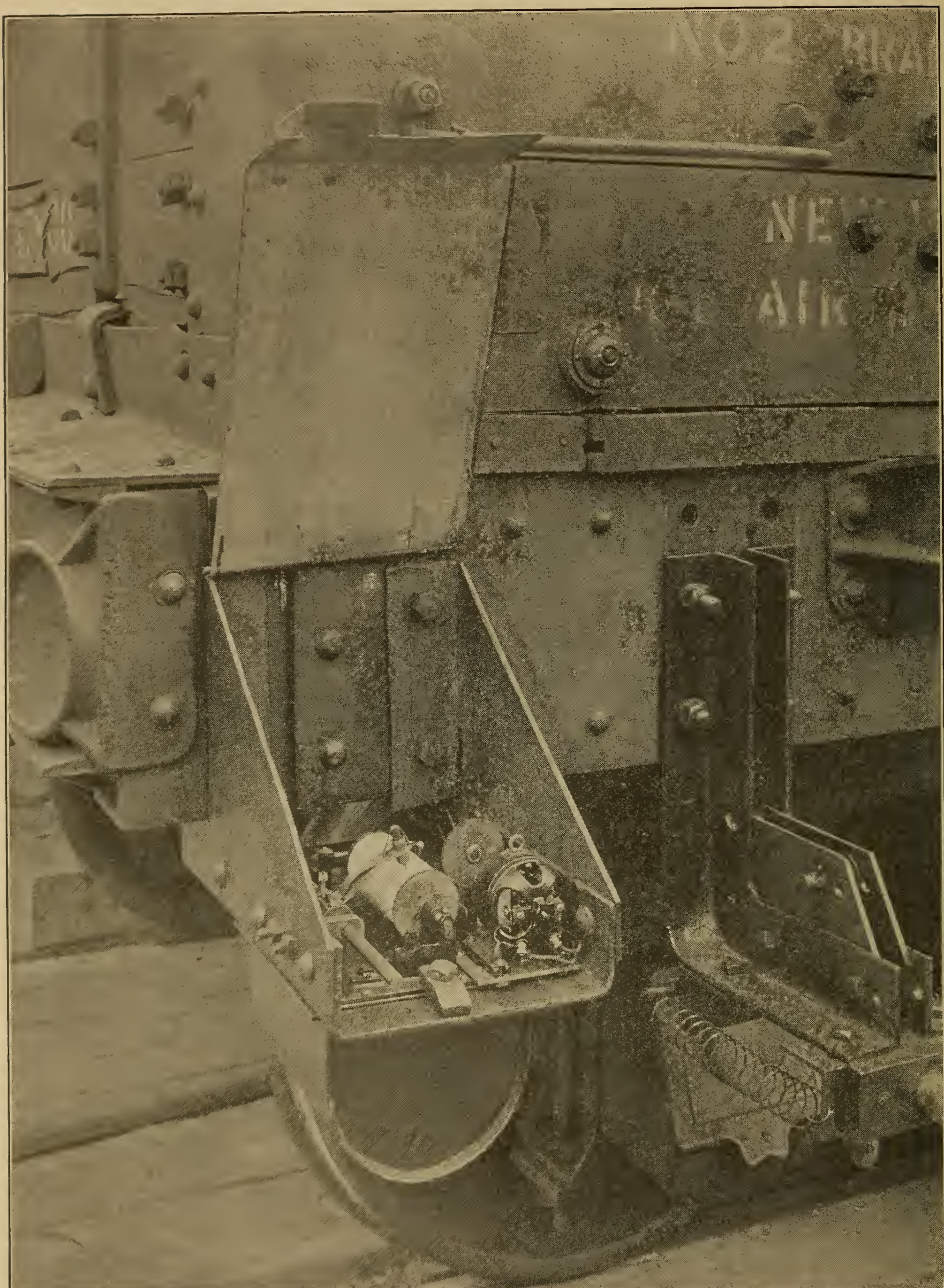


FIG. 49—INSTRUMENT ON CAR B FOR RECORDING DRAFT GEAR ACTION

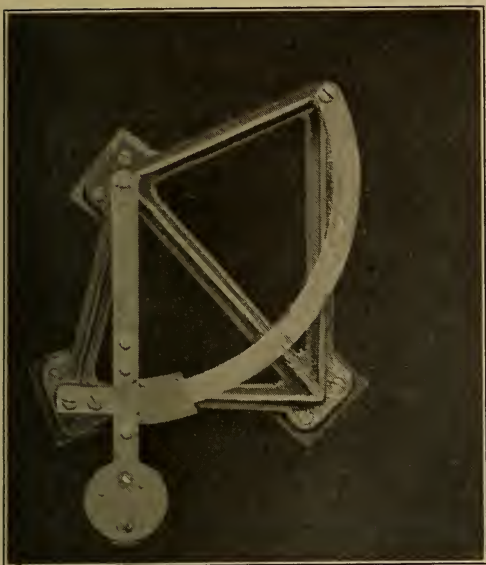


Fig. 51—Seismograph of Car A

tion or acceleration of the cars will cause the pendulums to swing upward by virtue of the inertia of their own masses. Graduated quadrants are arranged as guides for the pendulum weights, and a light friction runner is carried with the weight and is left standing upon the guide at the highest point reached by the pendulum. The graduations are proportional to the vertical lift of the pendulum. Thus when the registration is 4.0 the pendulum has reached a vertical displacement twice as great as when the registration is 2.0. A photographic reproduction of the seismograph of car A is shown in Fig. 51. The seismograph records are usually attractive to the observer but are not of great importance in the study of gears. This is due primarily to the fact that the sides of the car have some movement with respect to the center of the entire mass. The quick vibrations of the side of the car appear also to influence the seismograph readings. These instruments are frequently spoken of as "shift gages."

Graphs of Car Action

As the final study of draft gear action must lie in a study of the results of the use of the gear upon the car and its lading, arrangements were made to obtain a complete and accurate record of the performance of both cars during the brief period of the draft cycle. A recording apparatus, arranged to draw simultaneous time-displacement curves of both cars, was designed and installed and a system of certain reference lines worked out whereby these curves could be later so super-imposed that at any instant during the draft gear cycle an exact knowledge of the performance of both cars could be had. These curves are commonly referred to as "car-movement curves." Photographic reproductions of the instruments for producing these curves are shown in Figs. 52 and 53. The apparatus has two drums mounted upon a common shaft and is placed on a stand alongside of the track. Each drum is 20.05 in. in circumference over the paper, and 30 in. long. The axis of the shaft is parallel to the track and the drums are so mounted upon the shaft that one drum is alongside of the striking end of car A and the other alongside of the struck end of car B. Each drum has a pencil carriage that is moved lengthwise of the drum by the movement of the car, each car having a pencil-propelling plunger attached to its side sill (see Figs. 49-52-53). Suitable angle iron guides are arranged upon the instrument stand to cause the plungers to move into or out of engagement with the pencil carriages at the proper times.

Making a Test Run

In making a test the first operation is properly to apply the test gears to the cars. Care is taken to so adjust the length of the draft gear pockets that the gears will be held to their proper lengths. It is some-

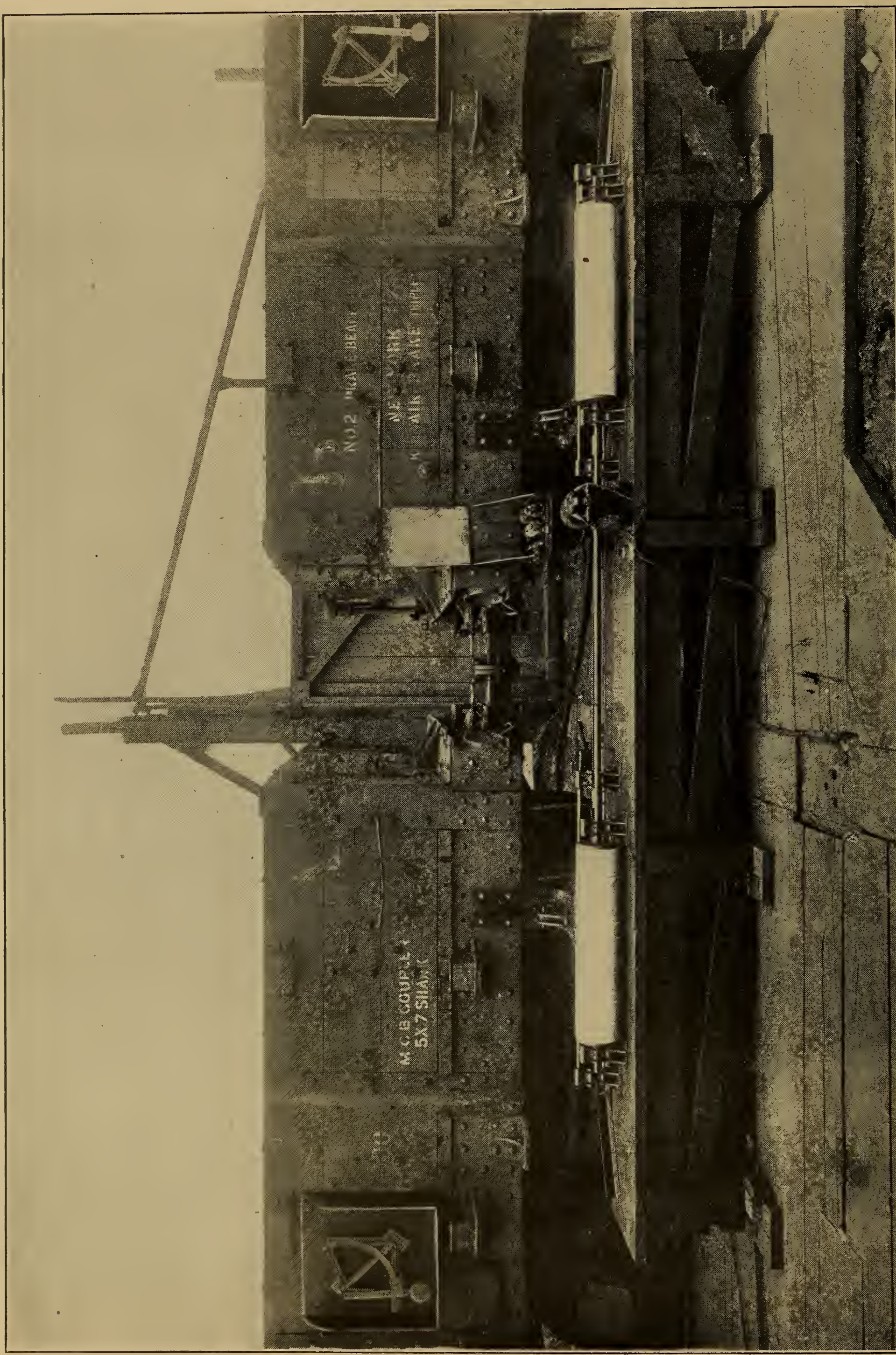


Fig. 52—Instrument for Recording Car Action

Fig. 53—Another View of Instrument for Recording Car Action

times necessary to apply liners behind the gear in order to accomplish this. After applying the gears to the cars, it is the practice to make ten preliminary runs at just slightly below the closing speed, in order to condition the gears before making the regular runs. Car B is then spotted, always at the same definite station along the track. Car A is also spotted, the buffing faces of the couplers being just in contact and all loose slack being eliminated or compensated for. With the cars so spotted and with the A and B pencils in positions on their respective drums corresponding with the positions of cars A and B respectively, the drums are rotated a few times, thereby drawing the datum lines, A-A for car A, and B-B for car B (see Figs. 54 and 55). At the same time the small drum is rotated a few times so that its pencil draws the datum lines for this record (see Fig. 50). It will thus be seen that all of the datum lines are drawn with the cars and gears in position as at the first instant of impact; or, in other words, at the beginning of a true gear compression. Accordingly, in comparing the cards, it is definitely known that all car movements and gear action can be compared from these common datum lines.

Without rotating the drums, each of the pencils is given a slight longitudinal movement, in order to draw the reference lines D-D and E-E on the A and B cards respectively (see Figs. 54 and 55). The pencil for car B is left standing exactly upon the datum lines B-B, or in other words, in position so that the first movement of car B will move this pencil along drum B. Car A is then drawn away from car B and the pencil for car A is drawn along the axis of drum A in order that the approaching car A may propel this pencil for some distance before the pencil reaches the datum line A-A, or the position where the two cars first meet. By this means the exact impact velocity of car A is determined, the speed of rotation of the drums being known. In order to obtain as nearly as possible the desired velocity, the trip is set at a prescribed point along the inclined portion of the track. The velocity developed from any station varies from time to time, hence, the exact velocity of impact must be determined for each run from the line drawn by car A below the datum line A-A. As car A approaches car B, all of the drums are set in motion, care being taken to start the instruments a sufficient time in advance to get the drums up to constant speed before the pencils are moved.

In the record from drum A reproduced in Fig. 54 the pencil was stationed at a position represented by the line F-F, until the approaching car picked up the pencil and began to propel it along the axis of the drum. The angular line drawn by the pencil between the lines F-F and A-A denotes the velocity of car A, the paper speed being known. This line being straight shows that the drums rotated at a constant speed. From the preliminary set-up of the cars and instruments it is known that when pencil A reaches the position of the line A-A on this drum, the cars have just met, for as previously explained, the datum line A-A was established prior to the tests for indicating the position of this pencil along the drum at the first instant of impact. As car A propels the pencil beyond the line A-A it is known that the draft gear cycle has begun, and from the convexity of the curve it can be seen that the velocity of car A is being reduced, due to the resistance of the gears.

At the instant when the cars first met, or when this car-movement curve crosses the datum line A-A, it will be seen that the pencil was $3\frac{1}{2}$ in. from the reference line D-D. It is known that at this instant the B pencil was exactly the same distance

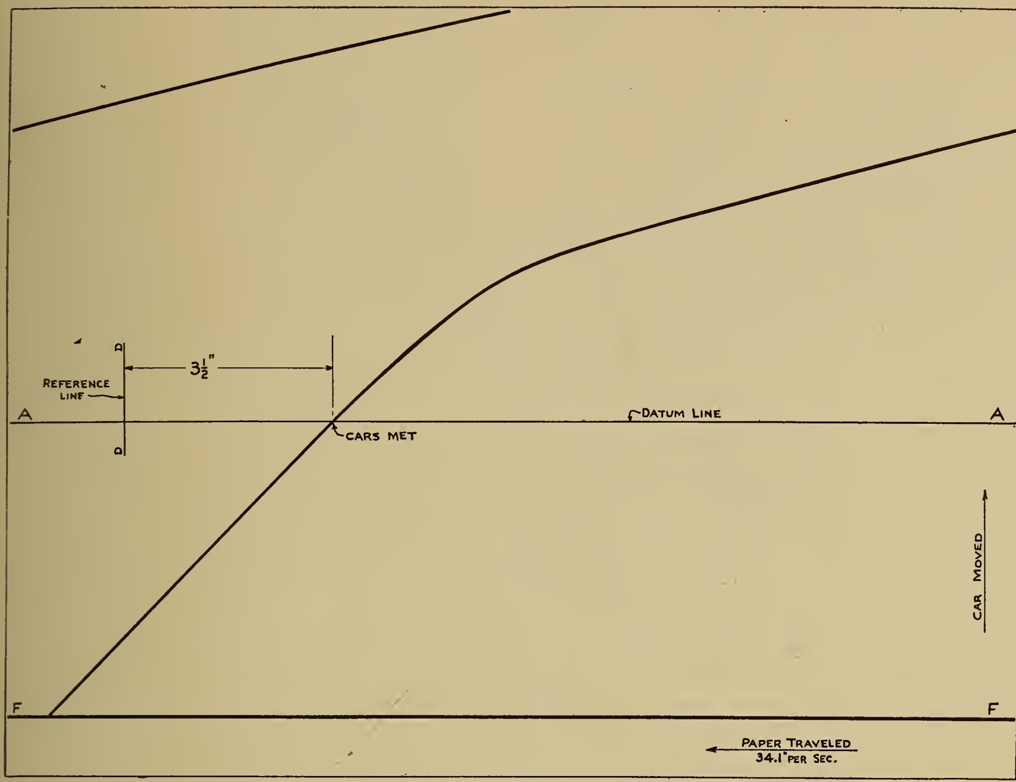


FIG. 54—SPECIMEN CAR-MOVEMENT CARD FROM DRUM A

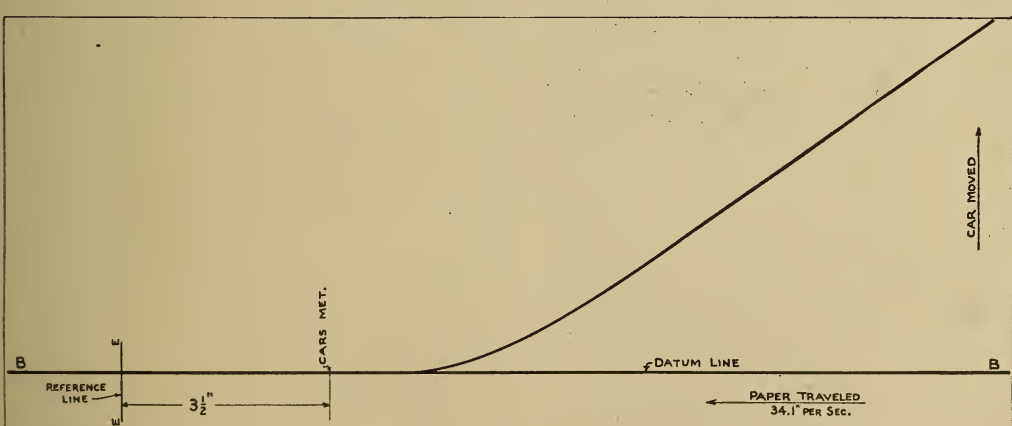


FIG. 55—SPECIMEN CAR-MOVEMENT CARD FROM DRUM B

from its reference line E-E, for these reference lines were previously drawn to denote the relative positions of the two pencils or the datum lines. It will be noted from card B, Fig. 55, that a small interval of time elapsed before car B began to move out of its spotted position, the gears in the meanwhile compressing. When it began to move, its velocity gradually increased as shown by the concavity of the curve.

By means of the datum and reference lines on these two cards a system of superimposition of the two curves has been developed and in Fig. 56 these curves have been so superimposed. This is done by matching up both the datum and reference lines and tracing one curve upon the other. The exact meeting point of the cars is thus established for both curves and both are also synchronized as to time. Consequently both the velocity of the cars and their relative positions can be determined for any instant. And at any instant, also, the distance either car has moved from its spotted positions is known. It will be seen that car A, during the first portion of the draft gear cycle continued to travel at a higher velocity than car B. As car A thus encroaches upon car B the draft gears are compressed, the distance betwen the two super-imposed curves representing draft gear compression, together with the slight yield of the car bodies. Car A continued to run down upon car B, its velocity gradually decreasing and the velocity of car B gradually increasing due to the draft gear forces exerted between the cars, until both cars were of equal velocity. This point corresponds with the point of maximum draft gear compression and can be readily determined by finding the maximum ordinate between the two curves. From this point on, the velocity of car B becomes greater than that of car A due to the forces of draft gear recoil between the cars. Consquently, car B moves away from car A, allowing the draft gears to continue their release. At the point where the two curves cross there is no relative displacement of the two cars, or in other words, each car has travelled the same distance from its datum line, and it is therefore definitely known that at this instant the cars parted and that the draft gear cycle was completed.

From the superimposed curves, Fig. 56, it is possible to obtain a wide range of information concerning car action and draft gear action. The dotted line erected upon the datum line, for example, shows the movement of the two draft gears during compression and release. This curve is obtained by the simple process of stepping off the ordinates between the two curves upon the datum line as a base. The point where this draft gear curve reaches its maximum height is the point of maximum draft gear compression, and a vertical line has been drawn to indicate this point on the curves. From this it is then seen that the period of draft gear compression was 0.090 seconds and the period of release 0.166 seconds, the entire draft gear cycle, or the total length of time the cars were in contact being 0.256 seconds.

At the instant of maximum draft gear compression, car A had moved 2.52 in. along the track from the point of impact, while car B had moved but 0.42 in., car A thus having encroached upon car B for 2.10 in., causing a corresponding amount of gear closure. At this instant, car A ceased encroaching upon car B, as shown by the falling off in gear closure. At the instant of maximum gear closure the velocities of the cars were equal, and the lines established tangential to the car-movement curves at this point denote the common velocity at this instant. These tangential lines also indicate the paths of the car-movement curves had there been no force of recoil, or if the draft gears had stuck. Angles have been drawn in to indicate the

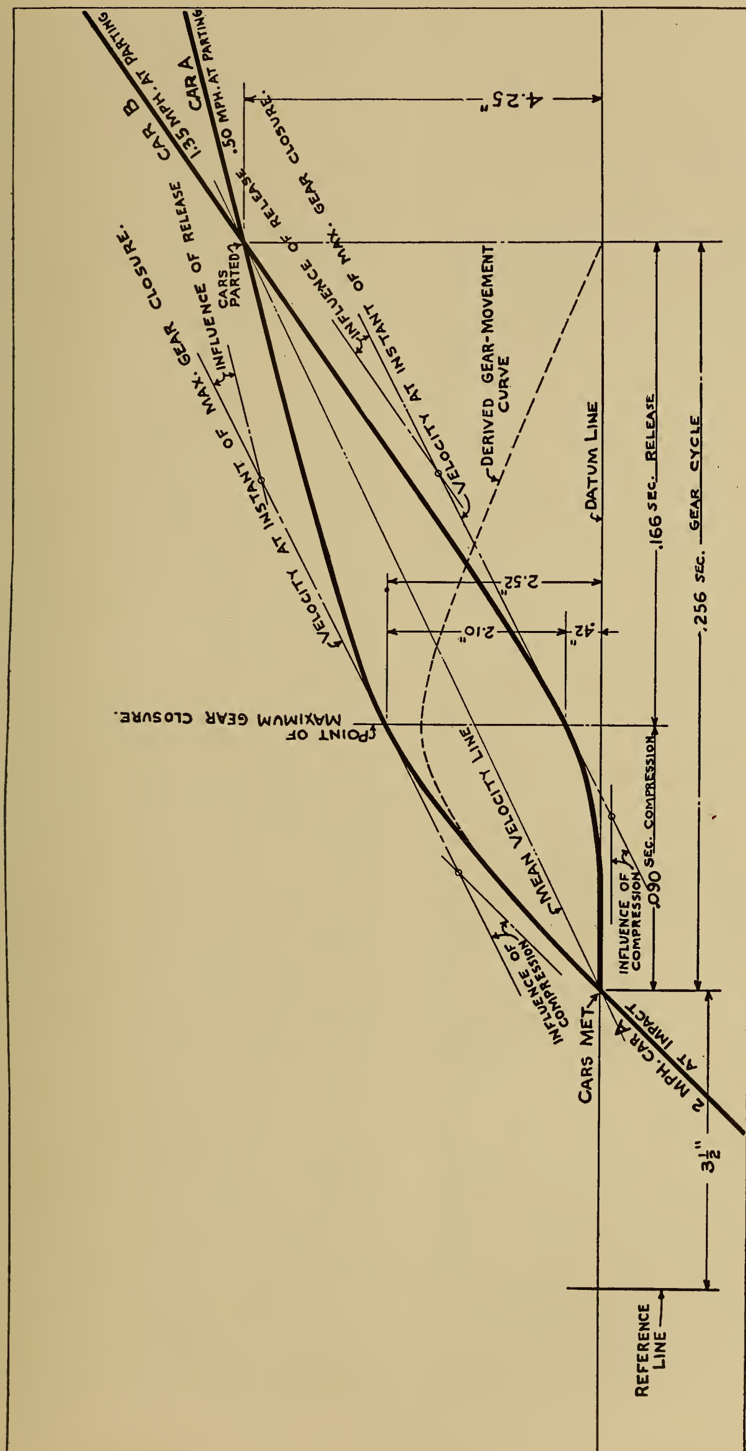


Fig. 56—Specimen Car-Movement Cards from Drums A and B Superimposed

influence of gear compression and gear release, and the dimension of 4.25 in. shows the track movement of the cars during the entire draft gear cycle.

The card of Fig. 50 was drawn by the action of the draft gear in car B during this same run. It will be seen that this gear closed 1.06 in., thus showing that the gear in car A closed 1.04 in. While the line in Fig. 56, representing the sum of the actions of the two gears is smooth and regular, yet the individual gears did not operate so regularly. The compression and release was attained by first one gear operating and then the other. This is to be expected from friction gears and indicates variations in the effective co-efficient of friction. No special demerit is attached to this action of a friction gear, as either one gear or the other is operating at all times.

It is important to have an exact record of the paper speed and especially important that there shall be no variations in speed during a run. To this latter end, the electric current for operating drums A and B was supplied by a set of twenty-four Edison batteries which were frequently recharged, and as no other current was drawn from these cells the speed of the drums was kept practically constant. The speed, however, was checked at frequent intervals to guard against errors in this respect. With a definite knowledge in each instance of the paper speed, it is possible to establish the time ordinates, and from this scale is deduced the time interval required to close the draft gears and the time interval for the release of the gears, the sum of these two intervals being designated throughout this report as the "draft gear cycle." The paper speed ranged around 34 in. per second throughout the tests, but the exact speed was known for each individual test. The time scale is, of course, necessary for determining car velocities, and from the superimposed curves it is a simple matter to determine the exact impact velocity of car A and also the exact velocities of both cars at the instant of parting. It is also possible to determine by tangents the change in velocity of each car during any period of draft gear compression or release. It is further possible to plot curves showing the instantaneous velocity of both cars, and from these it is a matter of simple calculation to produce curves showing the instantaneous energies in the two cars. From the rate of change of velocity, the mean or average forces working between the two cars throughout the period of impact may be computed and a continuous time-force curve plotted.

By stepping off and plotting the vertical distances between the superimposed car-movement curves as heretofore explained, a time-closure curve of draft gear action can be produced. This curve will show the complete draft gear action, both compression and release, plotted against time, and in cases where a gear is used in car B only, the curve will practically coincide with the curve drawn by the small drum on car B. This erected time-closure curve, however, includes not only the yield of the draft gears but has added to this the yield of the two car bodies. In this connection, it should be remembered that any yield of the car body constitutes additional draft gear action. By combining the time-closure curve and the time-force curve, the time element being eliminated, there can be produced a force-closure curve which corresponds with the ordinary static curve of draft gear testing, although produced from actual operation of the gear during impact.

A large number of runs were made for each type of gear but the limitations of space and the labor of working them up in complete form do not permit the reproduction of all of them in the report. The uniform practice has been followed of working up and reproducing for each type of gear the following runs:

1. A run, made at or near the closing point, with a calibrated test gear in car B only, car A being equipped with a solid steel block instead of a draft gear.

2. A run made at approximately one mile per hour, each car being equipped with a calibrated test gear.

3. A run made at or near the closing point, each car being equipped with a calibrated test gear.

The first of these is worked up primarily that the action of a single calibrated gear in the car-impact tests may be compared with the action of the same gear in all of the laboratory tests, the possible influence of a second gear being removed. The second is worked up that a complete knowledge may be had of the action of each type of gear at low impact speeds. These low speed runs are especially useful in a study of train starting. The third is worked up as showing the best that may be expected from each type of gear at the maximum impact speed it is capable of cushioning, and gives the true comparison of the gears from the standpoint of yard service.

The same gears of a type were used throughout the test, the general practice being to first make tests with both cars equipped, and then after replacing the gear in car A with the solid block, to make the single gear tests.

Study of Curves

A variety of interesting curves may be derived from the car-movement curves, but the essential features of the functioning of the gears will be shown in the following, which are reproduced for each of the three runs for each type of gear.

Master Curves

Car-Movement Curves—Superimposed.

Derived Curves

Velocity Curves.
Energy Curves.
Time-Force Curves.
Time-Closure Curves.
Force-Closure Curves.

Throughout the report the curves have been reproduced to the same scale, so that the action of the different gears may be directly compared. The curves of the Westinghouse D-3 gear will be used for purposes of general description.

Car-Movement Curves—Superimposed

In tracing and reproducing the car-movement curves for publication it is not possible to bring out all of the small variations and irregularities. In many instances these curves, although appearing smooth and regular to the eye, contain numerous perceptible variations in the originals. All of the derived curves were produced directly from the originals and hence in the further studies of the gears the presence of any irregularities will be seen. The arms for producing the car-movement curves were attached to the side sills of the cars and although these test cars are equipped with two complete sets of diagonal braces, yet in severe buffing there is some movement of the side sill relative to the center sill and always more or less vibration. The irregularities in the car-movement curves are therefore due in a large measure to the vibration or to the relative movement of the side sills. The effect of these vibrations upon the car-movement curves, the full significance of which is brought out forcibly in the derived velocity curves, is probably the best comparative measure of the smoothness of action of the draft gears that can be obtained, for the smoother the action of the gear the more gradual and regular will be the transfer of energy from the striking car to the standing car and the less will be the vibrations of the car structure.

The superimposed car-movement curves, Fig. 80a, were made when car A was equipped with a solid steel buffer and car B with test gear No. 2. These curves represent the closing run for a single Westinghouse type D-3 gear, the exact speed of impact being 2.68 M. P. H. At parting, car A had a speed of 0.74 M.P.H. and car B, 1.84 M.P.H. The instant of maximum gear compression, or in other words, the instant where the cars were of equal velocity, occurred 0.084 seconds after the first instant of impact. It required 0.166 seconds for the draft gears to release, or for the cars to part. The duration of the entire draft gear cycle was 0.25 seconds. The combined draft gear closure and car body yield, which includes the movement of the side sills of the cars, amounted to 2.65 in., this being the maximum ordinate between the two curves. At this instant car B had moved but 0.62 in. and car A, 3.24 in. along the track. Incidentally, throughout these tests, it has been found that the draft gears are closed and the maximum force developed between the cars before car B moves any material distance. Each of the cars moved 5.07 in. along the track while in contact, or during the complete draft gear cycle.

Velocity Curves

Fig. 80d shows the derived velocity curves for the single gear run of the Westinghouse D-3 gear at the closing speed. The irregular dotted line shows the exact first derivatives of the car-movement curves, the first derivative being instantaneous velocities. Any slight irregularity in the car-movement curve becomes very apparent in this differentiation. The curves for the Westinghouse D-3 gears are unusually smooth for its capacity.

The impact velocity of car A in this run was 3.93 feet per second (2.68 M.P.H.), the velocity of car B at this instant being zero. As the gears compressed, the velocity of car A decreased and the velocity of car B increased until at the instant of maximum gear compression both cars were of the same velocity, namely, 1.92 feet per second. The result of the closing of this gear, therefore, was to reduce the velocity of car A from 3.93 feet per second to 1.92 feet per second. The remainder of the change in velocity of the two cars is due to the recoil of the gear, the effect of the recoil being to increase the velocity of car B to 2.69 feet per second and to still further reduce the velocity of car A to 1.08 feet per second at parting.

The velocities represented by the irregular dotted lines are true representations of the actual velocities of the side sills of the cars with respect to a stationary point along the track. It is not to be understood, however, that the entire masses of the cars followed these velocity changes. Even though well constructed, these cars, like all others, are elastic and subject to more or less yield and vibration of parts. The irregularities in the velocity curves are accordingly due largely to the local surging and vibrations of the side sills. The frequency and amplitude of the irregularities are a direct comparison of the results of the use of the various gears upon the cars. Thus it will be seen that with a spring draft gear, and with some of the lower capacity friction gears, the transfer of motion from one car to the other is effected with practically no disturbance of the car structure, the velocity curves being relatively smooth. On the other hand, with the higher capacity gears, considerable vibrations are set up. It is not to be expected that a gear functioning up to, say, 4 miles per hour, will give as smooth and regular a velocity curve at its closing speed as one functioning only to 2 miles per hour. The point of real interest is to compare the relative smoothness of these curves from gears of the same capacities and at approximately the same impact speed.

It is not possible to obtain a true velocity record of an ordinary car from any one of its parts. Local vibrations and surges occur in every particle of the car, even in the center sills. It would be possible to record the change in velocity of a car if it were constructed of a solid block, as of cast iron or cast steel, for in such a structure the vibrations would be so small as to be negligible. In such a test it should be possible to determine draft gear resistance to a nicety. But because of the very fact that with a cast iron car vibrations and elastic yield are practically impossible, such an outfit is unfit for the test. Compressing a gear between two inelastic cars will not permit the development of the very things, viz., irregularity of gear action, that are being searched for. For if the structure, the inertia of which resists the compression of the gear, is incapable of yielding and vibrating, then the tendency of the gear to produce and to follow such vibrations in test action will be prevented, and any gear, unless of the most erratic nature, will produce a smooth closure curve. This fact makes it imperative that draft gears should be tested upon actual cars so that if a gear has a tendency to pinch and bind on compression, it will be developed and discovered.

It should be remembered that these car-body vibrations are a product of the individual car and that each car will produce its own variations in velocity curves, due to the peculiarities of the particular car construction. Further, these vibrations in the velocity curves should not be interpreted as meaning that the side sills of the cars vibrated through such distance. They represent instantaneous changes in velocity and the actual movement of the side sills that occurred were very slight; in many instances barely more than a tremble and seldom more than 1/8 in. Mean velocity curves, shown in full lines, have been established from the general trend of the original car-movement curves, and these represent, as closely as it is practical to obtain, the true mean velocity of the entire mass of the car. This mean velocity curve is used throughout the remainder of the cards for the determination of energy and force.

An interesting point in connection with the vibration of the cars was experienced when first developing the instruments at the Symington test plant. The first car-movement curves attempted were exceedingly irregular and showed a continuous series of waves, even when using spring draft gears at low impact speeds. These waves were found to be due to the longitudinal vibrations of the car body and truck bolsters upon the truck springs. Liners were applied between the truck bolsters and the bolster guides of the truck side frames to prevent this vibratory movement upon the springs, but at the same time allowing vertical movement. The next succeeding runs were smooth. It is recognized that in producing this artificial rigidity between bolsters and side frames, the action of all the gears may have been very slightly smoothed out. For the surging of the body upon the truck springs might under some circumstances be reflected in the action of the gear.

The production of the velocity curves from the car-movement curves, and especially the showing of all the variations, was made possible by the use of a mechanical differentiating machine devised and built by Mr. Armin Elmendorf, formerly professor at the University of Wisconsin, and at present consulting engineer, with offices at 819 Chamber of Commerce Building, Chicago. Mr. Elmendorf has been prominent in the art of mechanical differentiation, two of his papers on the subject appearing in the Journal of the Franklin Institute for January and February, 1918. The differentiating machine is based on the principle of similar triangles, a large triangle always being developed similar to a smaller differential triangle. The angle of the latter being varied according to the tangent of the car-movement curve causes a similar change in the larger, or plotting triangle, and the instantaneous velocity is thus plotted continuously and directly from the original car-movement curves, and with a much greater degree of accuracy than is possible by laying off tangents. This same instrument was also used to produce the time-force curves directly from the velocity curves. The instrument is invaluable for determining mechanically the first derivative of any curve. A photographic reproduction of this instrument is shown in Fig. 57.

Energy Curves

The energy curves shown in Fig. 80g have been produced by simple calculation from the preceding velocity curves. These energy values include not only the kinetic energy represented by the direct movement of the car as a whole, but also the energy of rotation of the wheels and axles, which in these cars amounts to an addition of 2.83 per cent to the ordinarily considered energy of translation. The total kinetic energy in one of these cars (143,000 lb. gross weight) including the above rotative energy, may be conveniently determined by the formula 4918 V^2, V being the car velocity in miles per hour.

In this particular run (Westinghouse D-3 single gear at closing speed) the kinetic energy of car A was reduced from 35,308 ft. lb. to 8,427 ft. lb. by the compression of the gear, while at the same time the kinetic energy of car B was increased from zero to 8,427 ft. lb. The sum of the kinetic energies of the cars at this instant, (the instant of maximum draft gear compression) amounted to 16,854 ft. lb., so that the work done in compressing the draft gear and the car structure, and in overcoming rolling and grade resistance, amounted to 18,454 ft. lb. This quantity corresponds with the expression "work done" as applied to drop testing of draft gears. The dotted line beneath the line of zero energy represents the instantaneous value of work done at any instant during draft gear compression up to the instant of maximum draft gear closure.

The energy curves during the period of draft gear release show the changes in kinetic energy produced in the cars by the recoil of the draft gear. In this particular run the recoil increased the kinetic energy of car A to 16,542 ft. lb. and reduced that of car B to 2,666 ft. lb., so that at the instant of parting the kinetic energy represented by the movement of the two cars amounted to 19,208 ft. lb. The original kinetic energy of car A being 35,308 ft. lb., there was thus a total absorption in this run of 16,100 ft. lb., this quantity corresponding with the expression "work absorbed" as applied to drop testing of gears.

The greatest possible absorption that could have taken place is always represented by the maximum ordinate to the dotted curve beneath the line of zero energy, and this point always coincides with the instant of maximum draft gear compression. During the period of compression the sum of the kinetic energies of the two cars is decreasing, a portion of it being stored or absorbed by the draft gear. During the period of release the draft gear returns more or less of this stored energy to the cars so that the sum of the kinetic energies of the two cars is gradually increasing during the period of release. The giving back of this energy is the measure of absorption of the gear. In a gear of 100 per cent absorption the dotted line would be horizontal throughout the release period. In a perfect spring gear (no absorption) the dotted line would be directed upward during this period, reaching the zero line at the instant of parting.

result of this influence is separated from true gear absorption.

Time-Force Curves

Fig. 80k shows the mean forces which develop between the two cars due to draft gear compression and release. The force is

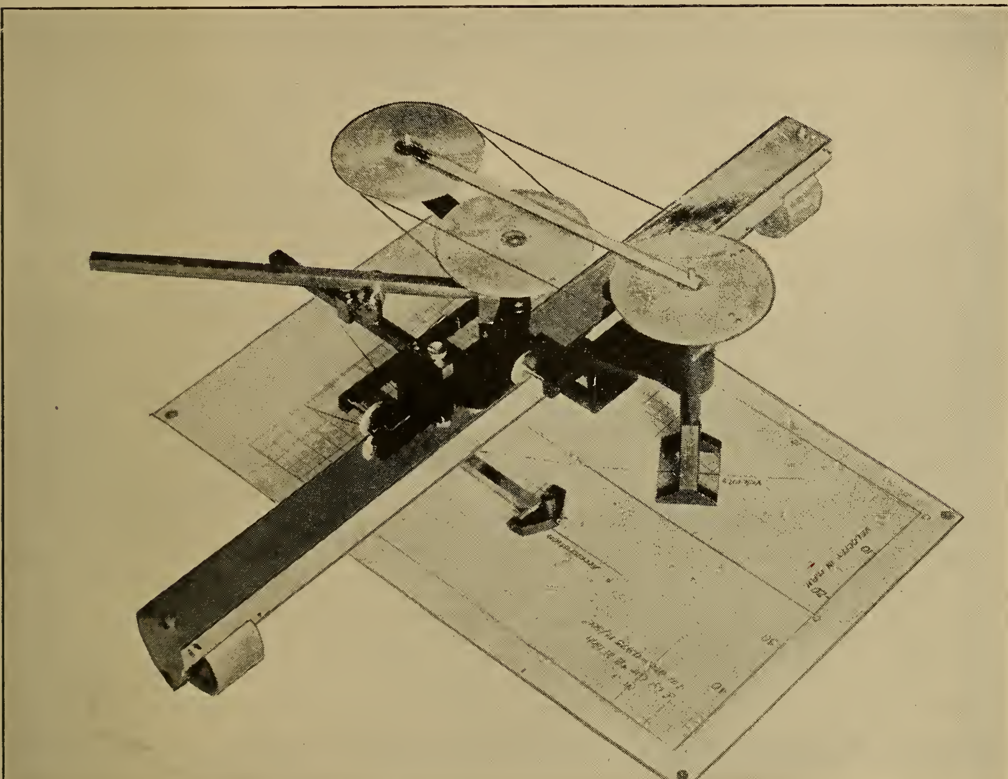


Fig. 57—Mechanical Differentiating Machine

The maximum possible absorption of this run, therefore, was not the full energy of impact, 35,308 ft. lb., but 18,454 ft. lb., the work done in closing the gear; and as the absorption amounted to 16,100 ft. lb., the percentage of gear absorption in this run was 87.2 per cent. Some slight amount of this absortion was due to car resistance. In the tabulations (Figs. 62 and 64) the

plotted against time, and the curve thus shows the building up of the force throughout the period of compression, to a peak at the point of maximum gear closure. During the release period the force falls off suddenly in the case of a friction gear.

The portion of the time-force curve to the left of the peak denotes mean draft gear compression forces while that to the right denotes the forces of release.

In the absence of any more workable and reliable method, the force has been obtained by calculating the forces required to produce the recorded changes of velocity over a given period of time, using the commonly understood laws of motion. It is admitted that the force as determined is deduced from its effect and has not been directly measured. No means for directly measuring a dynamic force has ever been devised. Various methods of a more or less refined nature have been employed to deductively determine the force from one or another of its results. Among the simplest and most elementary of these methods is the deduction of the force from the acceleration of a moving body. The possibility of error must be recognized in this method of figuring. In fact, any effort to compute a force from the result of the force assumes a constancy and uniform continuance over some accepted period of time that is especially questionable in the case of draft gear resistance. Such an assumption does not recognize the probable presence of a succession of higher forces working through lesser periods of time which would be capable of producing and would produce the identical records as to acceleration as a considerably lower force working uninterruptedly over a longer period of time. It is unquestionable that in many of the gears, probably in every case, the sticking and irregularity of gear closure was accompanied by high forces which, because of their very limited duration, could not manifest themselves in the time-displacement curves. Such forces would produce a momentary penetration or over-compression of the car sills, and the very storing and release of this would in itself smooth out the car-movement curves. The mean or average forces and the ultimate peak forces as deduced in these curves, however, are substantially correct and it is questionable whether after all the mean force as depicted, or in other words the force supplied over a long enough period of time to produce penetration or to do the work of rupture, is not the real damaging factor. For the high force of but momentary duration could possibly do little more than to overcome the inertia of the contiguous particles of the sills to which the force is first applied.

The time-force curves will assist in an understanding of the fact that the force between colliding cars is not governed or in any manner reduced by the action of a friction gear over a spring gear of the same characteristics. Energy absorption has in itself no effect whatever upon the compression line. But its influence is immediately apparent in the forces of release. For while it requires high forces to overcome the frictional resistance and to compress a friction gear, the force immediately disappears when the gear starts to release. This action is clearly shown in the time-force curves.

While the peaks of each of these time-force curves show the maximum pressure finally developed between the cars in the particular run, these peaks are not to be considered as the closing forces of the gear. This force is usually higher than the true ultimate resistance of the gear, due to the fact that it is not possible to control the car speeds delicately enough to just close the gears and not over-close them. A very slight over-solid speed will, in a sturdily constructed gear, produce an immediate increase in force, because of the very small yield of the gear housing. In the force-closure curves, which will be discussed hereafter, the true force at the very point of gear closure is given and the results of any slight over-closure are eliminated. It should not, however, be assumed from the foregoing that the closing speeds as given for the various gears are only roughly approximate, as they were in all instances searched out by means of many runs at close intervals around the closing point. An over-solid velocity of even 0.05 M.P.H. will, with a rigid gear construction, greatly increase the momentary peak of this force curve.

Time-Closure Curves

Time-closure curves are developed for each of the runs, Fig. 80n showing such a curve for the single gear run for the Westinghouse D-3 gear. Curve D in this figure has been derived and erected from the superimposed car-movement curves and shows the full yield that took place between the cars, including draft gear compression, center sill compression, and side sill movement. This yield is plotted against time. Curve C is obtained by subtracting from Curve D the amount of the center sill yield and side sill movement, this having been determined from runs made at low speeds when both cars were equipped with solid steel blocks instead of draft gears. Curve C therefore represents the amount of and nature of the true draft gear action, all other influences being eliminated. Curve B was obtained from an entirely different source, namely, from the small drum carried by car B for recording the action of the draft gear in that car (see Figs. 49 and 50). Curves C and B show a remarkable coincidence for all of the gears, incidentally forming a valuable check upon the action of the entire set of recording instruments.

From the time-closure curve it will be seen that the draft gear in this run actually compressed 2.40 in., the difference between this figure and the nominal travel of $2\frac{7}{16}$ in. being due to a shortness of $\frac{1}{32}$ in. in the length of the draft gear pocket in car B, the gear, in other words, being under $\frac{1}{32}$ in. more compression than normal. In general, throughout the tests, slight variations will be found between the gear travels obtained in the car-impact tests and other tests. Such differences are due to the inability to adjust the gear to a nicety in the rough draft gear pockets of the cars. The actual point of gear closure was determined in each instance by the shearing of one or more lead records. The combined compression of the center sills and yield of the side sills of the two cars is represented by the maximum distance between the lines C and D, and in this particular instance amounted to 0.13 in.

In the time-closure curve for the two Westinghouse gears, Fig. 80q, curves B, C and D are similar to curves B, C and D respectively of Fig. 80n. In this case, however, each car was equipped with a draft gear and the curve B, drawn by the draft gear of car B, shows the action of that gear only. Curve A has therefore been produced to show what the gear in car A was doing at the same time, these two curves when combined in the vertical scale producing curve C of the same figure. It will be seen that the two gears did not act in an entirely uniform manner, but that occasionally one of the gears would cease acting for an instant while the other moved. At other times both gears were acting. This character of action occurred both on compression and release, and was visible to the eye when closely watching the movement of the buffers.

Force-Closure Curves

In Fig. 80r is shown a force-closure curve for the closing run of Westinghouse D-3 gear (single gear run). This curve is produced directly from the time-force curve, Fig. 80r, and the time-closure curve, Fig. 80n, by the simple method of eliminating the time element from both of these curves and plotting the force directly against gear closure. This diagram corresponds with the ordinary static card except that it represents the dynamic action of the gear. All of the force-closure diagrams are drawn to the same scale as the static test diagrams, so that the dynamic and static force-closure cards may be directly compared for the same gears. For example, this dynamic card, Fig. 80r, should be compared with the static card shown in Fig. 18 for the identical gear (test gear No. 2).

These curves provide a valuable check upon the fact of complete gear closure. In this particular run a peak force of 207,000 lb. was finally developed between the cars, but from the force-closure curve, Fig. 80r, it will be seen that the peak was reached when the gear was slightly over-solid and that the true solid point of the gear was at a load of 170,000 lb. This latter should therefore be taken as the ultimate dynamic resistance of this particular gear and is the true comparative measure of the load imposed upon the car sills at the instant of gear closure. In all cases a gear was not considered fully closed until one or more lead wires were sheared, following the usual practice as in drop testing; hence in almost every instance a very slight over-solid speed resulted from this effort to credit each gear with its full value. It requires but a very slight impact directly upon the gear barrel or housing to produce a high force peak, especially in a sturdily constructed gear.

The amount of work done and work absorbed may be figured from this card in the same manner as from the ordinary static card, these figures being given in later tabulations. In the double gear runs, Fig. 80t, the two gears did not do equal amounts of work, as can be seen from the superimposed work diagrams.

Solid Buffer Runs

The collision of two cars must always result in more or less penetration or yield of the car structures, the amount of the yield being dependent upon the sturdiness of the cars. In the car-impact tests the car-movement curves are obtained from the side sills of the cars. The records, therefore, are for the movements of the side sills with respect to a fixed point along the track (the datum lines on the drums). The records accordingly do not represent the true and exact movements of the entire masses of the cars, but include the vibrations and relative movements of the side sills with respect to the center sills. In order to ascertain the yield of the Symington test cars under different forces a series of runs was made with both cars equipped with solid steel blocks, 24 sq. in. cross sectional area, instead of draft gears. These were made at approximate impact speeds of $\frac{1}{4}$, $\frac{1}{2}$, $\frac{3}{4}$, 1, $1\frac{1}{2}$, 2, $2\frac{1}{2}$ and 3 miles per hour.

Special arrangements were made to obtain independent records of the yield of the center sills and the whip of the side sills. Certain fundamental data have been set up from these runs as to the yield of the center sills, the whip of the side sills, and the forces between cars with no draft gears, or the forces that should be expected from over-solid velocities with any gear, provided it is as strong a column as the D coupler. These runs also give information in regard to work done and work absorbed by the car bodies and the lading. The records from the runs at approximate speeds of 1 M.P.H. and 2 M.P.H., together with the derived curves, are reproduced in Fig. 58. These curves, while appearing very small in comparison with the later curves made with draft gears in the cars, are reproduced to the same scale as the latter and are directly comparable as to magnitude.

The exact impact velocity in the first of these runs was 1.06 M.P.H., and 1.95

M.P.H. in the second. The period of contact was very short, the entire cycle being but 0.057 seconds in the first run and 0.063 seconds in the latter run. In the first run the cars were in contact for 0.53 inches and for 1.09 inches in the second run. In the first run the maximum force was reached when car B had moved but $\frac{1}{16}$ in. and in the second run when it had moved but $\frac{1}{8}$ in.

In the matter of energy absorption, in run No. 1 there was a possible absorption of 2764 ft. lb. and an actual absorption of 1775 ft. lb., or 64.2 per cent. In run No. 2 the possible absorption was 9357 ft. lb. and the actual absorption 7607 ft. lb., or 82.5 per cent.

The curves, Figs. 59 and 60, have been plotted from the results of these runs. The first of these, Fig. 59, shows the combined yield of the two car bodies at various speeds of impact. The second, Fig. 60, shows the force developed between the cars at various speeds. These curves form the basis for the general deductions made for the influence of the car bodies throughout the tests. It should be remembered that these definite results are for the two particular cars only, but it is believed that they are indicative of the performance of modern cars generally. Incidentally, this force-curve has been compared with a similar curve produced in an entirely different manner by Col. B. W. Dunn, Chief of the Bureau of Explosives, the two curves showing a remarkable coincidence. The yield of the car bodies and the whip of the side sills as determined in these runs form the basis for the corresponding corrections in the succeeding time-closure curves of the draft gears.

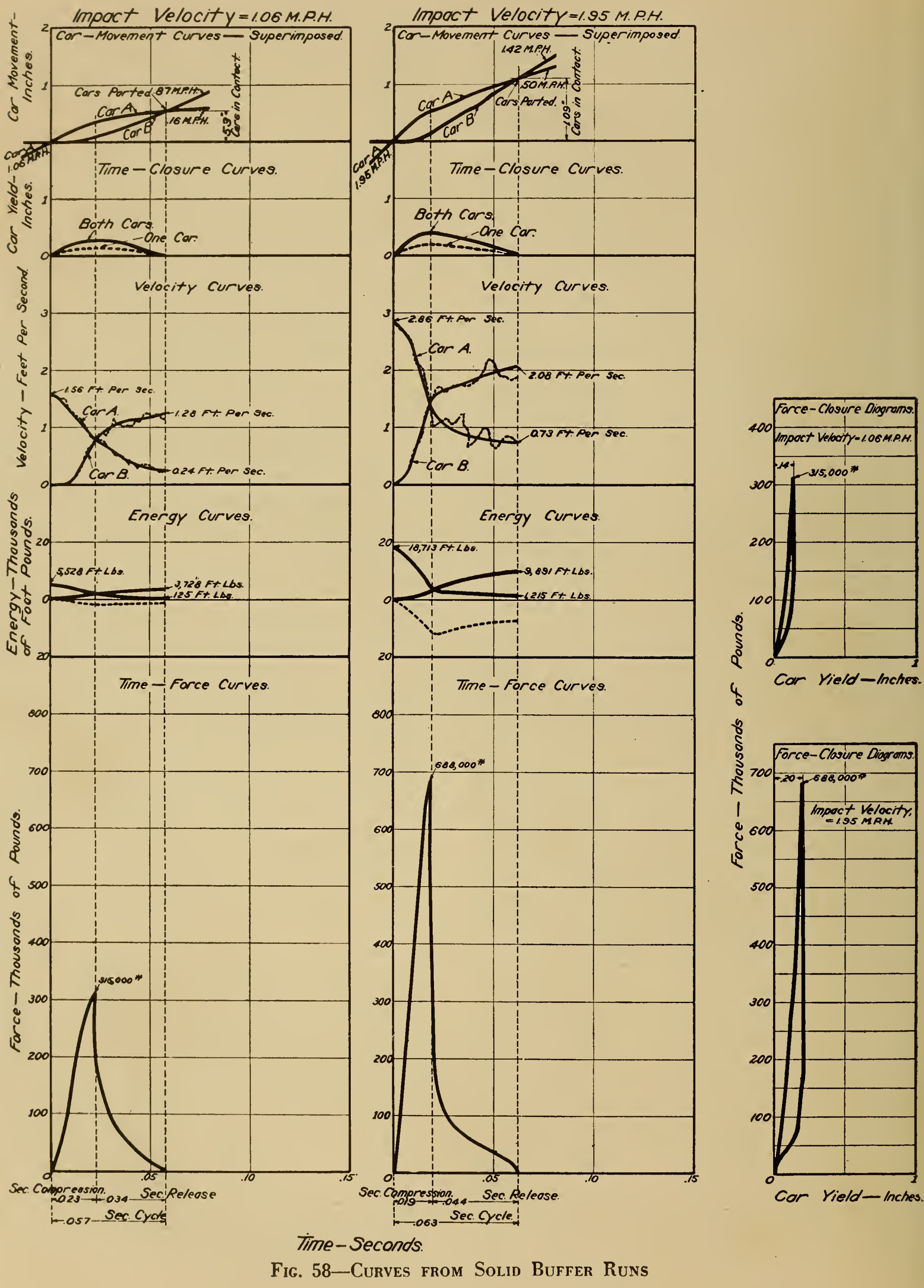


Fig. 58—Curves from Solid Buffer Runs

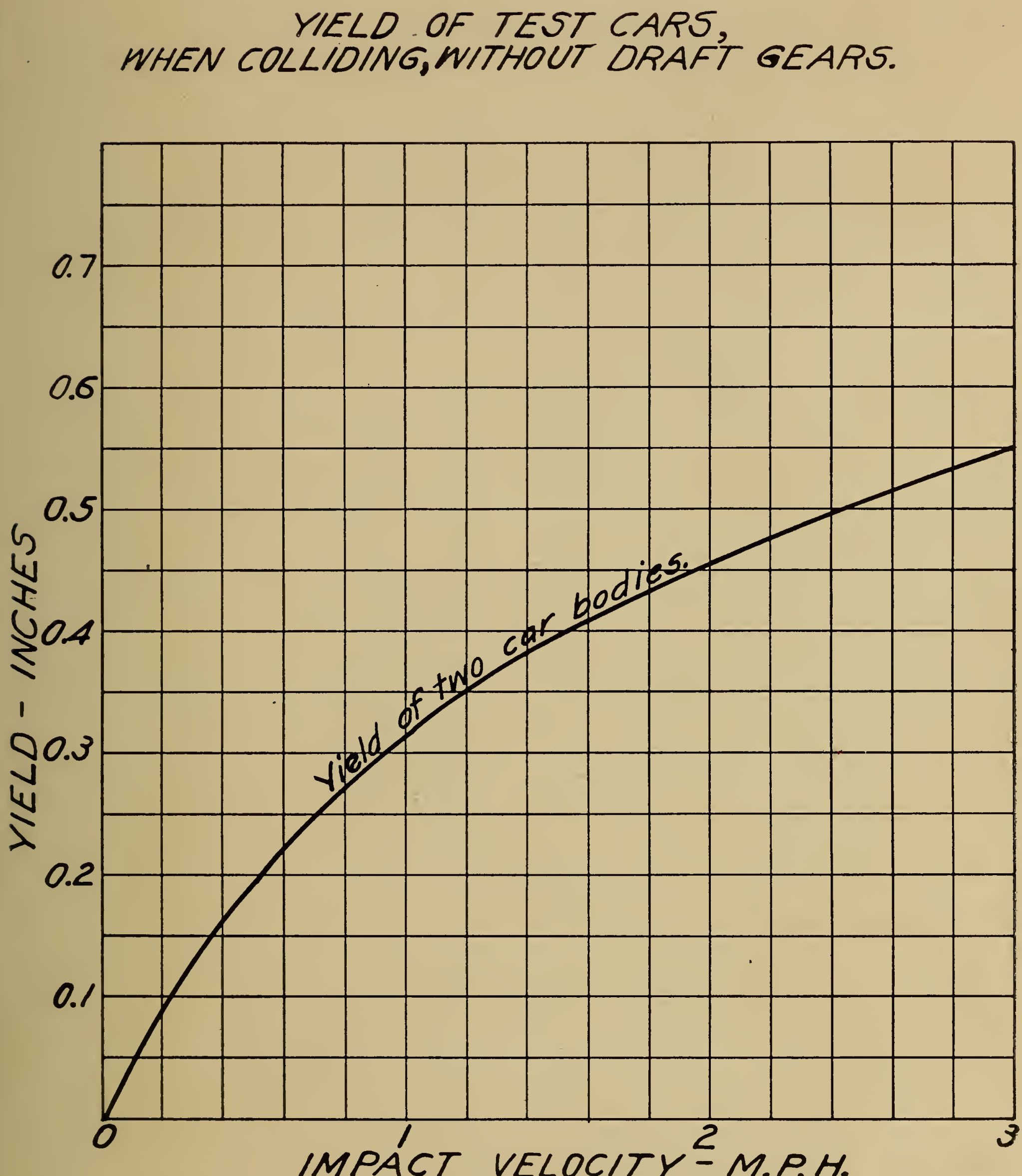


FIG. 59—PLOT OF CAR BODY YIELD AT VARYING IMPACT VELOCITIES

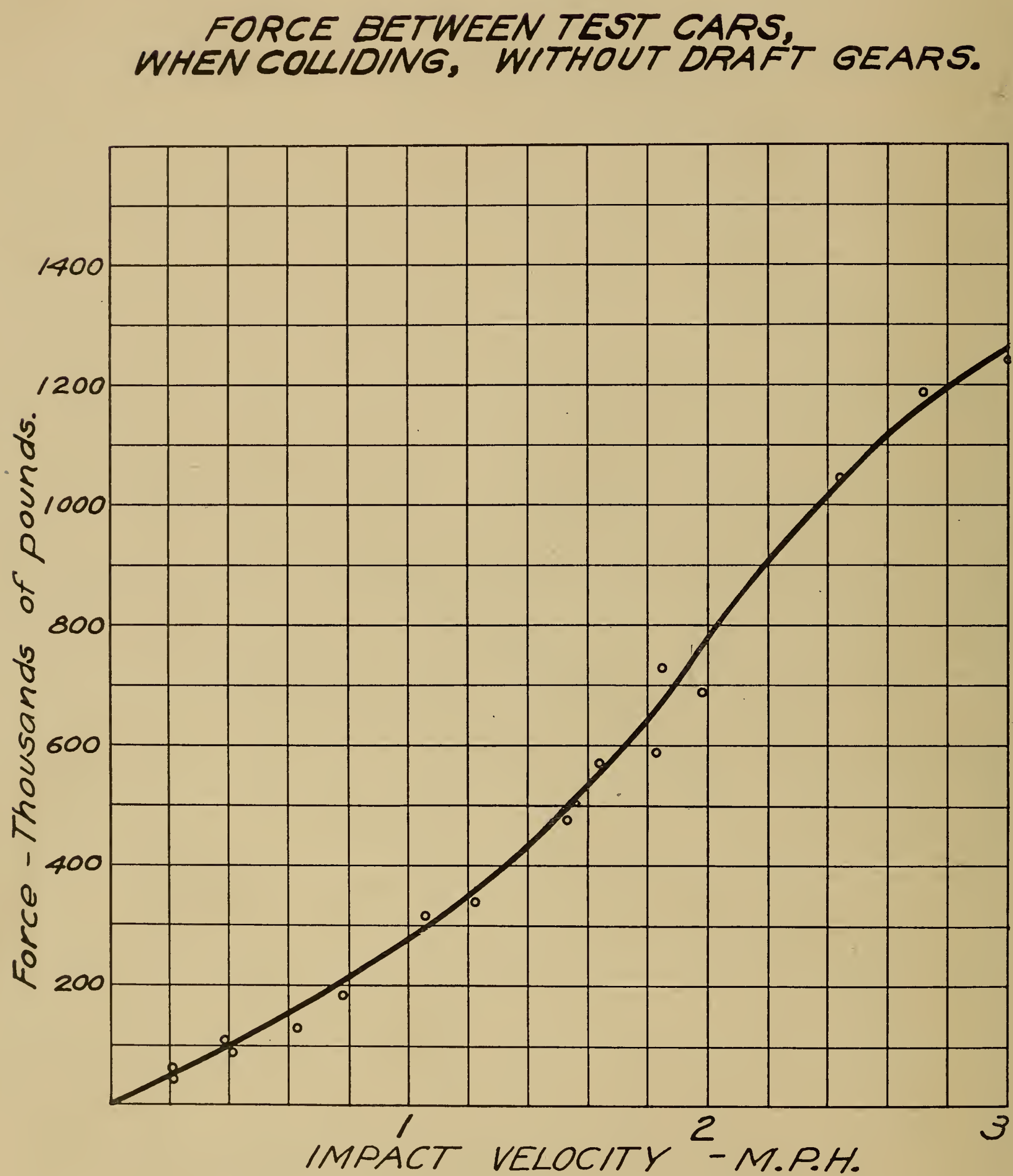


Fig. 60—Plot of Force at Varying Impact Velocities

DISCUSSION OF GEARS IN CAR-IMPACT TESTS

As the first measure of a draft gear is its capacity, or reduced to terms of practice, the impact speed at which it will close, the performance of the several gears in the car-impact tests will be discussed in the order of the closing speeds of the test gears. The later tabulations also are arranged in this order.

National H-1

Gear No. 29 in Car B

Gear No. 30, or Solid Buffer, in Car A

These gears showed the highest capacity, both in the drop test and in the car-impact tests of any of the gears, and their action was good, considering the high closing speed. While the velocity curves show many slight irregularities, yet there were no violent disturbances. The closing run was at a velocity of 5.07 M.P.H., representing almost eight times the energy of the closing run of the spring gear, and more than one and one-half times the energy of a run at 4 M.P.H., hence trembling of the car sides is to be expected. The gear action itself was not so smooth in the closing speed run, but the amount of gear action and the smoothness of car action at 1 M.P.H. with two of these gears is surprising. The two test gears went solid at an impact velocity of 5.07 M.P.H. and the single gear at 3.95 M.P.H. At 1.14 M.P.H., with two gears, the combined gear closure was 1.25 in. In the final run the average work done per gear was 27,184 ft. lb., and the average work absorbed 20,750 ft. lb., or a gear absorption of 76 per cent. The total energy loss in the run from all causes was 51,461 ft. lb., or 41 per cent of the original kinetic energy of the striking car.

In this connection it should be noted that if the gears themselves were of 100 per cent absorption efficiency, the percentage of energy loss in the run from gear absorption could not amount to more than 50 per cent of the original kinetic energy of car A. The gears were slightly over-solid, as can be seen from the force-closure curves, the force peak reaching 820,000 lb. in the run, whereas gear No. 29 closed at 390,000 lb. and No. 30 at 550,000 lb. In all gears the closing point was determined by the shearing of lead wires and the regular practice was to just slightly exceed the capacity rather than to credit a gear with a reduced closing speed. But it should not be assumed that the run was far beyond the capacity of the gear, as in a sturdy gear such as the H-1 a slight excess of energy delivered directly to the gear as a column will at once produce a high force peak. Based upon the average drop test value of this type of gear, a closing speed of 5.09 M.P.H. with 143,000 lb. cars may be expected from two average commercial gears of this type, with a closing force of 466,000 lb.

Sessions Type K

Gear No. 11 in Car B

Gear No. 12, or Solid Buffer, in Car A

The action of these gears in the car-impact tests was not satisfactory. The gears closed in an irregular manner and the car movement curves are the roughest and most irregular obtained in the tests, indicating violent disturbance of the car and lading. The spring barrels of both gears scaled during the test and all of the springs had been solid. The two test gears closed at 4.37 M.P.H. and the single gear at 3.81 M.P.H. At 1.12 M.P.H, with two gears, the combined gear closure was 1.07 in., showing a satisfactory cushioning at this speed. In the final run the average work done per gear was 19,367 ft. lb., and the average work absorbed 16,375 ft. lb., or a gear absorption of 84½ per cent. The total energy loss in the run from all causes was 43,040 ft. lb., or 45 per cent of the original kinetic energy of the striking car. The gears were slightly over-solid, the force peak in the run reaching 400,000 lb., while gear No. 11 closed at 260,000 lb. and gear No. 12 at 165,000 lb. Based upon the average drop test value of this type of gear, a closing speed of 4.33 M.P.H. with 143,000 lb. cars, may be expected from two average commercial gears of this type, with a closing force of 210,000 lb.

The velocity curves of this gear being especially irregular, it may be worth while to repeat here a previous notation regarding the mean velocity curves. After following all the minute variations in the car movement curves with the mechanical differentiater and producing the irregular dotted curves, a second differentiation was made, following the trend of the car-movement curves rather than the local variations. The mean velocity curves were established by this method.

It is especially noticeable that the Sessions K gear, which in the drop test shows considerably less capacity than the Sessions Jumbo (18.8 in. Sessions K, 28.1 in. Sessions Jumbo), required an impact velocity of 4.40 M.P.H. to close two gears, whereas two Jumbo gears, to be discussed later, required a speed of but 4.22 M.P.H.

Miner A-18-S

Gear No. 23 in Car B

Gear No. 24, or Solid Buffer, in Car A

These gears showed high capacity in the car-impact tests. The car-movement curves show some irregularities and the derived velocity curves, while not smooth, are good for a run of this high speed. The gear action was good except when nearly closed, where some pulsations occurred. The two gears did not work uniformly, one being almost closed before the other had compressed more than ⅛ in. This does not mean that at any time there was a lack of cushioning between the cars, as either one or the other of the two gears was yielding at all points of the gear cycle. Such alternating action between the two gears is typical of friction draft gears. Likewise on the release, the gears operated alternately but positively. The two test gears went solid at an impact velocity of 4.46 M.P.H. and the single gear at 3.57 M.P.H. At 1.06 M.P.H. with two gears the combined gear closure was 0.64 in. In the final run the average work done per gear was 18,717 ft. lb. and the average work absorbed 13,334 ft. lb., or a gear absorption of 71 per cent. The total loss in the run from all causes was 40,990 ft. lb., or 42 per cent of the original kinetic energy of the striking car. The gears were slightly over-solid, as can be seen from the force-closure curves, the force peak reaching 640,000 lb. in the run, whereas gear No. 23 closed at 390,000 lb. and gear No. 24 at 390,000 lb. Based upon the average drop test value of this type of gear, a closing speed of 4.33 M.P.H. with 143,000 lb. cars may be expected from two average commercial gears of this type, with a closing force of 368,000 lb.

Westinghouse NA-1

Gear No. 7 in Car B

Gear No. 8, or Solid Buffer, in Car A

The Westinghouse type NA-1 gears in the car-impact tests were highly satisfactory. Both the gear action and the car action were especially smooth, although the two gears did not act together either on compression or release. The velocity curves show the least disturbance of cars and lading found in any gear, except in the case of a few of the very low capacity ones. The two test gears went solid at an impact velocity of 4.16 M.P.H. and the single gear at 3.06 M.P.H. At 0.96 M.P.H, with two gears, the combined gear closure was 1.62 in. In the final run the average work done per gear was 19,167 ft. lb. and the average work absorbed 16,717 ft. lb., or a gear absorption of 87 per cent. The total energy loss in the run from all causes was 39,370 ft. lb., or 46 per cent of the original kinetic energy of the striking car. The gears were slightly over-solid, as can be seen from the force-closure curves, the force peak reaching 500,000 lb. in the run, whereas gear No. 7 closed at 158,000 lb. and No. 8 at 187,000 lb. The curves made with this gear, which has 3 in. travel, show clearly the value of increased length of draft gear travel. Based upon the average drop test value of this type of gear, a closing speed of 4.24 M.P.H. with 143,000 lb. cars may be expected from two average commercial gears of this type, with a closing force of 179,000 lb.

National M-1

Gear No. 32 in Car B

Gear No. 33, or Solid Buffer, in Car A

These gears in the car-impact tests showed rather irregular gear action. The car-movement curves, however, are not bad considering the speed of the run, and the velocity curves do not indicate a violent disturbance of the cars. The two test gears went solid at an impact velocity of 4.26 M.P.H. and the single gear at 3.08 M.P.H. At 1.06 M.P.H., with two gears, the combined gear closure was 1.10 in. In the final run the average work done per gear was 20,000 ft. lb., and the average work absorbed 16,784 ft. lb., or a gear absorption of 84 per cent. The total energy loss in the run from all causes was 40,312 ft. lb., or 45 per cent of the original kinetic energy of the striking car. The gears were slightly over-solid, as can be seen from the force-closure curves, the force peak reaching 580,000 lb. in the run, whereas gear No. 32 closed at 400,000 lb. and No. 33 at 218,000 lb. Based upon the average drop test value of this type of gear, a closing speed of 4.22 M.P.H. with 143,000 lb. cars may be expected from two average commercial gears, with a closing force of 303,000 lb.

Sessions Jumbo

Gear No. 14 in Car B

Gear No. 15, or Solid Buffer, in Car A

This gear made a much better showing in the car-impact tests than the previous Sessions K gear. In fact, for a gear of its capacity, its action is not unsatisfactory. This gear again demonstrates the value of longer gear travel. The two test gears closed at 4.30 M.P.H. and the single gear at 3.26 M.P.H. At 1.02 M.P.H., with two gears, the combined gear closure was 1.14 in. In the final run, with two gears, the average work done per gear was 19,025 ft. lb., and the average work absorbed 14,317 ft. lb., or a gear absorption of 75 per cent. The total energy loss in the run from all causes was 35,660 ft. lb., or 39 per cent of the original kinetic energy of the striking car. The gears were slightly over-solid, the force peak in the run reaching 465,000 lb., while gear No. 14 closed at 137,000 lb. and gear No. 15 at 250,000 lb. Based upon the average drop test value of this type of gear, a closing speed of 4.22 M.P.H. with 143,000 lb. cars may be expected from two average commercial gears, with a closing force of 186,000 lb. The relationship between the drop tests and the car-impact tests of this gear is much closer than in the Sessions Type K.

National M-4

Gear No. 35 in Car B

Gear No. 36, or Solid Buffer, in Car A

The M-4 gear is the smoothest acting and most regular of the National gears, and also shows the highest percentage of absorption. It also shows the lowest ultimate resistance. The individual gears did not work together, but the car-movement curves and the velocity curves, considering the impact velocity, are satisfactory. The two test gears went solid at an impact velocity of 4.12 M.P.H., and the single gear at 3.88 M.P.H. At 1.06 M.P.H., with two gears, the combined gear closure was 1.10 in. In the final run the average work done per gear was 18,467 ft. lb., and the average work absorbed 15,817 ft. lb., or a gear absorption of 86 per cent. The total energy loss in the run from all causes was 38,670 ft. lb., or 46 per cent of the original kinetic energy of the striking car. The gears were slightly over-solid, as can be seen from the force-closure curves, the force peak reaching 360,000 lb. in the run, whereas gear No. 35 closed at 159,000 lb. and No. 36 at 138,000 lb. Based upon the average drop test value of this type of gear, a closing speed of 4.03 M.P.H. with 143,000 lb. cars may be expected from two average commercial gears, with a closing force of 143,000 lb.

Cardwell G-18-A

Gear No. 20 in Car B

Gear No. 21, or Solid Buffer, in Car A

The action of these gears in the car-impact tests was satisfactory. The gear is of $3\frac{3}{16}$ in. travel and this is apparent in length of gear cycle and track movement of cars during the gear cycle. The G-18-A gears have less initial compression than the G-25-A gears, and this is reflected in the greater yield of the two gears in the runs at approximately 1 M.P.H. The two test gears went solid at an impact velocity of 3.85 M.P.H. and the single gear at 2.79 M.P.H. At 1.10 M.P.H., with two gears, the combined gear closure was 1.32 in. In the final run the average work done per gear was 17,117 ft. lb., and the average work absorbed 15,575 ft. lb., or a gear absorption of 91 per cent. The total energy loss in the run from all causes was 35,476 ft. lb., or 49 per cent of the original kinetic energy of the striking car. The gears were slightly over-solid, as can be seen from the force-closure curves, the force peak reaching 295,000 lb. in the run, whereas gear No. 20 closed at 110,000 lb. and No. 21 at 186,000 lb. Based upon the average drop test value of this type of gear a closing speed of 3.89 M.P.H. with 143,000 lb. cars may be expected from two average commercial gears, with a closing force of 214,000 lb.

Cardwell G-25-A

Gear No. 17 in Car B

Gear No. 18, or Solid Buffer, in Car A

The test gears of this type were, as heretofore explained, of higher capacity than commercial gears of the same type previously tested. Consequently it required a higher impact velocity (4.05 M.P.H.) to close the two test gears than is to be expected from the average product. But even though of abnormal capacity the Cardwell test gears showed smooth action both as to gears and cars. In fact, for its capacity, it stands in this respect as one of the most satisfactory of the gears. It is not to be expected that any gear of higher capacity will give the ease of car movement and the smoothness of gear action shown by spring gear with and at its lower capacity. But when a friction gear with a closing capacity of 4 M.P.H. and of $2\frac{3}{4}$ in. travel or less shows reasonably smooth velocity curves in these tests, it may be accepted as a satisfactory gear so far as the service performances of the new gear is concerned. The single test gear went solid at 2.97 M.P.H. At 0.92 M.P.H. the combined travel of the two gears was 0.60 in., reflecting the high initial compression of these gears. In the final run the average work done per gear was 17,917 ft. lb., and the average work absorbed 15,534 ft. lb., or a gear absorption of 87 per cent. The total energy loss in this run from all causes was 38,190 ft. lb., or 47 per cent of the original kinetic energy of the striking car. These gears in the run at 4.05 M.P.H. were slightly over-solid. The force peak reached in the run was 368,000 lb., whereas gear No. 17 went solid at 295,000 lb. and gear No. 18 at 315,000 lb. Based upon the average drop test value of this type of gear, a closing speed of 3.86 M.P.H. (143,000 lb. cars) may be expected from two average commercial gears, with a closing force of 277,000 lb.

Westinghouse D-3

Gear No. 2 in Car B
Gear No. 3, or Solid Buffer, in Car A

In all the runs the Westinghouse D-3 gear showed smooth and regular gear action and a noticeable absence of shock to cars and lading. The two gears closed at 3.65 M.P.H., and the single gear at 2.68 M.P.H. At 1.13 M.P.H., with two gears, the combined gear closure was 2.44 in., reflecting the easy initial movement of this gear. The draft gear action, while slightly variable between the two gears in the double gear runs, is exceptionally good. The velocity curves are good for a friction gear of this capacity. In the final run the average work done per gear was 14,667 ft. lb., and the average work absorbed 12,167 ft. lb., or a gear absorption of 83 per cent. The total energy loss in this run from all causes was 29,864 ft. lb., or 46 per cent of the original kinetic energy of the striking car. The final run was just slightly over-solid, as can be seen from the force-closure diagram. The peak of the force curve reached 285,000 lb., gear No. 2 closing at 195,000 lb. and gear No. 3 at 240,000 lb. Based upon the average drop test value of this type of gear a closing speed of 3.59 M.P.H. (143,000 lb. cars) may be expected from two average commercial gears, with a closing force of 210,000 lb.

Gould 175

Gear No. 41 in Car B
Gear No. 42, or Solid Buffer, in Car A

The Gould gears showed smooth action, but high recoil. The two test gears closed at 3.56 M.P.H. and the single gear at 2.72 M.P.H. At 0.96 M.P.H., with two gears, the combined gear closure was 1.63 in. The velocity curves, while not bad, are yet more irregular than other gears of equal capacity. In the final run the average work done per gear was 13,767 ft. lb., and the average work absorbed 10,100 ft. lb., or a gear absorption of 73 per cent. The total energy loss in this run from all causes was 24,523 ft. lb., or 39 per cent of the original kinetic energy of the striking car. These gears also were slightly over-solid, the force peak in the run reaching 405,000 lb.; gear No. 41 closed at 260,000 lb. and gear No. 42 at 230,000 lb. Based upon the average drop test value of this type of gear, a closing speed of 3.59 M.P.H. with 143,000 lb. cars may be expected from two average commercial gears, with a closing force of 249,000 lb.

Murray H-25

Gear No. 38 in Car B
Gear No. 39, or Solid Buffer, in Car A

In the car-impact tests, as in all the tests of the full program, the Murray gear showed exceptionally smooth and regular action. The car movement curves and velocity curves are among the best, considering the speed of impact, and indicate that there was no violent disturbance of the cars and lading. The two test gears closed at a speed of 3.45 M.P.H. with the 143,000 lb. cars, and the single gear at 2.76 M.P.H. At 0.98 M.P.H. the combined travel of two gears was 0.92 in., reflecting the higher initial resistance of this gear. In the final run the average work done per gear was 13,900 ft. lb., and the average work absorbed 11,584 ft. lb., or a gear absorption of 83 per cent. The total energy loss in this run from all causes was 27,730 ft. lb., or 47 per cent of the original kinetic energy of the striking car. The gears were slightly over-closed, the force of impact finally reaching a peak of 315,000 lb., gear No. 38 closing at 210,000 lb. and gear No. 39 at 130,000 lb. Based upon the average drop test value of this type of gear, a closing speed of 3.52 M.P.H. (143,000 lb. cars) may be expected from two average commercial gears, with a closing force of 227,000 lb.

Christy

Gear No. 52 in Car B
Gear No. 53, or Solid Buffer, in Car A

This gear, closing at a comparatively low speed, produced irregular and erratic gear closure curves and unsatisfactory car-movement curves. Even the low speed runs were not smooth and regular as in most gears. The velocity curves indicate a violent disturbance of the cars. The two test gears went solid at an impact velocity of 3.73 M.P.H. and the single gear at 3.56 M.P.H. At 1.06 M.P.H., with two gears, the combined gear closure was 0.84 in. In the final run the average work done per gear was 12,934 ft. lb., and the average work absorbed 10,909 ft. lb., or a gear absorption of 84 per cent. The total energy loss in the run from all causes was 32,026 ft. lb., or 47 per cent of the original kinetic energy of the striking car. The gears were slightly over-solid, as can be seen from the force-closure curves, the force peak reaching 370,000 lb. in the run, whereas gear No. 52 closed at 194,000 lb. and No. 53 at 150,000 lb. Based upon the average drop test value of this type of gear, a closing speed of 3.50 M.P.H. with 143,000 lb. cars may be expected from two average commercial gears, with a closing force of 151,000 lb.

Miner A-2-S

Gear No. 26 in Car B
Gear No. 27, or Solid Buffer, in Car A

The Miner A-2-S gear showed good action but rather low capacity in the car-impact tests. Both the car action and gear action were especially smooth and among the most satisfactory in the tests. It is noticeable that in the final run with two gears, the gear in car A closed entirely before the gear in car B began to compress. The two test gears went solid at an impact velocity of 3.21 M.P.H. and the single gear at 2.47 M.P.H. At 1.07 M.P.H., with two gears, the combined gear closure was 0.81 in., reflecting the high initial resistance of these gears. In the final run the average work done per gear was 10,025 ft. lb. and the average work absorbed 8,417 ft. lb., or a gear absorption of 84 per cent. The total energy loss in the run from all causes was 24,754 ft. lb., or 49 per cent of the original kinetic energy of the striking car. The gears were slightly over-solid, as can be seen from the force-closure curves, the force peak reaching 525,000 lb. in the run, whereas gear No. 26 closed at 105,000 lb. and No. 27 at 68,000 lb. This high force peak, at a very slight excess of energy, reflects the sturdy nature of the barrel of this gear when called upon to function as a column in over-solid blows. Based upon the average drop test value of this type of gear, a closing speed of 3.26 M.P.H. with 143,000 lb. cars may be expected from two average commercial gears, with a closing force of 89,000 lb.

Waugh Plate Type

Gear No. 49 in Car B
Gear No. 50, or Solid Buffer, in Car A

The Waugh gear showed excellent results in the car-impact tests, although its capacity is limited. The ease with which the standing car is set in motion, considered alone, must commend this gear. Even though showing a high ultimate force, the regularity with which the force is built up eases off the blow and prevents severe shocks and vibrations. The curves show, however, that the action is almost entirely spring action, the absorption being low. The two test gears went solid at an impact velocity of 3.02 M.P.H. and the final run of the single gear was at 1.94 M.P.H. The records from this run show, however, that the single gear was not solid at this speed and that the gear should have been given an impact at 2.20 M.P.H. to fully close the one gear. At 1.06 M.P.H., with two gears, the combined gear closure was 2.34 in., showing a very high yield at this low speed. In the final run the average work done per gear was 9,100 ft. lb., and the average work absorbed 4,117 ft. lb., or a gear absorption of 45 per cent. The total energy loss in the run from all causes was 10,818 ft. lb., or 24 per cent of the original kinetic energy of the striking car. The gears were just closed, as can be seen from the vertical direction of the force-closure curves. The force peak reached 335,000 lb. in the run, and gear No. 49 closed at this point. Gear No. 50 closed at 285,000 lb. Based upon the average drop test value of this type of gear, a closing speed of 2.98 M.P.H. with 143,000 lb. cars may be expected from two average commercial gears, with a closing force of 302,000 lb.

Bradford K

Gear No. 46 in Car B
Gear No. 47, or Solid Buffer, in Car A

This gear in the car-impact tests showed the same unsatisfactory conditions as to the development of the design as in the previous laboratory tests. The springs went solid before the housing came together, thus setting up abnormal wedging forces and high ultimate resistance. One of the rockers cracked during these tests. The gears were of low capacity, the curves showing almost entirely spring action with but little friction. This can be readily seen by comparing the compression and release periods of the gear cycle, which are almost equal. The two test gears went solid at an impact velocity of 2.78 M.P.H. and the single gear at 2.04 M.P.H. At 1.12 M.P.H., with two gears, the combined gear closure was 2.67 in., this being the maximum yield obtained from any of the gears in the low speed run. In the final run the average work done per gear was 6,833 ft. lb., and the average work absorbed 2,150 ft. lb., or a gear absorption of 31 per cent. The total energy loss in the run from all causes was 9,835 ft. lb., or 26 per cent of the original kinetic energy of the striking car. While the heads never came together, the gears were slightly over-solid on the springs, the force peak reaching 340,000 lb. in the run, whereas gear No. 46 closed at 270,000 lb. and No. 47 at 220,000 lb. In view of the defective design of this gear it is hardly proper to set values to be expected from the commercial gears from these test results. But following the same methods used for grading all other gears, namely, based upon the average drop test value found for this type of gear, a closing speed of 2.87 M.P.H. with 143,000 lb. cars may be expected from two average commercial gears, with a closing force of 252,000 lb.

Harvey Springs

Gear No. 55 in Car B
Gear No. 56, or Solid Buffer, in Car A

Two 8 in. x 8 in. Harvey springs, throughout the tests, constituted one gear unit, and in the car-impact tests these springs were applied in twin fashion, one above the other, with a horizontal yoke. The gear action was reasonably smooth, but it is noticeable that most of the yield of the springs had taken place at 1 M.P.H. The car-impact tests of these springs show the same character of compression line as found in the static test and the gear absorption shows furthermore that the high force is the result of friction. The two gears went solid, or reached the previously determined statically solid point of travel, at an impact velocity of 2.33 M.P.H. and the single gear at 1.97 M.P.H. At 1.02 M.P.H., with two gears, the combined gear closure was 2.62 in. In the final run the average work done per gear was 4,992 ft. lb., and the average work absorbed 2,709 ft. lb., or a gear absorption of 54 per cent. The total energy loss in the run from all causes was 8,074 ft. lb., or 30 per cent of the original kinetic energy of the striking car. The gears at this run had reached a solid condition, as can be seen from the vertical trend of the force-closure curves, and also from the increasing roughness of the car-movement curves and the irregularities at the top of the time-closure curves. The force peak reached 490,000 lb. in the run, whereas gear No. 55 closed at 245,000 lb. and No. 56 at 300,000 lb. Based upon the average drop test value of this type of gear, a closing speed of 2.27 M.P.H. with 143,000 lb. cars may be expected from two commercial gears, with a closing force of 259,000 lb.

Class G Coil Springs

Gear No. 58 in Car B
Gear No. 59, or Solid Buffer, in Car A

Two Class G springs, throughout the tests, constituted one gear unit, and in the car-impact tests the springs were applied in twin fashion, one above the other, with a horizontal yoke. The coils were not protected from going solid. The curves obtained from these springs represent the best action obtained, both as to gear action and smooth and gradual movement of the cars. From the velocity curves it will be seen that the cars were eased off with no vibrations or disturbance whatsoever. The capacity, however, is extremely limited and the spring recoil almost 100 per cent. It is of especial interest to note that car A came to rest by the time the cars parted, and that car B, at parting, had almost the initial velocity of car A, this indicating practically total recoil of energy. Another point of interest, and also reflecting the absence of gear absorption, is that the time of draft gear release is practically the same as that of draft gear compression. In fact, if the car-movement curve of car A is reversed and laid upon that of car B the two curves will be found to almost coincide.

The two test gears went solid at an impact velocity of 1.84 M.P.H. and the single gear at 1.45 M.P.H. At 1.07 M.P.H., with two gears, the combined gear closure was 2.00 in. In the final run the average work done per gear was 4,117 ft. lb., and the average work absorbed 450 ft. lb., or a gear absorption of 11 per cent. The total energy loss in the run from all causes was 2,205 ft. lb., or 13 per cent of the original kinetic energy of the striking car. The gears were just solid, as can be seen from the force-closure curves, the force peak reaching 78,000 lb. in the run, whereas both gears closed at 60,000 lb. Based upon the average drop test value of this type of gear, a closing speed of 1.87 M.P.H. with 143,000 lb. cars may be expected from the use of two Class G springs per car, with a closing force of 62,000 lb.

The spring gears were the final ones in the car-impact tests, the Westinghouse D-3 being the first. The excellence of both sets of runs is a check upon the uniform condition of the cars and the instruments throughout the full series of tests.

Summary of Car-Impact Tests

It will be understood that the car-impact tests were made upon two gears only of each type. The table, Fig. 61, shows in Columns 3 and 4 the drop test values of the two test gears used for this purpose and of two average commercial gears respectively. In Columns 5 and 6 are then given the closing speeds of the two test gears and the closing speed that may be expected from two commercial gears. This latter quantity is based upon the relative drop test values of the test gears and the commercial gears. The three general tabulations, Figs. 62, 63 and 64, have been prepared to summarize the actual performance of the test gears in the car-impact tests. In these tabulations the gears appear in the order of the closing speeds of the commercial gears and have been classified according to closing speeds. In studying the performance of the car and the action of the gears there is but little interest in comparing a low speed gear with a high speed gear. The interest lies in comparing the action of and the results from the use of gears of different types and of approximately equal capacities. The gears have accordingly been grouped in these and succeeding tables into four classes, as follows:

Class 1: Gears closing at 5 M.P.H. and over.
- National Type H-1.

Class 2: Gears closing at from 4 to 5 M.P.H.
- Sessions Type K.
- Miner Type A-18-S.
- Westinghouse Type NA-1.
- National Type M-1.
- Sessions Jumbo.
- National Type M-4.

Class 3: Gears closing at from 3 to 4 M.P.H.
- Cardwell Type G-18-A.
- Cardwell Type G-25-A.
- Westinghouse Type D-3.
- Gould Type 175.
- Christy.
- Miner Type A-2-S.

Class 4: Gears closing at less than 3 M.P.H.
- Waugh Plate.
- Bradford Type K.
- Harvey, two 8 in. x 8 in. springs,
- Coil Springs, two 8 in x 8 in., Class G.

In the above classification of gears it will be noticed that while the Cardwell G-25-A test gears actually closed at 4.05 M.P.H., yet from the table, Fig. 61, it will be seen that the average commercial gears of this type properly fall in Class 3, and this gear has accordingly been entered in this class in the general tabulations. Likewise the test gears of the Waugh type actually required 3.02 M.P.H. to close them, but from the average commercial gear this type belongs in Class 4. Asterisks (*) have been placed opposite these gears in the tables because of this fact.

The table, Fig. 62, gives the results of the car-impact tests at the closing speed runs, using a test gear in each car; that of Fig. 63 the results at impact velocities of approximately 1 M.P.H. with a test gear in each car; and that of Fig. 64 the results at the closing speed runs with a test gear in car B only, car A being equipped with a solid steel block instead of a draft gear. These tables need no especial explanation except that it will again be stated that the results as tabulated are for test gears. The comparative action of average commercial gears, from which gear ratings should be deduced, are shown in a later table, Fig. 67.

One of the most noticeable facts brought out by the general action of the gears in the car-impact tests is that draft gears are closed and the maximum force is delivered to the cars before the standing car (car B) has moved any material distance along the track. The maximum distance car B had moved in any of the tests at the instant of maximum force was with the Sessions Jumbo gear, and amounted to 1.59 in. From this it decreased to 0.43 in. with the Harvey springs. This shows that the force between colliding cars is substantially the same, whether the struck car be standing alone or at the head of a draft of cars, as in classification yards. Furthermore, it shows that the force of impact does not extend throughout the sills from end to end, in a horizontal line, but that it is divided into many components, the average of which must be directed toward the center of gravity of the entire mass, and that it gradually decreases in magnitude, due to the fact that each increment of the load resists the force in proportion to its own inertia. As a further demonstration of this, a run was made with a test gear in car B only. At an impact speed of 1.98 M.P.H. the gear closed $1\frac{1}{4}$ in. All of the wheels of the standing car (car B) were then blocked, and the run repeated, car A being released from the same station. In this run the impact speed was 2.02 M.P.H. and the draft gear closed $1\frac{9}{32}$ in., or just $\frac{1}{32}$ in. more than in the preceding run when car B was free to move off. In the second run the wheels of car B slid $1\frac{3}{4}$ in. along the track.

A point of some interest is with respect to the position of the gears in the cars; whether the gear in the striking car or the standing car tends to close first. A number of double gear runs were made, changing the gears from one car to the other. A number of single gear runs were also made, using a test gear in car B, with the solid buffer in car A, and then placing the same gear in car A and applying the solid buffer to car B. No difference in gear action occurred from these manipulations, and the tests showed conclusively that the location of the gear, whether in car A or car B, is immaterial.

When but one of the two cars is equipped with a gear the action is restricted to that gear, and laboratory tests are more nearly reproduced. Throughout the tests the gears used in car B for the single gear runs had been previously tested in the laboratory, and a direct comparison of individual gear action in service and laboratory operation can thus be made by means of the single gear runs. The time-closure curves show generally that when but one car is equipped with a gear the line of actual gear action corresponds closely with the derived line of gear action (lines B and C, time-closure curves). In the double gear runs, however, where two gears are working in opposition, and one or both of them may operate or stick, it will be seen that almost invariably the closure takes place by a succession of alternating movements between the two gears.

Another point of interest in comparing these two classes of runs is in connection with the gear capacities, or, in other words, the closing speed when using one gear of the type or using two gears of the type. The table, Fig. 65, has been prepared to show the relative performance of the single gears and double gears of each type. In this table Column 3 gives the actual closing speed when using the two test gears. Column 4 shows the calculated impact speed

MAKE AND TYPE OF GEAR	TEST GEAR NUMBER	COMBINED DROP TEST VALUE OF THESE TWO TEST GEARS	COMBINED DROP TEST VALUE OF TWO AVERAGE COMMERCIAL GEARS	ACTUAL CLOSING SPEED WITH TWO TEST GEARS M.P.H.	CLOSING SPEED WITH TWO AVERAGE COMMERCIAL GEARS M.P.H.
1	2	3	4	5	6
WESTINGHOUSE D-3	2 CAR B 3 CAR A	41.0"	39.6"	3.65	3.59
WESTINGHOUSE NA-1	7 CAR B 8 CAR A	51.0"	52.0"	4.16	4.24
SESSIONS K	11 CAR B 12 CAR A	38.1"	37.6"	4.37	4.33
SESSIONS JUMBO	14 CAR B 15 CAR A	58.1"	56.2"	4.30	4.22
CARDWELL G-25-A	17 CAR B 18 CAR A	41.5"	37.8"	4.05	3.86
CARDWELL G-18-A	20 CAR B 21 CAR A	38.5"	39.2"	3.85	3.89
MINER A-18-S	23 CAR B 24 CAR A	42.1"	39.8"	4.46	4.33
MINER A-2-S	26 CAR B 27 CAR A	26.0"	26.4"	3.21	3.26
NATIONAL H-1	29 CAR B 30 CAR A	62.0"	62.4"	5.07	5.09
NATIONAL M-1	32 CAR B 33 CAR A	39.1"	38.4"	4.26	4.22
NATIONAL M-4	35 CAR B 36 CAR A	45.0"	43.0"	4.12	4.03
MURRAY H-25	38 CAR B 39 CAR A	33.3"	34.0"	3.45	3.52
GOULD 175	41 CAR B 42 CAR A	35.9"	36.2"	3.56	3.59
BRADFORD K	46 CAR B 47 CAR A	20.9"	21.6"	2.78	2.87
WAUGH PLATE	49 CAR B 50 CAR A	28.5"	27.8"	3.02	2.98
CHRISTY	52 CAR B 53 CAR A	44.5"	39.2"	3.73	3.50
HARVEY 2-8"x8" SPGS.	55 CAR B 56 CAR A	20.6"	19.0"	2.33	2.27
COIL SPRINGS 2-8x8-CLASS G	58 CAR B 59 CAR A	11.4"	11.6"	1.84	1.87

Note - The above speeds are for two cars, each of 143,000 Lbs. total weight.

FIG. 61—TABULATION OF CLOSING SPEEDS OF GEARS; CAR-IMPACT TESTS

Classification of gears	Make and type of gear	Test gear number Car A	Test gear number Car B	Actual velocity M.P.H. Impact Car A	Actual velocity M.P.H. Parting Car A	Actual velocity M.P.H. Parting Car B	Seismograph readings Car A	Seismograph readings Car B	Distance cars traveled from point of impact ft. Car A	Distance cars traveled from point of impact ft. Car B	Amount of draft gear closure inches Car A	Amount of draft gear closure inches Car B	Yield of two car bodies – inches	Time–seconds Compression	Time–seconds Release	Time–seconds Total of D.G. cycle	Track movement of cars during D.G. cycle — inches	Kinetic energy in cars ft. lbs. At impact Car A	Kinetic energy in cars ft. lbs. At parting Car A	Kinetic energy in cars ft. lbs. At parting Car B	Energy loss during D.G. cycle ft. lbs.	Details of energy loss foot lbs. Track and grade resistance	Details of energy loss foot lbs. Absorbed by 2 car bodies	Details of energy loss foot lbs. Absorbed by 2 draft gears	Energy returned by 2 draftgears ft. lbs.	Work done by 2 draftgears ft. lbs.	Absorption efficiency of gears	Force transmitted to car sills by draftgears lbs.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29
5 M.P.H. and over	National H-1	30	29	5.07	1.25	3.70	5.9	6.0	10'-1"	43'-2"	2.46	2.59	17	.091	.155	.246	10.5	126300	7639	67200	51461	2133	7828	41500	12867	54367	76%	550000
4 M.P.H. to 5 M.P.H.	Sessions K	12	11	4.37	1.27	2.95	4.4	4.2	12'-11"	29'-6"	2	2.06	.31	.095	.187	.282	10.7	93820	7930	42850	43040	2213	8077	32750	5983	38733	85%	260000
	Miner A-18-S	24	23	4.46	1.17	3.19	4.6	4.4	11'-1"	37'-10"	2.61	2.61	17	.098	.207	.305	11.7	97800	6760	50050	40990	2617	11706	26667	10766	37433	71%	390000
	Westinghouse NA-1	8	7	4.16	1.22	2.80	4.1	3.8	12'-11"	27'-10"	3.05	2.91	.16	.124	.287	.411	14.8	85100	7320	38410	39370	3268	2669	33433	4901	38334	87%	187000
	National M-1	33	32	4.26	1.21	3.18	4.3	3.9	11'-5"	32'-5"	2.61	2.61	.19	.104	.231	.335	12.4	89591	7203	42076	40312	2805	3940	33567	6433	40000	84%	404000
	Sessions Jumbo	15	14	4.30	1.01	3.20	4.7	4.4	9'-9"	31'-8"	3	2.83	.17	.121	.226	.347	13.8	90740	5030	50050	35660	3105	3922	28633	9417	38050	75%	250000
	National M-4	36	35	4.12	1.22	2.76	4.1	3.7	12'-3"	26'-9"	2.55	2.42	.13	.111	.244	.355	12.7	83600	7330	37600	38670	2910	4127	31633	5300	36933	86%	159000

Fig. 62—Tabulation of Car-Impact Tests. Closing Speed Runs. Double Gear Tests. 143,000-lb. Cars.

(See also facing page)

3M.P.H. TO 4 M.P.H.	CARDWELL G-18-A	21	20	3.85	.19	2.49	2.9	2.8	12'-4"	25'-9"	3.21	3.21	.16	.144	.358	.508	16.7	72930	6946	30508	35476	3437	889	31150	3083	34233	91%	186000	
	CARDWELL G-25-A	18	17	*4.05	1.22	2.67	3.7	3.3	12'-6"	28'-6"	2.73	2.79	.14	.116	.280	.396	14.0	80638	7317	35131	38190	3086	4037	31061	4766	35833	87%	315000	
	WESTINGHOUSE D-3	3	2	3.65	1.01	2.42	3.1	3.1	9'-4"	24'-6"	2.52	2.43	.11	.112	.235	.347	10.9	65437	5235	30338	29864	2329	3202	24333	5000	29333	83%	240000	
	GOULD 175	42	41	3.56	.80	2.66	3.5	3.4	6'-0"	26'-11"	2.57	2.40	.18	.114	.213	.327	10.0	62295	3124	34648	24523	1928	2395	20200	7334	27534	73%	260000	
	MURRAY H-25	39	38	3.45	.91	2.32	2.6	2.6	7'-5"	21'-7"	2.76	2.45	.16	.127	.270	.397	11.6	58419	4105	26584	27730	2588	1975	23167	4633	27800	83%	210000	
	CHRISTY	53	52	3.73	1.09	2.44	3.1	3.2	11'-9"	23'-2"	2.07	2.23	.17	.104	.235	.339	11.1	68230	7018	29186	32026	2295	7914	21817	4050	25867	84%	194000	
	MINER A-2-S	27	26	3.21	.88	2.12	2.0	1.8	7'-0"	17'-0"	2.59	2.45	.27	.138	.287	.425	11.6	50615	3760	22101	24754	2560	5361	16833	3217	20050	84%	105000	
LESS THAN 3M.P.H.	WAUGH PLATE	50	49	*3.02	.43	2.52	2.8	3.0	23"	24'-4"	2.32	2.30	.16	.116	.176	.292	7.6	44866	928	33120	10818	1483	1102	8233	9967	18200	45%	335000	
	BRADFORD K	47	46	2.78	.28	2.37	2.5	2.2	14"	21'-2"	2.52	2.38	.16	.137	.186	.323	7.6	37927	397	27695	9835	1482	4053	4300	9366	13666	31%	270000	
	HARVEY 2-8"x8" SPGS.	56	55	2.33	.32	1.92	1.7	1.6	16"	14'-9"	1.97	1.76	.12	.117	.179	.296	5.9	26647	523	18050	8074	1328	1329	5417	4566	9983	54%	300000	
	COIL SPGS. 2-8"x8" CL. G.	59	58	1.84	0	1.71	1.0	1.0	6"	10'-4"	1.94	1.86	.01	.175	.181	.356	5.7	16666	0	14461	2205	1279	26	900	7333	8233	11%	60000	

Fig. 62—Tabulation of Car-Impact Tests. Closing Speed Runs. Double Gear Tests. 143,000 lb. Cars

(See also facing page)

Classification of Gears.	Make and Type of Gear	Test Gear Number Car A	Test Gear Number Car B	Actual Velocity M.P.H. Impact Car A	Actual Velocity M.P.H. Parting Car A	Actual Velocity M.P.H. Parting Car B	Seismograph Readings Car A	Seismograph Readings Car B	Distance Cars Traveled from Point of Impact Ft. Car A	Distance Cars Traveled from Point of Impact Ft. Car B	Amount of Draft Gear Closure. Inches. Car A	Amount of Draft Gear Closure. Inches. Car B	Time—Seconds Compression	Time—Seconds Release	Time—Seconds Total of D.G. Cycle	Track Movement of Cars During D.G. Cycle Inches	Kinetic Energy in Cars Ft. Lbs. At Impact Car A	Kinetic Energy in Cars Ft. Lbs. At Parting Car A	Kinetic Energy in Cars Ft. Lbs. At Parting Car B	Energy Loss During D.G. Cycle Ft. Lbs.
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)	(20)	(21)
5 M.P.H. and over	National H-1	30	29	1.14	.27	.75	.35	.35	8.0"	21"	.75	.50	.092	.168	.260	2.5	6430	358	2765	3307
4 M.P.H. to 5 M.P.H.	Sessions K	12	11	1.12	.28	.80	.36	.35	7.0"	19"	.79	.28	.075	.128	.203	1.9	6120	380	3110	2630
	Miner A-18-S	24	23	1.06	.30	.70	.13	.22	8.6"	16"	.12	.52	.065	.108	.173	1.5	5500	428	2405	2667
	Westinghouse NA-1	8	7	.96	.15	.67	.10	.16	5.0"	19"	.97	.65	.143	.238	.381	3.0	4545	116	2175	2254
	National M-1	33	32	1.06	.24	.68	.20	.20	7.3"	14"	78	.32	.092	.174	.266	2.3	5570	294	2280	2996
	Sessions Jumbo	15	14	1.05	.27	.67	.22	.25	7.5"	15"	.62	.52	.097	.193	.290	2.5	5445	351	2210	2884
	National M-4	36	35	1.06	.25	.69	.20	.15	8.8"	20"	.61	.49	.093	.182	.275	2.4	5480	307	2360	2813

Fig. 63—Tabulation of Car-Impact Tests. One-Mile-Per-Hour Runs. Double Gear Tests

(*See also facing page*)

3 M.P.H. TO 4 M.P.H.	CARDWELL G-18-A	21	20	1.10	.31	.68	.13	.16	10.3"	16"	.35	.97	.119	.280	.399	3.5	5910	463	2290	3157
	CARDWELL G-25-A	18	17	.92	.25	.59	.17	.17	5.3"	10"	.19	.42	.072	.131	.203	1.6	4075	311	1715	2049
	WESTINGHOUSE D-3	3	2	1.13	.14	.83	.18	.21	5.2"	29"	1.27	1.12	.173	.247	.420	3.9	6150	102	3129	2919
	GOULD 175	42	41	.96	.16	.66	.15	.15	4.8"	16"	.88	.75	.148	.244	.392	3.1	4545	122	2116	2307
	MURRAY H-25	39	38	.98	.24	.57	.13	.17	6.8"	12"	.69	.63	.126	.257	.383	3.0	4754	290	1590	2874
	CHRISTY	53	52	1.06	.26	.74	.30	.30	7.8"	19"	.40	.44	.063	.104	.167	1.5	5580	334	2730	2516
	MINER A-2-S	27	26	1.07	.30	.67	.27	.25	8.8"	15"	.74	.07	.074	.138	212	1.9	5615	446	2210	2959
LESS THAN 3 M.P.H.	WAUGH PLATE	50	49	1.06	.06	.85	.16	.20	4'-0"	31"	1.13	1.21	.189	.230	.419	3.6	5550	159	3575	1816
	BRADFORD K	47	46	1.12	0	.93	.16	.13	4.3"	3'-2"	1.46	1.22	.211	.254	.465	4.2	6215	0	4270	1945
	HARVEY 2-8"x8" SPGS.	56	55	1.02	0	.83	.20	.16	3.6"	30"	1.44	1.18	.206	.248	.454	3.7	5170	0	3365	1805
	COIL SPGS. 2-8"x8" CL. G	59	58	1.07	0	.98	.20	.23	3.1"	39"	1.00	.98	.164	.178	.342	3.1	5670	0	4755	915
	SOLID BUFFER.	X	X	1.06	.16	.87	.45	.40	2.7"	18"	X	X	.023	.034	.057	.53"	5528	125	3728	1675

Fig. 63—Tabulation of Car-Impact Tests. One-Mile-Per-Hour Runs. Double Gear Tests
(See also facing page)

Classification of Gears. (1)	Make and Type of Gear (2)	Test Gear Number Car A (3)	Test Gear Number Car B (4)	Actual Velocity M.P.H. Impact Car A (5)	Actual Velocity M.P.H. Parting Car A (6)	Actual Velocity M.P.H. Parting Car B (7)	Seismograph Readings Car A (8)	Seismograph Readings Car B (9)	Distance Cars Traveled from Point of Impact Ft. Car A (10)	Distance Cars Traveled from Point of Impact Ft. Car B (11)	Amount of Draft Gear Closure. Inches. Car A (12)	Amount of Draft Gear Closure. Inches. Car B (13)	Yield of Two Car Bodies—Inches. (14)	Time—Seconds Compression (15)	Time—Seconds Release (16)	Time—Seconds Total of D.G. Cycle (17)	Track Movement of Cars During D.G. Cycle—Inches (18)	Kinetic Energy in Cars Ft. Lbs. At Impact Car A (19)	Kinetic Energy in Cars Ft. Lbs. At Parting Car A (20)	Kinetic Energy in Cars Ft. Lbs. At Parting Car B (21)	Energy Loss During D.G. Cycle Ft. Lbs. (22)	Details of Energy Loss Foot Lbs. Track and Grade Resistance (23)	Details of Energy Loss Foot Lbs. Absorbed by 2 Car Bodies (24)	Details of Energy Loss Foot Lbs. Absorbed by Draft Gear. (25)	Energy Returned by Draft Gear Ft. Lbs. (26)	Work Done by Draft Gear. (27)	Absorption Efficiency of Gear. (28)	Force Transmitted to Car Sills by Draft Gear. (29)
5 M.P.H. and over	National H-1	X	29	3.95	1.04	2.86	4.1	3.9	7'-7"	28'-4"	0	2.60	.24	.061	.110	.171	5.8	76519	5304	40009	31206	1197	2809	27200	5433	33633	81%	420000*
4 M.P.H. to 5 M.P.H.	Sessions K	X	11	3.81	1.12	2.64	3.7	3.5	10'-1"	23'-0"	0	2.10	.32	.064	.155	.219	6.3	71238	6196	34247	30795	1384	5044	24367	3650	28017	87%	225000
	Miner A-18-S	X	23	3.57	.98	2.55	3.2	3.2	7'-9½"	24'-10"	0	2.54	.42	.062	.155	.217	6.7	62779	4687	31870	26222	1422	9117	15683	4434	20117	78%	300000
	Westinghouse NA-1	X	7	3.06	.84	2.10	2.3	2.2	6'-3"	15'-11"	0	2.96	.35	.081	.188	.269	7.2	45935	3475	21536	20924	1455	6136	13333	2834	16167	83%	227000
	National M-1	X	32	3.08	.88	2.10	2.3	2.2	6'-7"	18'-1"	0	2.52	.25	.083	.171	.254	6.8	46638	3786	21696	21156	1433	856	18867	2433	21300	88%	310000
	Sessions Jumbo	X	14	3.26	.84	2.30	2.9	2.7	6'-2"	18'-11"	0	2.85	.28	.079	.167	.246	6.9	52400	3475	25926	22999	1436	4246	17317	4000	21317	81%	377000
	National M-4	X	35	3.88	1.16	2.61	3.6	3.2	10'-10"	23'-8"	0	2.45	.28	.055	.159	.214	7.2	74027	6653	33470	33904	1455	11599	20856	4150	25000	84%	362000

Fig. 64—Tabulation of Car-Impact Tests. Closing Speed Runs. Single Gear Tests. 143,000-lb. Cars.

(See also facing page)

3M.P.H. TO 4M.P.H.	CARDWELL G-18-A	X	20	2.79	.83	1.83	1.7	1.6	6'-3"	13'-11"	0	3.21	.22	.101	.236	.337	8.0	38317	3361	16484	18472	1498	2674	14300	2200	16500	87%	235000	
	CARDWELL G-25-A	X	17	2.97	.86	2.03	2.2	2.1	6'-11"	16'-10"	0	2.63	.16	.078	.178	.256	6.7	43300	3717	20302	19281	1421	1660	16200	2033	18233	89%	260000	
	WESTINGHOUSE D-3	X	2	2.68	.74	1.84	1.6	1.7	5'-1"	12'-10"	0	2.40	.24"	.084	.166	.250	5.7	35308	2666	16542	16100	1297	1036	13767	1850	15617	88%	170000	
	GOULD 175	X	41	2.72	.60	2.49	2.1	2.1	3'-3"	16'-1"	0	2.37	.27	.074	.146	.220	5.1	36470	1754	20631	14085	1013	1922	11150	3750	14900	75%	300000	
	MURRAY H-25	X	38	2.76	.72	1.92	1.8	1.7	4'-6"	13'-3"	0	2.44	.26	.076	.167	.243	5.8	37592	2559	18181	16852	1305	3664	11883	2917	14800	80%	255000	
	CHRISTY	X	52	3.56	1.12	2.35	3.0	3.2	9'-7"	22'-1"	0	2.08	.33	.059	.141	.200	6.2	62505	6104	27206	29195	1433	8162	19600	2217	21817	90%	250000	
	MINER A-2-S	X	26	2.47	.66	1.73	1.5	1.5	4'-6"	11'-3"	0	2.50	.27	.095	.192	.287	6.1	30109	2167	14655	13287	1361	3459	8467	2033	10500	81%	195000	
LESS THAN 3M.P.H.	WAUGH PLATE	X	49	1.94	.24	1.59	1.3	1.3	8.5"	9'-5"	0	2.23	.15	.095	.135	.230	3.8	18450	283	12438	5729	816	346	4567	3716	8283	55%	180000	
	BRADFORD K	X	46	2.04	.20	1.75	1.4	1.4	7.1"	11'-6"	0	2.45	.23	.099	.136	.235	4.1	20416	204	15089	5123	878	1245	3000	4833	7833	38%	207000	
	HARVEY 2-8"x8" SPGS.	X	55	1.97	.35	1.54	1.2	1.2	12"	8'-10"	0	1.76	.21	.077	.126	.203	3.4	18975	601	11705	6669	707	2179	3783	2867	6650	57%	243000	
	COIL SPGS 2-8x8 CL. G.	X	58	1.45	0	1.31	.5	.3	4"	4'-0"	0	1.78	0	.118	.122	.240	2.9	9836	0	8439	1397								

Fig. 64—Tabulation of Car-Impact Tests. Closing Speed Runs. Single Gear Tests. 143,000-lb. Cars
(*See also facing page*)

at which the single gear in car B should have closed, provided it functioned exactly as in the double gear run, the test gear having been removed from car A. This expected closing speed is based upon the relative work done by the two gears in the double gear runs, the work done by the two car structures being constant. Column 5 gives for comparison the actual speed required to close the single test gear in car B.

Make and Type of Gear	Test Gear Number	Actual Closing Speed with Test Gears in Both Gears M.P.H.	Expected Closing Speed with Test Gear in Car B Only. Based on Double Gear Action.	Actual Closing Speed with Test Gear in Car B Only.
①	②	③	④	⑤
Westinghouse D-3	2 Car B 3 Car A	3.65	2.86	2.68
Westinghouse NA-1	7 Car B 8 Car A	4.16	3.24	3.06
Sessions K	11 Car B 12 Car A	4.37	3.62	3.81
Sessions Jumbo	14 Car B 15 Car A	4.30	3.18	3.26
Cardwell G-25-A	17 Car B 18 Car A	4.05	2.91	2.97
Cardwell G-18-A	20 Car B 21 Car A	3.85	2.60	2.79
Miner A-18-S	23 Car B 24 Car A	4.46	3.74	3.57
Miner A-2-S	26 Car B 27 Car A	3.21	2.62	2.47
National H-1	29 Car B 30 Car A	5.07	3.78	3.95
National M-1	32 Car B 33 Car A	4.26	3.36	3.08
National M-4	35 Car B 36 Car A	4.12	3.24	3.88
Murray H-25	38 Car B 39 Car A	3.45	2.59	2.76
Gould 175	41 Car B 42 Car A	3.56	2.63	2.72
Bradford K	46 Car B 47 Car A	2.78	2.28	2.04
Waugh Plate	49 Car B 50 Car A	3.02	2.23	1.94
Christy	52 Car B 53 Car A	3.73	3.37	3.56
Harvey 2-8"x8" Sprgs.	55 Car B 56 Car A	2.33	1.64	1.97
Coil Springs 2-8"x8" Class G	58 Car B 59 Car A	1.84	1.31	1.45

Fig. 65—Comparison of Double Gear and Single Gear Action. Car-Impact Tests. 143,000-lb. Cars.

COMPARISON OF THE DIFFERENT METHODS OF TESTING

A study of the performance of the individual gears throughout the several different tests can be best made from the tables of Figs. 17 and 66. In most gears a wide difference appears between the static test results and the dynamic results, but in general there is not a wide difference between the drop test results and those in the car-impact tests. Static tests in general are usually made to determine the ultimate resistance of the gear, and the work done and work absorbed. It has generally been supposed that the character of the compression line was indicated by the static tests. These present tests show that the static test is not a measure, either absolute or comparative, of work done, work absorbed or ultimate resistance. For example, in the static test the Westinghouse D-3 gears averaged 18,550 ft. lb. of work done, while in the drop test the average work done was 15,375 ft. lb., the static capacity being 12 per cent higher than the drop capacity. On the other hand, in the National M-1 gear the static result is 263 per cent higher than that of the drop test. No uniformity whatsoever obtains in this percentage.

An interesting example of static testing is in the case of two of the National H-1 gears tested after three years' service on the Norfolk & Western Railroad. These two particular gears were first tested in the static machine and showed an ultimate resistance of 296,000 lb. and 392,000 lb. respectively. The gears were then drop tested and the total fall of the 9,000 lb. weight required to close them was 17½ in. and 16½ in. respectively. The building-up test was then made under the 9,000 lb. drop and after an average of 21 blows per gear the total fall had increased to 29½ in. and 24½ in. respectively. The gears were then retested in the static machine and showed an ultimate resistance of 112,000 lb. and 104,000 lb. respectively. All of these tests were made in a short period of time and under identical conditions. They show most clearly the erratic nature of the static tests. It is found in general, however, that the line of static compression follows the characteristics of the line of dynamic action, and that the ultimate resistance in the two tests are closely proportional to the work done in the tests.

With some few exceptions, the drop test results, as to capacity and absorption, show a fairly uniform relationship to the car-impact results, the latter in general being from 5 per cent to 20 per cent higher than the drop test results. The drop test accordingly would appear to be a fair comparative measure of draft gears for capacity and absorption. The table, Fig. 66, shows the average capacity results from the gears in the different tests, the quantities being the average of those actually obtained for the two gears of a type used in the car-impact tests.

The following general conclusions are drawn from a comparison of the action of the gears throughout the different tests:

1. That the speed of static testing within the limits of the average testing machine has in general but little influence upon the ultimate resistance of the gear.

2. That gears of a type may vary greatly in the static test and at the same time be of approximately equal capacity under the drop.

3. That the static capacity of a gear is no indication whatsoever of its dynamic capacity.

4. That in general, friction gears show greater capacity and higher ultimate resistance in the static test than in any other test.

5. That the ratio of ultimate resistance to work done varies but slightly as between different gears of the same type in the static test.

6. That the ultimate resistance in the static test and in the car-impact test is in general closely proportional to the work done by the gear in these two tests.

7. That the ultimate resistance in the car-impact test and the computed ultimate resistance in the drop test (Column 10, Fig. 17) are in reasonably close proportion to the relative amounts of work done by the gear in these two tests.

8. That in the majority of cases the static curve shows the characteristics of the dynamic action of the gear, but that it is not a true measure of its dynamic capacity or ultimate resistance.

9. That the drop test, with a single gear supported upon the solid anvil, is in general a fair comparative test of gears as to dynamic capacity.

10. That the car-impact results will in general be greater than the drop test results by from 10 per cent to 20 per cent.

11. That the relative recoil of gears may be satisfactorily measured under the 9,000 lb. drop.

12. That neither the drop test, the static test, nor any other test using inelastic means for closing the gear will disclose roughness or irregularity of gear action: That tests upon a resilient body such as a standard car will alone disclose this feature of gear action.

The car-impact tests themselves have established and confirmed numerous principles of gear and car action, among which may be noted:

1. The relative merits of the different methods of draft gear testing.

2. The exact impact velocities at which the various gears will cease to offer further protection to the cars.

3. The production of complete dynamic cards of gear action.

4. The independent and inharmonious action of gears when dynamically closed in opposition to each other.

5. That gear action and car action in practice are not smooth and regular, even with the best friction gears.

6. That a friction gear is necessary for obtaining capacity and for eliminating recoil.

7. That the yield of the car structure and the lading do not afford any material aid in the dissipation of energy, and that friction draft gears in modern cars are essential to avoid high forces and early failure of parts.

8. That preliminary spring action shows no especial value in buffing and that heavy initial gear compression is not disadvantageous.

9. That the force developed between cars in buffing is due to the inertia of the cars, and when the slack is not bunched is the same whether the struck car be standing alone or whether it be at the head of a draft of cars; that the force is practically the same whether the struck car be standing with or without the brakes set.

10. That there is a positive displacement of the center sills relative to the side sills of a car, the amount of which is dependent upon the character of the construction tying these members together.

11. That in a modern steel car, a force equal to the ultimate resistance of the highest capacity gear in these tests will be developed between cars, without draft gears, at an impact velocity of 1½ miles per hour.

12. That if a gear is properly constructed as to sturdiness it requires but a slight over-solid speed to produce a high

Make and Type of Gear	Average Work Done Per Gear. Ft. Lbs.			Average Work Absorbed Per Gear. Ft. Lbs.		
	In Static Test	In Drop Test	In Car Impact Test	In Static Test	In Drop Test	In Car Impact Test
(1)	(2)	(3)	(4)	(5)	(6)	(7)
Westinghouse D-3	18550	15375	14667	16467 88.7%	12533 81.6%	12167 83.0%
Westinghouse NA-1		17250	19167		14712 85.4%	16717 87.2%
Sessions K	34700	14245	19367	32400 93.4%	11164 78.4%	16375 84.5%
Sessions Jumbo	47135	21773	19025	42725 90.6%	17460 80.3%	14317 75.0%
Cardwell G-25-A	49550	15563	17917	47517 96.0%	13298 85.4%	15534 86.6%
Cardwell G-18-A	26250	14453	17117	25000 95.2%	13500 93.5%	15575 90.9%
Miner A-18-S	41084	15769	18717	38417 93.5%	12244 77.7%	13334 71.3%
Miner A-2-S	34667	9762	10025	33167 95.9%	6908 70.8%	8417 83.8%
National H-1		23250	27184		19962 85.7%	20750 76.3%
National M-1	53100	14648	20000	50267 94.7%	12087 82.5%	16784 84.0%
National M-4	31465	16872	18467	29234 93.0%	14337 85.0%	15817 85.8"
Murray H-25	18134	12466	13900	16250 89.6%	10002 80.3%	11584 83.3%
Gould 175	20184	13478	13767	17050 84.6%	8142 60.4%	10100 73.5%
Bradford K	6409	7830	6833	1708 26.7%	4340 55.5%	2150 31.4%
Waugh Plate	8600	10313	9100	4417 51.4%	4512 43.8%	4117 45.2%
Christy		16684	12934		12623 75.7%	10919 84.4%
Harvey 2-8"x8" Spgs.	9034	7722	4992	4767 52.8%	4448 57.5%	2709 54.0%
Coil Springs 2-8"x8" Class G	3800	4279	4117	43.4 11.4%	1208 28.2%	450 10.9%

Fig. 66—Comparison of Work Done and Work Absorbed by Test Gears in Static, Drop and Car-Impact Tests

force peak; conversely, if a gear is not sturdily constructed an over-solid blow may never produce a high force peak, but such over-solid blows will quickly deteriorate the gear, and so reduce its efficiency that low impact speeds will cause damage to the car.

13. That the average period of draft gear compression with a friction draft gear is equal to approximately 1/3 of the entire cycle of impact and that the release occupies approximately 2/3 of the cycle. The maximum period of impact experienced was approximately ½ second.

14. That with a spring draft gear the period of compression and of release are approximately equal and that the spring returns practically all of the energy, bringing the striking car to complete rest and imparting almost the original velocity of impact to the struck car.

15. That several acceptable draft gears are now available capable of protecting a 57½-ton car up to a switching speed of 4 M.P.H. Furthermore, that there is not an occasion for higher switching speeds than 4 M.P.H.

General Deductions

From the tests as a whole the following general deductions can now be made and are recommended by the Inspection and Test Section of the United States Railroad Administration:

1. That for use on any car a gear should be selected which will not go solid at less than 3½ M.P.H. nor more than 4½ M.P.H. when the weight of the particular car to which it is to be applied is considered together with the complete information given in this report.

2. That there is no advantage in buffing from preliminary spring action, and that a draft gear should preferably be under some initial friction compression; not only for the increased capacity effected, but also to hold the friction elements in positive engagement at all times, in order to provide a greater latitude of wear and to prevent the deposit of foreign material upon the friction surfaces.

3. That draft gears should have an effective area for receiving over-solid blows slightly greater in extent than the area of the coupler shank; that this area should be presented in direct line with the force and should preferably be relieved of all other draft gear forces.

4. That all gear units should be of interchangeable dimensions and of equal travel. That considering the results of the high capacity Miner and National gears of 2½ in. travel, both in new condition and after prolonged service, together with the results from the Westinghouse NA-1 gear which is also of high capacity and of 3 in. travel, it is believed that the maximum travel figure of 2¾ in., as set by the Committee on Standards of the United States Railroad Administration, might well be set as a fixed and required standard travel for all new gears.

5. That from this standpoint of satisfactory operation there is no reason why a draft gear of 2¾ in. travel should not be designed with an ultimate dynamic resistance of 500,000 lb., provided the rate of increase of resistance is uniform throughout the travel of the gear.

6. That no gear should be of a greater capacity at this travel than will close at an impact velocity of 5 M.P.H., with 57½-ton cars, or show a greater drop test capacity than 25,000 ft. lb. Such a gear will close in a 120-ton car at 3½ M.P.H.

7. That the expression, "a draft gear of 150,000 lb. capacity," is erroneous and should not be used; and that the ½ in. rivet shearing test as used to define the above expression should be abandoned in favor

of regular 9,000 lb. drop tests, or preferably car-impact tests, until such time as a more convenient test for smoothness of gear action can be developed.

8. That the American Railroad Association, Section 3, should provide itself, with a gravity car testing plant of the general character of that used for these tests, whereupon to conduct such draft gear and car construction tests as may be desired.

RESULTS TO BE EXPECTED FROM COMMERCIAL GEARS

The table, Fig. 67, has been prepared to show in condensed form the average results that may be expected from new commercial gears of the different types. This tabulation embraces all of the different tests and the results in general are based upon the average performance of all of the gears of a type in the tests. This tabulation may be used as the basis for any comparisons desired of average gears.

In Fig. 68 are shown energy curves for cars of different weights, the rotative energy or fly-wheel effect of the wheels and axles, which amounts to an addition of approximately 3 per cent, being included. Horizontal lines representing the closing points of the various gears have been located on this diagram so that the value of any gear upon cars of the different weights may be readily obtained. These horizontal lines for the several gears are based upon the action of the average commercial gear. By means of this diagram the application of the results may be readily converted from a specific case to general cases.

In considering the cushioning value or closing speed of a gear it should be remembered that the kinetic energy of the striking car should be equal to approximately four times the energy required to close one draft gear.

The present report contains much information deduced from the car-impact tests relating to draft gear functioning such as, time of gear cycle, vibrations in car bodies, travel of cars along the track during the several portions of the gear cycle, instantaneous car velocities, transition and absorption of energy, forces developed, comparison of dynamic and static work diagrams, car body absorption and other gear characteristics. This is given, in general, for the closing runs with the single gears and for the 1 M.P.H. runs and the closing runs with the double gears. A wide range of further draft gear information is obtainable from these tests, especially from the intermediate runs made upon each gear and particularly from those just slightly below the closing point. As a specific example of what may be done in this respect, the intermediate runs have been worked up for the Westinghouse D-3 gear and summary curves have been developed. These are shown in Fig. 89, where can be seen for various impact velocities:

(a) The velocities of the cars at parting.
(b) The coefficient of restitution.
(c) The energy absorption.
(d) The absorption efficiency.
(e) The track movement of the cars.
(f) The force developed between the cars.
(g) The time of the draft gear cycle.
(h) The amount of gear closure.

The same factors are also expressed in terms of gear closure instead of impact velocities in curves j to q inclusive of this same figure.

Lack of time has prevented an analysis of all of the gears in this manner, as the immediate effort has been to present sufficient information for each of the several gears to properly compare and grade them. It is hoped to make further studies of an analytical character from these tests, the results to be published when completed. From such studies can be established and verified many of the fundamental laws of draft gears which are at present undeveloped. From the present test data also such studies can be made as: the coefficient of friction under a wide range of conditions, such as various materials, unit pressures and relative velocities of one friction face upon the other; the effect of various spring and friction relationships; angularity of friction faces, etc. In short any further work should be the development of the intermediate runs, the production of summary curves, a study of the fundamentals of gear construction, and the formulation therefrom of mathematical laws of draft gear action.

Classification of Gears	Make and Type of Gear	Nominal Travel	Comparative Weight Per Car	Drop Test: Total Fall To Close	Drop Test: Total Recoil of Weight	Drop Test: Total Fall To Close Surfaces Coated	Drop Test: Blows Req'd To Restore	Drop Test: Total Fall To Shear 18-½" Rivets	Ultimate Resistance: Static Test	Ultimate Resistance: Car-Impact Test	Work Done: Static	Work Done: Drop	Work Done: Car-Impact	Work Absorbed: Static	Work Absorbed: Drop	Work Absorbed: Car-Impact	Percentage of Energy Absorption - Car Impact Tests	Miles Per Hour To Close 2 Gears. 143,000# Cars	Miles Per Hour at Which Gear Failure Will Begin. 143000# Cars	Miles Per Hour To Produce 500000# In Sills of Sym. Test Cars
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)	(17)	(18)	(19)	(20)	(21)
5 M.P.H. and Over	National H-1	2½ Inches	856 Pounds	31.2 Inches	4.6 Inches	15.4 Inches	34 Blows	12.0 Inches	448000 Pounds	466000 Pounds	Ft.Lbs	23400 Ft.Lbs.	26900 Ft.Lbs.	Ft.Lbs	19850 Ft.Lbs	20600 Ft.Lbs.	77 %	5.09 M.P.H.	6.33 M.P.H.	5.12 M.P.H.
4 M.P.H. to 5 M.P.H.	Sessions K	$2\frac{1}{16}$	788	18.8	4.3	8.3		12.5	413000	210000	34700	14100	19100	32400	10875	16140	85	4.33	4.55	4.52
	Miner A-18-S	2½	834	19.9	4.6	13.1	40	19.5	375000	368000	41084	14925	17700	38417	11475	12600	71	4.33	5.03	4.44
	Westinghouse NA-1	3	878	26.0	3.4	12.5	41	23.7		179000		18750	19900		16200	17400	87	4.24	4.56	4.46
	National M-1	2½	744	19.2	3.4	9.9	17	19.5	487000	303000	53100	14400	19600	50261	11850	16500	84	4.22	5.20	4.37
	Sessions Jumbo	3	866	28.1	5.2	17.3		17.0	413000	186000	47135	21075	18300	42725	17175	13800	75	4.22	4.70	4.43
	National M-4	2½	644	21.5	2.4	9.4	4	19.5	397000	143000	31465	16125	17700	29234	14325	15200	86	4.03	4.56	4.40

Fig. 67—Comparative Performance of Commercial Gears, Showing Average Results That May Be Expected from New Gears of Each Type
(See also facing page)

3 M.P.H. TO 4 M.P.H.	CARDWELL G-18-A	$3\frac{3}{16}$	880	19.6	1.5	15.2	30	22.6	272000	214000	36270	14800	17500	33680	13575	15900	91	3.89	4.17	4.11	
	CARDWELL G-25-A	$2\frac{3}{4}$	880	18.9	2.8	11.5	28	20.6	272000	214000	36250	14175	16300	33650	12075	14100	87	3.86	4.04	4.08	
	WEST'NGH'SE D-3	$2\frac{7}{16}$	684	19.8	3.8	11.2	18	20.8	199000	210000	18550	14850	13400	16467	12000	11100	83	3.59	3.93	3.78	
	GOULD 175	$2\frac{1}{2}$	816	18.1	7.1	11.2	52	19.0	262000	249000	20184	13575	14000	17050	8250	10300	74	3.59	3.96	3.75	
	MURRAY H-25	$2\frac{3}{4}$	752	17.0	3.3	9.3	54	18.2	224000	177000	18134	12750	14400	16250	10275	12070	84	3.52	4.00	3.81	
	CHRISTY	$2\frac{1}{2}$	1026	19.6	5.1	12.7	30	14.0	284000	151000		14700	11400		10875	9760	86	3.50	4.16	3.78	
	MINER A-2-S	$2\frac{1}{2}$	698	13.2	3.8	7.4	27	13.2	242000	89000	34667	9900	10300	33167	7050	8660	84	3.26	4.03	3.59	
LESS THAN 3 M.P.H.	WAUGH PLATE	$2\frac{1}{4}$	960	13.9	7.6	11.0		13.3	168000	302000	8600	10425	8870	4417	4725	4020	45	2.98	3.18	3.16	
	BRADFORD K	$2\frac{1}{2}$	772	10.8	5.3	8.9	11	13.0	93000	252000	6409	8100	7280	1708	4125	2290	29	2.87	3.12	3.12	
	HARVEY 2-8"x8" SPGS.	$1\frac{7}{8}$	670	9.5	4.2	7.1	13	9.8	384000	259000	9034	7025	4750	4767	3975	2580	54	2.27	2.80	2.58	
	COIL SPRINGS 2-8"x8"-CL. G	$1\frac{7}{8}$	682	5.8	4.1	4.1	0	7.0	58000	62000	3800	4350	4260	434	1275	465	11	1.87	2.18	2.39	

Fig. 67—Comparative Performance of Commercial Gears, Showing Average Results That May Be Expected from New Gears of Each Type

(*See also facing page*)

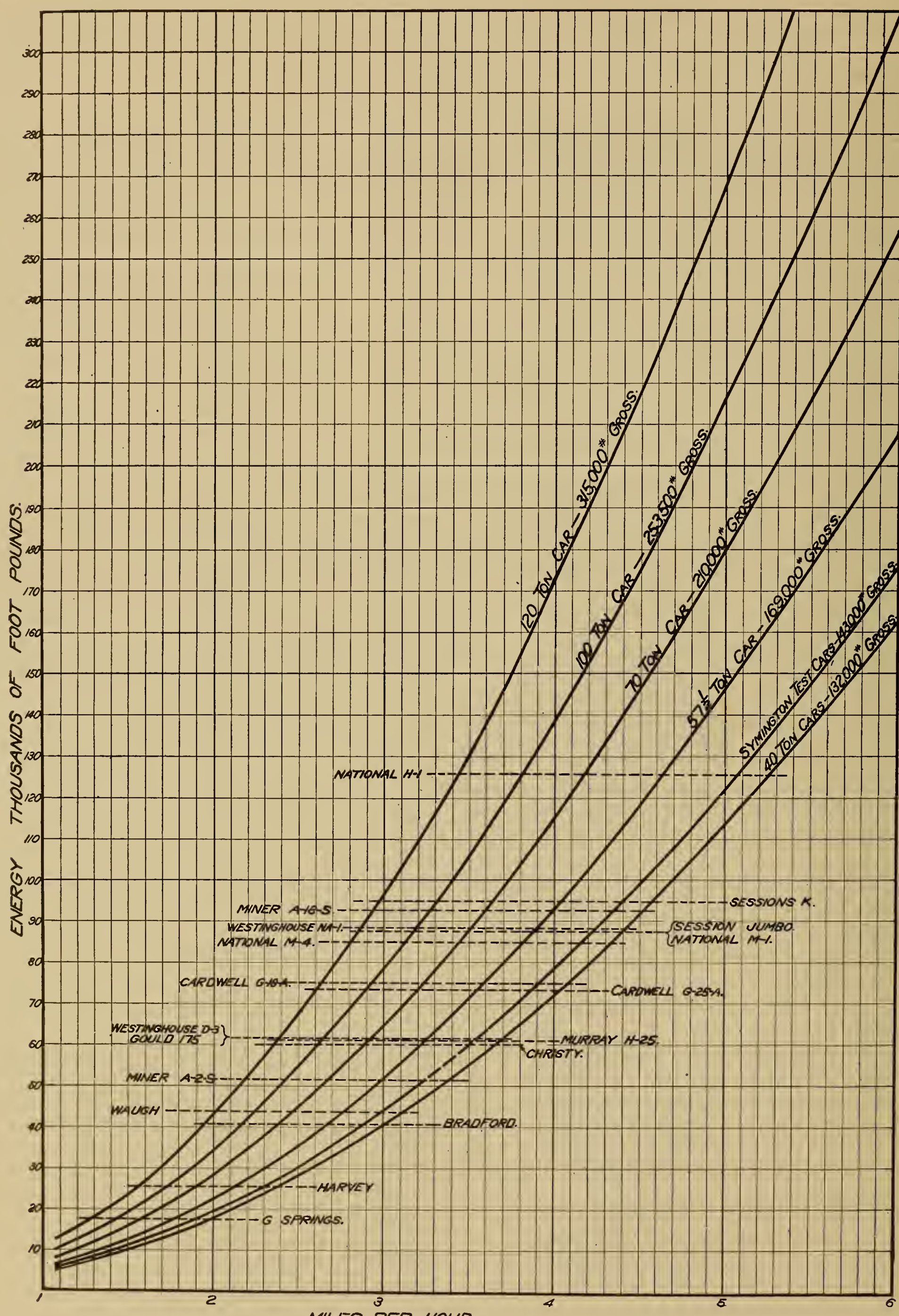


Fig. 68—Energy Curves for Cars of Various Weights, with Commercial Gear Capacities Indicated

GRADING OF AVERAGE COMMERCIAL GEARS

Any one familiar with draft gear operation and testing can from the foregoing results, and particularly from table Fig. 66, establish his own rating of the gears. The relative total merits of the types will differ, depending upon the importance attached to the several features of gear action. No one gear excells in all points. One represents the highest capacity; another the highest percentage of absorption; another the highest degree of smoothness of action.

The tabulation, Fig. 69, has been prepared on the basis of the following relative weights or percentages for the several phases of gear performance:

Capacity	50 points
Smoothness of action	15 points
Closing pressure	5 points
Absorption	15 points
Over-solid sturdiness	10 points
Workmanship and General operation	5 points
Total	100 points

The gradings on the above basis are made directly from the test results, except for the last item of 5 points which represents those features that it is impossible to denote in abstract figures.

Capacity

In setting percentages as above, gear capacity is unquestionably the prime measure. A gear might excell in all other points and yet properly belong at the bottom of the list because of an extremely low closing speed. After a gear is closed it becomes a question of metal to metal for the remainder of the blow, hence the importance of continued gear action at higher impact velocities. The grading of the gears as to the capacity is based upon the square of the closing speed of the commercial gear.

Smoothness of Action

After capacity, the next feature is smoothness of gear and car action. With equal capacities, that gear is the best that will start off the struck car with the least disturbance and vibration of the car structure and the least shifting of the lading. But it is not to be expected that a gear capable of cushioning the blow up to five miles per hour will ease off the cars at its high closing speed like a 2 M.P.H. gear at its lower closing speed. The first gear is doing six times the work of the second gear and doing it in the same limited distance, hence more disturbance is to be expected with this gear at 5 M.P.H. than with the light gear at 2 M.P.H. The grading of the gears for smoothness of action is based upon the relative smoothness of the velocity curves in the closing runs, with the square of the actual impact velocity of the run introduced as a factor.

Ultimate Force or Closing Pressure

All other things, and particularly capacity, being equal, the gear that puts the least force into the sills at the closing point of the gear is entitled to a credit. This is, however, largely allowed for in the preceding grading of smoothness of gear action, inasmuch as the lower and more regular force will produce the smoothest velocity curves. The closing force of a gear, furthermore, is largely governed by the amount of travel of the gear. But in order that those gears that have a dynamic card of decidedly full area may have credit, a weight of five points has been allowed in addition to the previous allowance of 15 points for smoothness of car action. The ratings for the several gears in this respect are not based directly upon the closing pressure of the gear, as it could not be expected that a 5 M.P.H. gear should close at the same ultimate force as a 2 M.P.H. gear. The grading in this respect is based upon the ultimate force per foot pound of closing capacity.

Absorption

While energy absorption, contrary to a popular understanding, does not in any manner reduce or absorb the force between two colliding cars, it is of importance as indicating whether the force between the second and third cars will be the same, due to high recoil of the gears and rebound of the cars, or whether the energy of closure will be partly absorbed. These gradings are made on the basis of percentage of absorption instead of absolute absorption, as a certain amount of recoil is necessary for parting of trains and to insure gear release, the amount of which varies according to the capacities of the gears. A gear with too high a percentage of absorption is likely to stick, especially in train service. The higher the gear capacity the more foot-pounds of energy are needed to insure its release. Hence the percentage of absorption is undoubtedly the fair basis of grading in this respect. In allowing 15 points for absorption it has been borne in mind that the capacity grading alone takes care of absorption in a large measure, for high capacity is impossible except by means of friction, and the introduction of friction at once produces absorption. Hence any gear of high capacity has necessarily a high amount of absorption.

Over-Solid Sturdiness

It is highly important that gears be sturdy enough to withstand reasonable over-solid impacts. For a good showing in over-solid laboratory testing, it is desirable to have a weakly constructed gear, but for endurance and life in service it is necessary to have sturdy parts to receive the solid blows. The grading in this report is based upon the number of over-solid blows required to produce visible gear failure.

Workmanship and General Operation

Under the title of workmanship and general operation are included not only the finished and workmanlike manner in which the gears are constructed but those facts and impressions which have been gained during the progress of the test. Certain gears are finished articles throughout, well designed mechanically and exhibiting careful and accurate manufacturing practices. Other gears are carelessly produced and put together with apparently no thought as to the accurate relationship of the various parts. Some gears failed in certain details before reaching the solid point in the test. Other gears stood extreme punishment without failure. Five points only have been allowed to cover this large variation between the greatest and the least excellence, and it is conceded that this is not enough to represent these differences. The reason that five points only was chosen is because this one item of workmanship and general operation is to a degree a matter of opinion on the part of the testing engineer, and the element of personal opinion is thereby reduced to a minimum.

Service Performance of Gears

It is recognized that the service performance of the gears is one of the most important considerations, but in the absence of positive and uniform service tests for all gears no grading has been made in this respect. Some notes on service tests and service testing appear hereafter.

State of Development of Gears

It is recognized also that those gears are entitled to credit which have been under development and in use for a longer period of time. This factor cannot be reduced to abstract figures, but can be best judged by the history of any particular type of gear on the specific railroad.

Classification of Gears	Make and Type of Gear.	Capacity. 50 Points.	Smoothness. 15 Points.	Ultimate Force. 5 Points.	Absorption. 15 Points.	Sturdiness. 10 Points.	Workmanship and General Operation. 5 Points.	Total
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)
Class 1 5 M.P.H. & over	National H1	50	7	3	12	10	5	87
Class 2 4 M.P.H. to 5 M.P.H.	Westinghouse NA-1	36	15	4	15	3	5	78
	Miner A-18-S	38	8	3	13	6	5	73
	National M1	36	9	3	13	7	5	73
	National M4	35	8	5	14	5	5	72
	Sessions Jumbo	36	9	4	13	3	4	69
	Sessions K	38	2	4	13	1	2	60
Class 3 3 M.P.H. to 4 M.P.H.	Cardwell G-25-A	30	11	4	15	2	3	65
	Cardwell G-18-A	30	9	4	15	2	4	64
	Westinghouse D3	25	11	4	14	3	5	62
	Murray H-25	25	10	4	14	3	4	60
	Miner A-2-S	21	10	5	14	5	5	60
	Gould 17,5	25	8	3	12	3	4	55
	Christy	24	5	4	14	6	1	54
Class 4 Less than 3 M.P.H.	Waugh Plate	18	10	1	8	2	5	44
	Harvey 2-8"x8" Springs	10	5	1	9	3	5	33
	Bradford K	16	7	2	5	1	1	32
	Coil Springs 2-ARA-Class G	7	4	3	2	2	5	23

(A word of caution is necessary in using this table. While in most cases the statement is made in the chapter on "Selection and Condition of Test Gears," page 24, that the results of the tests are believed to be representative of the action of the commercial product, certain exceptions are noted, namely: Cardwell G-25-A, Murray H-25, Bradford K and Christy.—EDITOR.)

FIG. 69—GRADING OF GEARS, BASED UPON PERFORMANCE OF NEW COMMERCIAL GEARS

SERVICE TESTS

It has not been possible during the period of the present test work to begin the comprehensive series of service tests desired. It has been planned to equip not less than 20 cars with each type of gear in these tests and to run all of the cars in the same restricted service so that uniform treatment may be accorded each type of gear. The gears are to be inspected, measured, and drop tested before application to the cars and are to be kept under continual observation. Ten cars with each type of gear to have Farlow draft gear attachment and ten to have yoke and lug attachments. Five cars in turn with each type of attachment are to have the draft gear protected by allowing the coupler horn or the Farlow middle key to strike. The remaining five, with each type of attachments are to have the full load delivered to the sills through the draft gear. The condition of the gears, attachments and cars is to be reported each year and all gears are to be removed for laboratory tests after two years of service. Those worthy of further testing are to be continued in the test for an additional period of two years. It is only in such a careful and comprehensive test that reliable service information can be obtained.

A similar service test embracing five types of gears has been under way on the Norfolk & Western Railway for approximately four years. The United States Railroad Administration Inspection and Test Section was thereby afforded an opportunity to observe the service action of test gears of several different types. In the Norfolk & Western tests the National H-1 and Miner A-18 gears showed the least percentage of depreciation. Gears representative of the average condition of these two types, after three years of service on Norfolk & Western 100-ton coal cars in restricted tide-water-service and averaging at least 50 miles per day, were given car-impact tests by this Section. A number of gears of each type in the Norfolk & Western test had been removed and tested under the 9,000 lb. drop. These had been carefully handled from the cars to the testing machine and the closing point was determined in as few blows as possible without disturbing the foreign material upon the friction surfaces. Two average gears of the National H-1 and the Miner A-18 were also taken to the car-impact test plant at Rochester and tested in the same condition as to friction surfaces. The worn Miner gears closed at an impact speed of 3.48 M.P.H. and the National at 4.21 M.P.H. The capacity of the National gears averaged 21,000 ft. lb. and the Miner 14,200 ft. lb. The gear action and cushioning were good in both instances and the actual capacity of these gears and the protection being afforded the cars after such a period of service is unexpectedly high.

The table, Fig. 37, shows the average condition of the several gears in the Norfolk & Western tests, and shows by means of the drop test, what each type is actually doing in service. The table shows also, by means of the restoration test results, what portion of the depreciation is probably due to wear and what to foreign material upon the friction surfaces. These service tests were made in a careful and exact manner and with uniform conditions for all gears.

TRAIN-OPERATION TESTS

It is desirable to make a series of train-operation tests of draft gears before fully determining upon ideal gear characteristics. But a complete mathematical analysis of the car-impact data contained in this report should be made before beginning such road tests. All of the necessary information is at hand in these present results for the accurate calculation and determination of the ideal gear for train starting and handling as well as in yard service. After such analysis is made a rational program of train tests can then be formed for confirming the calculated results. This method will not only insure a test program directed straight to the ends sought but will also obviate many unnecessary tests that would otherwise be made in searching for the desired information.

In connection with the Norfolk & Western service tests an opportunity was presented for obtaining some limited information as to the action of draft gears in actual train service. In this test, adjoining cars were equipped with gears of different capacities and characteristics, and by means of chronographs, each gear was caused to draw a continuous line of gear action upon a moving ribbon of paper. The report of this test, dated November 4, 1918, is appended to the present draft gear report as a matter of general information and record. (See Appendix A.)

TESTS OF DRAFT GEAR ATTACHMENTS

While testing draft gears, the opportunity was presented for making car-impact tests of draft gear attachments. Two 70-ton United States Railroad Administration low side gondolas were equipped with Farlow two-key draft gear attachments and tested in comparison with United States Railroad Administration standard cast steel yoke and lug attachments. A full report of this test, together with tests of wood car construction with and without metal draft arms, is attached to this report as Appendix B.

MAKE AND TYPE OF GEAR	CAR-MOVEMENT CURVES			VELOCITY CURVES			ENERGY CURVES			TIME-FORCE CURVES			TIME-CLOSURE CURVES			FORCE-CLOSURE CURVES		
	SINGLE GEAR. CLOSING	DOUBLE GEAR. 1-MPH.	DOUBLE GEAR. CLOSING.	SINGLE GEAR. CLOSING.	DOUBLE GEAR. 1-MPH.	DOUBLE GEAR. CLOSING	SINGLE GEAR. CLOSING	DOUBLE GEAR. 1-MPH.	DOUBLE GEAR. CLOSING.	SINGLE GEAR. CLOSING.	DOUBLE GEAR. 1-MPH.	DOUBLE GEAR. CLOSING.	SINGLE GEAR. CLOSING	DOUBLE GEAR. 1-MPH.	DOUBLE GEAR. CLOSING.	SINGLE GEAR, CLOSING.	DOUBLE GEAR. 1-MPH.	DOUBLE GEAR. CLOSING.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19
NATIONAL H-1	FIG. 71-a	FIG. b	FIG. c	FIG. d	FIG. e	FIG. f	FIG. g	FIG.	FIG. j	FIG. k	FIG	FIG. m	FIG. n	FIG.	FIG. q	FIG. r	FIG.	FIG. t
SESSIONS K	72-a	b	c	d	e	f	g		j	k		m	n	p	q	r		t
MINER A-18-S	73-a	b	c	d	e	f	g		j	k		m	n	p	q	r		t
WESTINGHOUSE NA-1	74-a	b	c	d	e	f	g		j	k		m	n		q	r		t
NATIONAL M-1	75-a	b	c	d	e	f	g		j	k		m	n		q	r		t
SESSIONS JUMBO	76-a	b	c	d	e	f	g		j	k		m	n	p	q	r		t
NATIONAL M-4	77-a	b	c	d	e	f	g		j	k		m	n		q	r		t
CARDWELL G-18-A	78-a	b	c	d	e	f	g		j	k		m	n		q	r		t
CARDWELL G-25-A	79-a	b	c	d	e	f	g	h	j	k	l	m	n	p	q	r	s	t
WESTINGHOUSE D-3	80-a	b	c	d	e	f	g	h	j	k	l	m	n	p	q	r	s	t
GOULD 175	81-a	b	c	d	e	f	g	h	j	k	l	m	n	p	q	r	s	t
MURRAY H-25	82-a	b	c	d	e	f	g	h	j	k	l	m	n	p	q	r	s	t
CHRISTY	83-a	b	c	d	e	f	g		j	k		m	n		q	r		t
MINER A-2-S	84-a	b	c	d	e	f	g		j	k		m	n	p	q	r		t
WAUGH	85-a	b	c	d	e	f	g		j	k		m	n	p	q	r		t
BRADFORD K	86-a	b	c	d	e	f	g		j	k		m	n	p	q	r		t
HARVEY	87-a	b	c	d	e	f	g		j	k		m	n	p	q	r		t
COIL SPRING	88	b	c		e	f			j			m		p	q	r		t

FIG. 70—LIST OF AND INDEX OF CAR-MOVEMENT CURVES AND DERIVATIVE CURVES, EMBRACING FIGURES 71 (A TO T) TO 88 (A TO T) INCLUSIVE

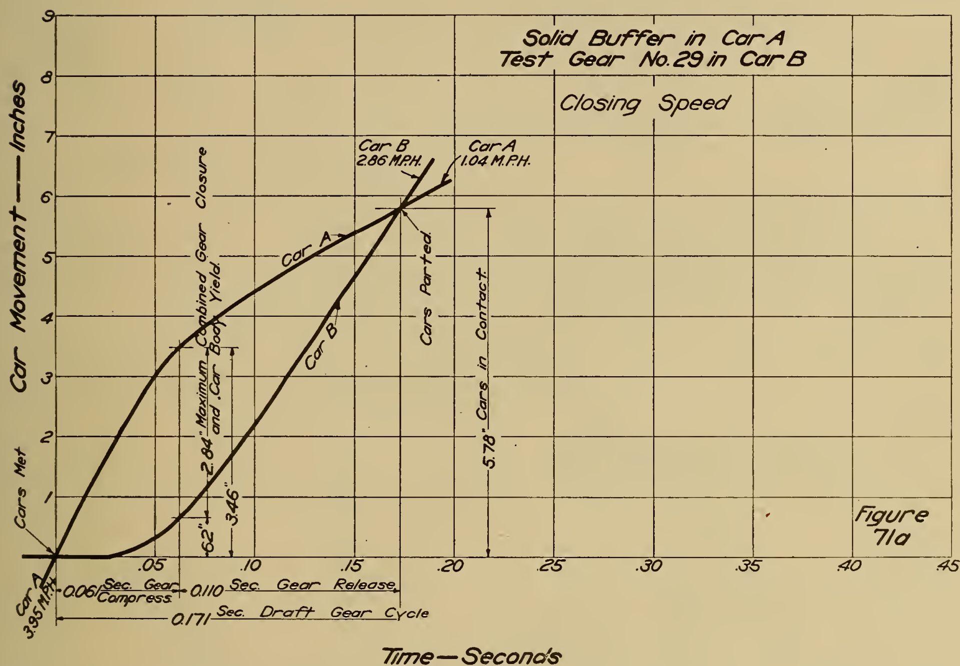
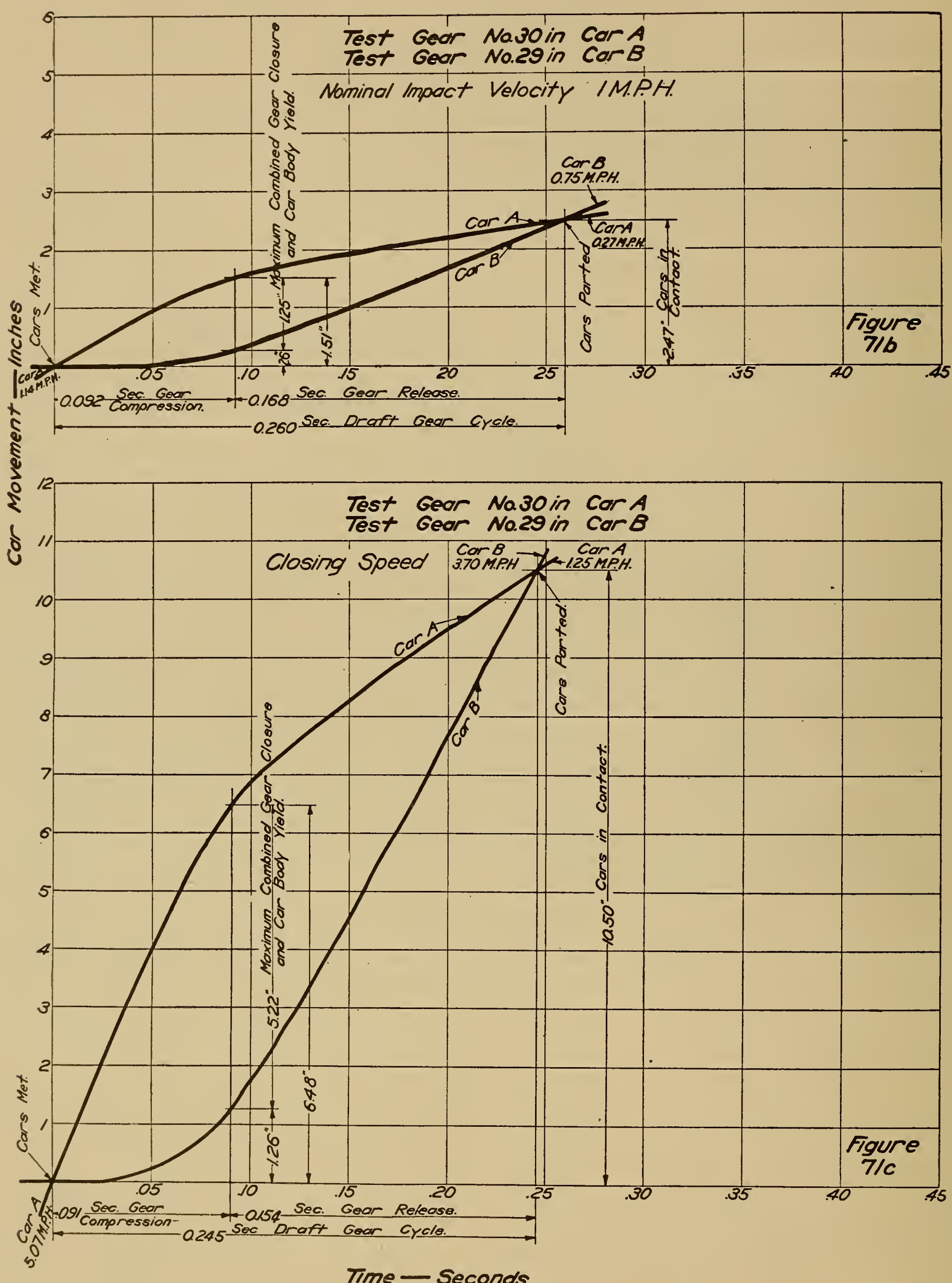
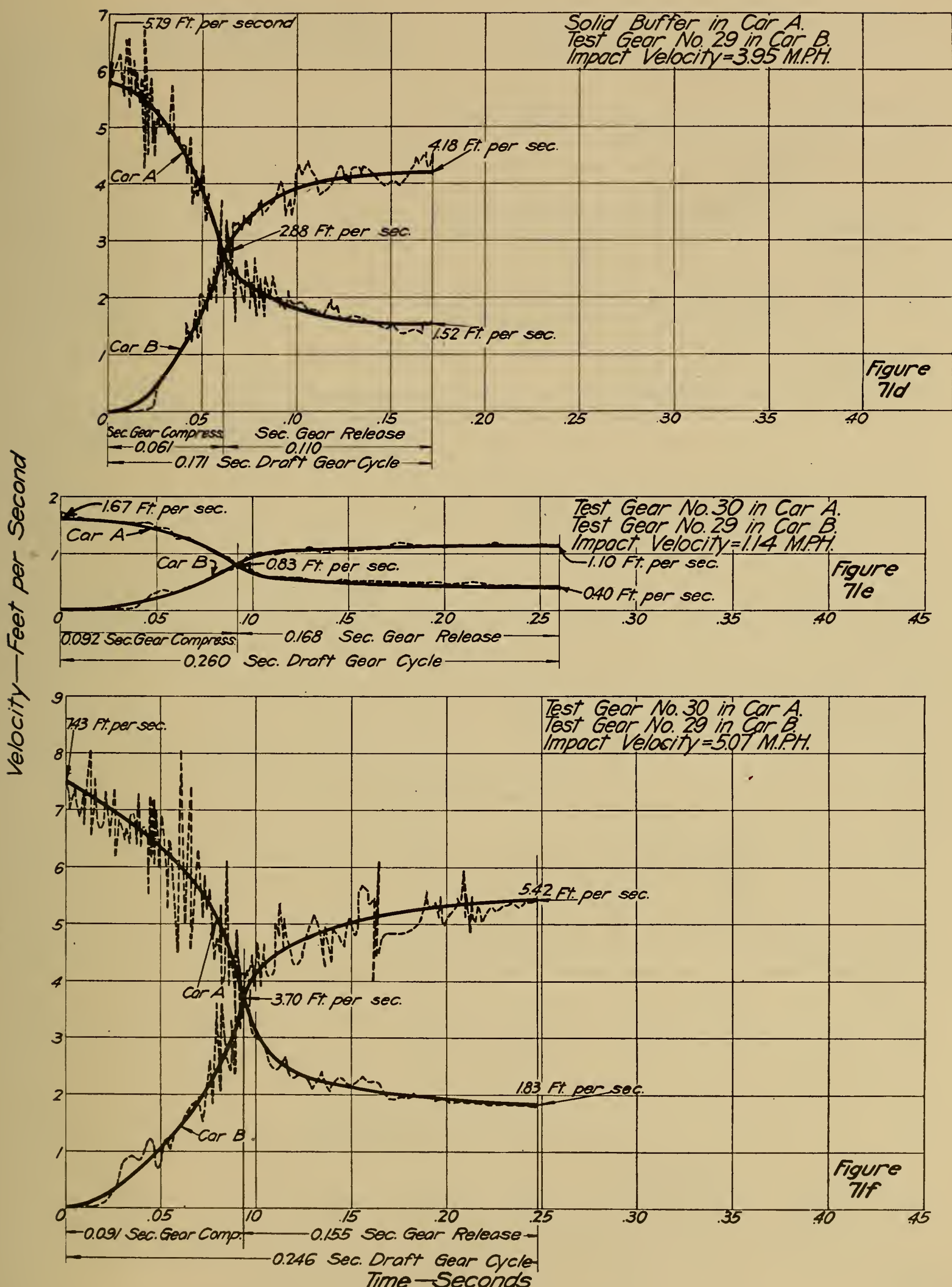


FIG. 71A—CAR-MOVEMENT CURVES, SUPERIMPOSED. NATIONAL H-1 GEARS. THESE CURVES DRAWN BY CARS IN TEST



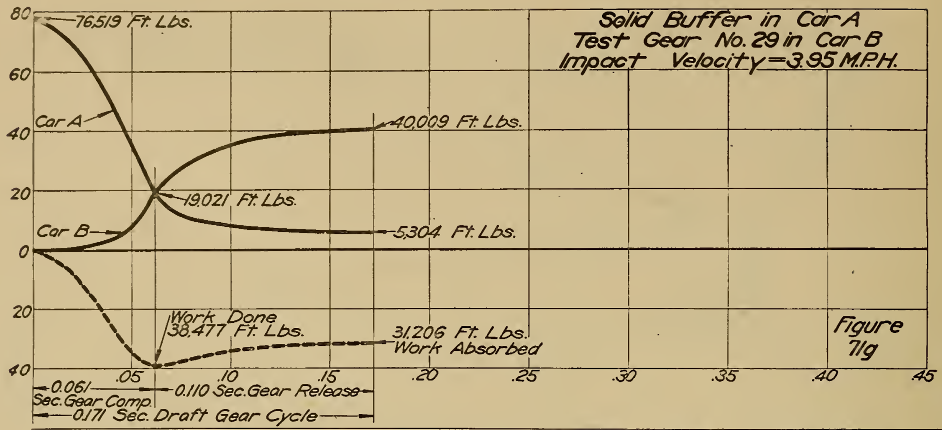
Figs. 71b and 71c—Car-Movement Curves, Superimposed. National H-1 Gears
These Curves Drawn by Cars in Tests

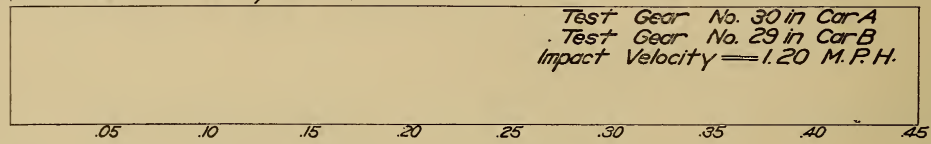


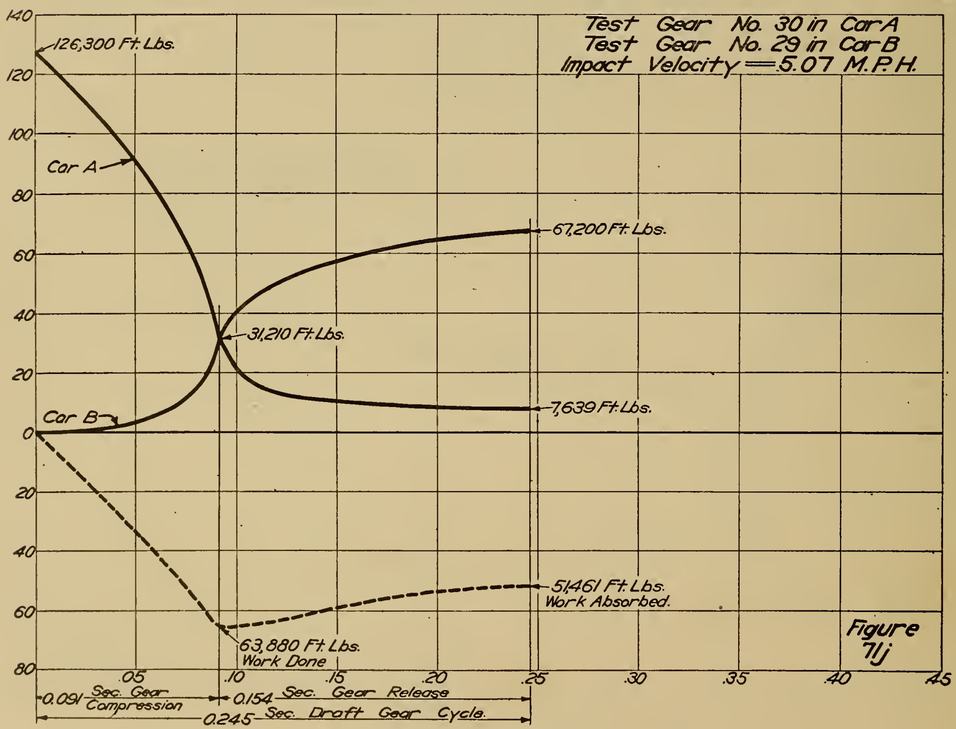
Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

The irregularities are due in general to vibrations of the car structure induced by draft gear action.
Full lines represent mean velocity curves.

FIGS. 71D, 71E AND 71F—VELOCITY CURVES, NATIONAL H-1 GEARS



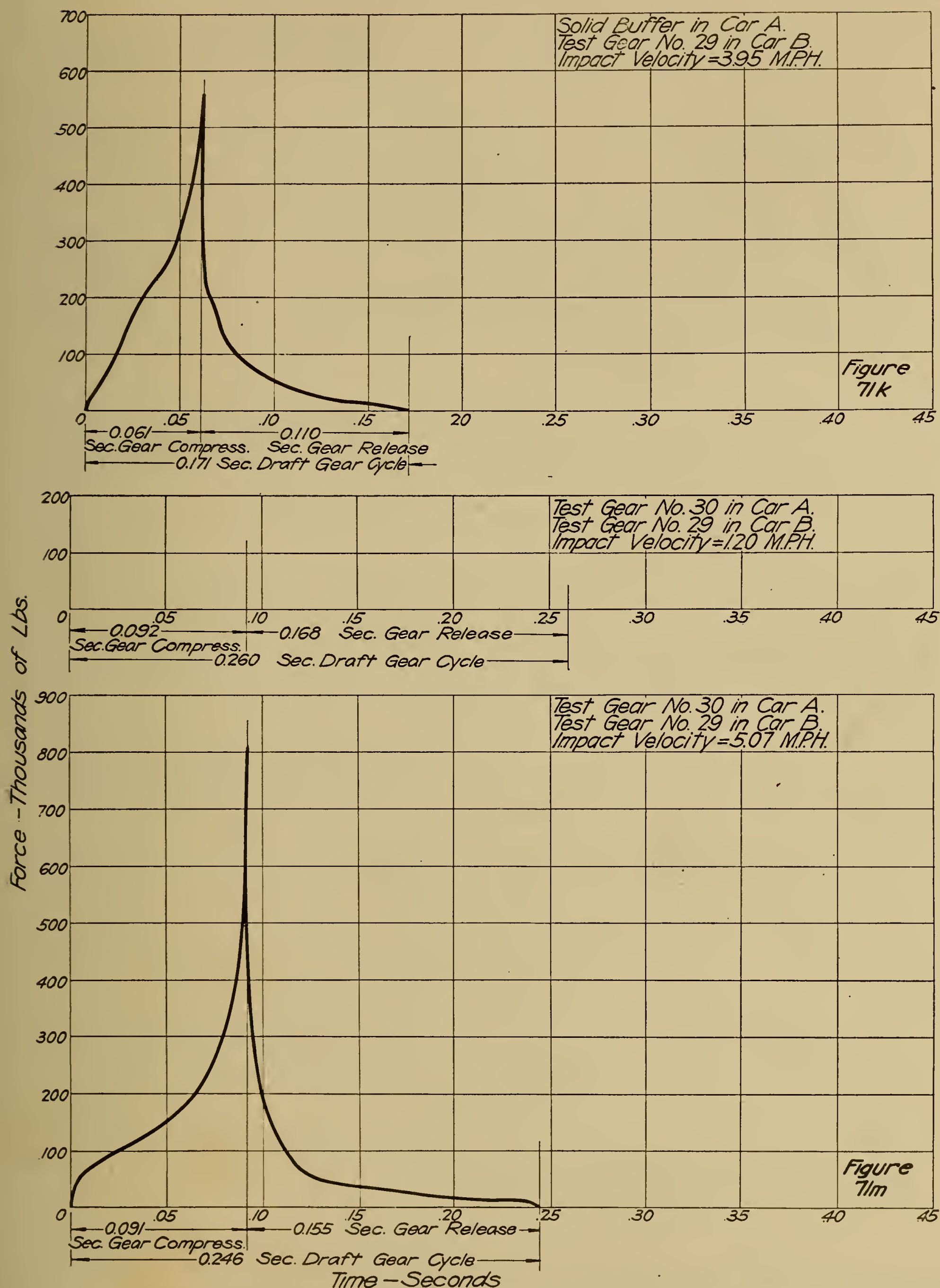




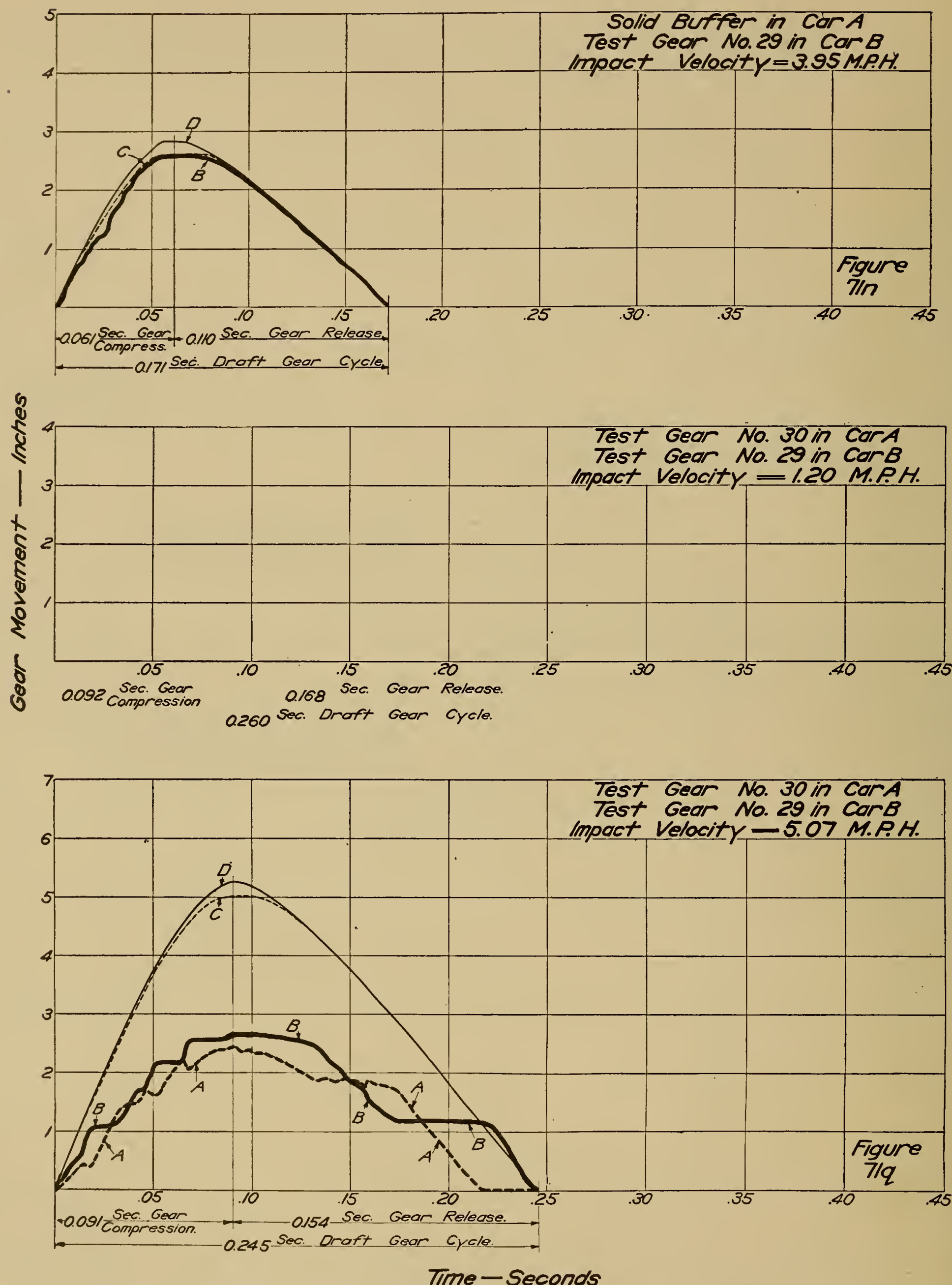
Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft gear cycle.

FIGS. 71G AND 71J—ENERGY CURVES, NATIONAL H-1 GEARS



Figs. 71k and 71m—Time-Force Curves, National H-1 Gears



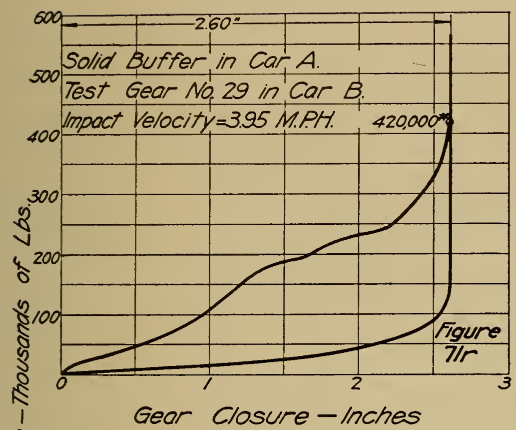
Curve D, determined from superimposed car-movement curves, represents combined draft gear movement and yield of car bodies.

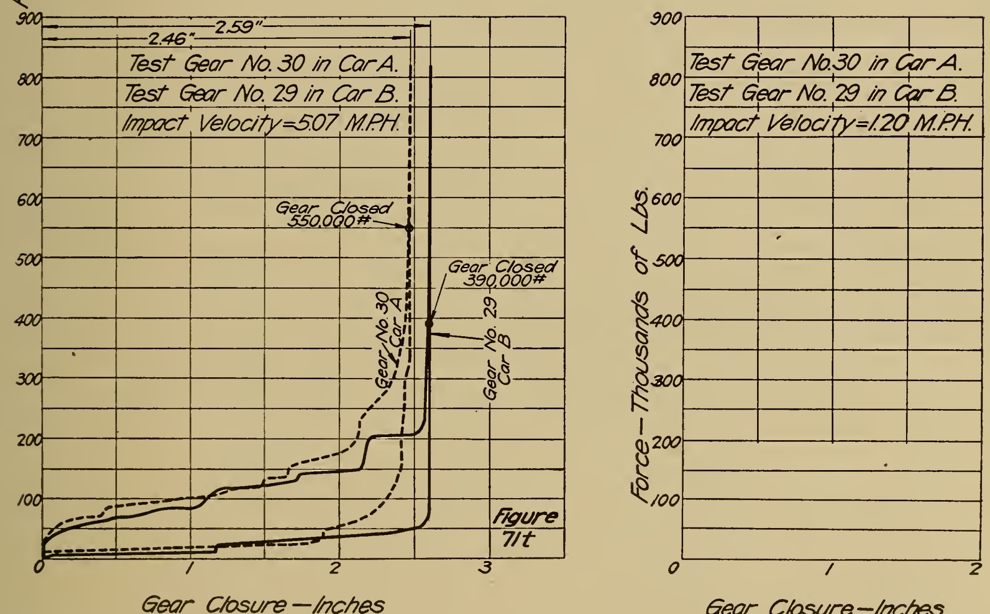
Curve C, obtained by eliminating car body yield from curve D, represents true combined movement of both gears.

Curve B, traced on small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

FIGS. 71N AND 71Q—TIME-CLOSURE CURVES, NATIONAL H-1 GEARS





Figs. 71r and 71t—Force-Closure Diagrams, National H-1 Gears

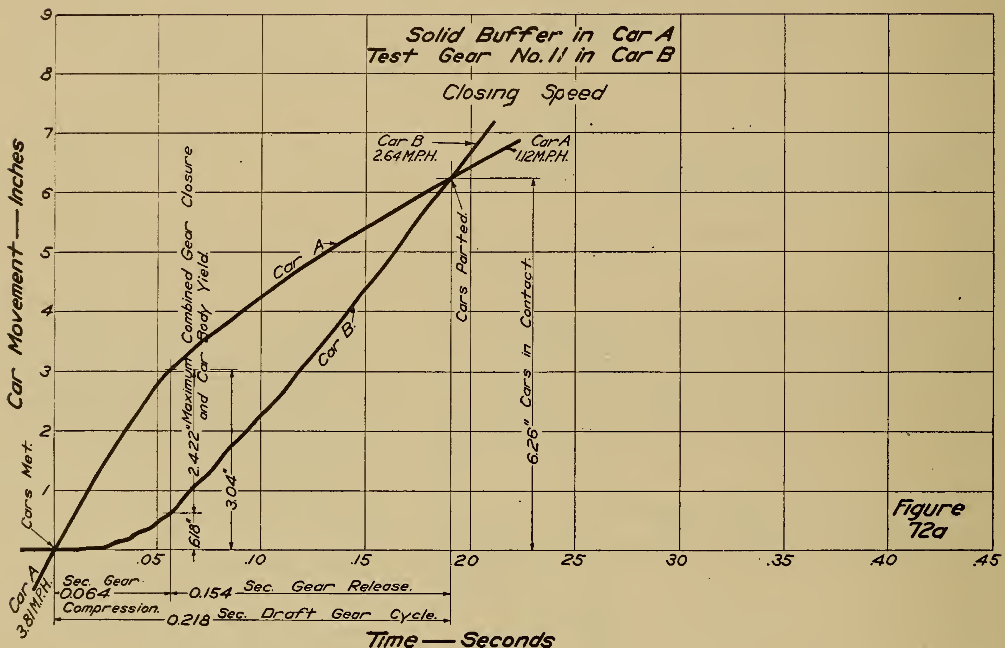
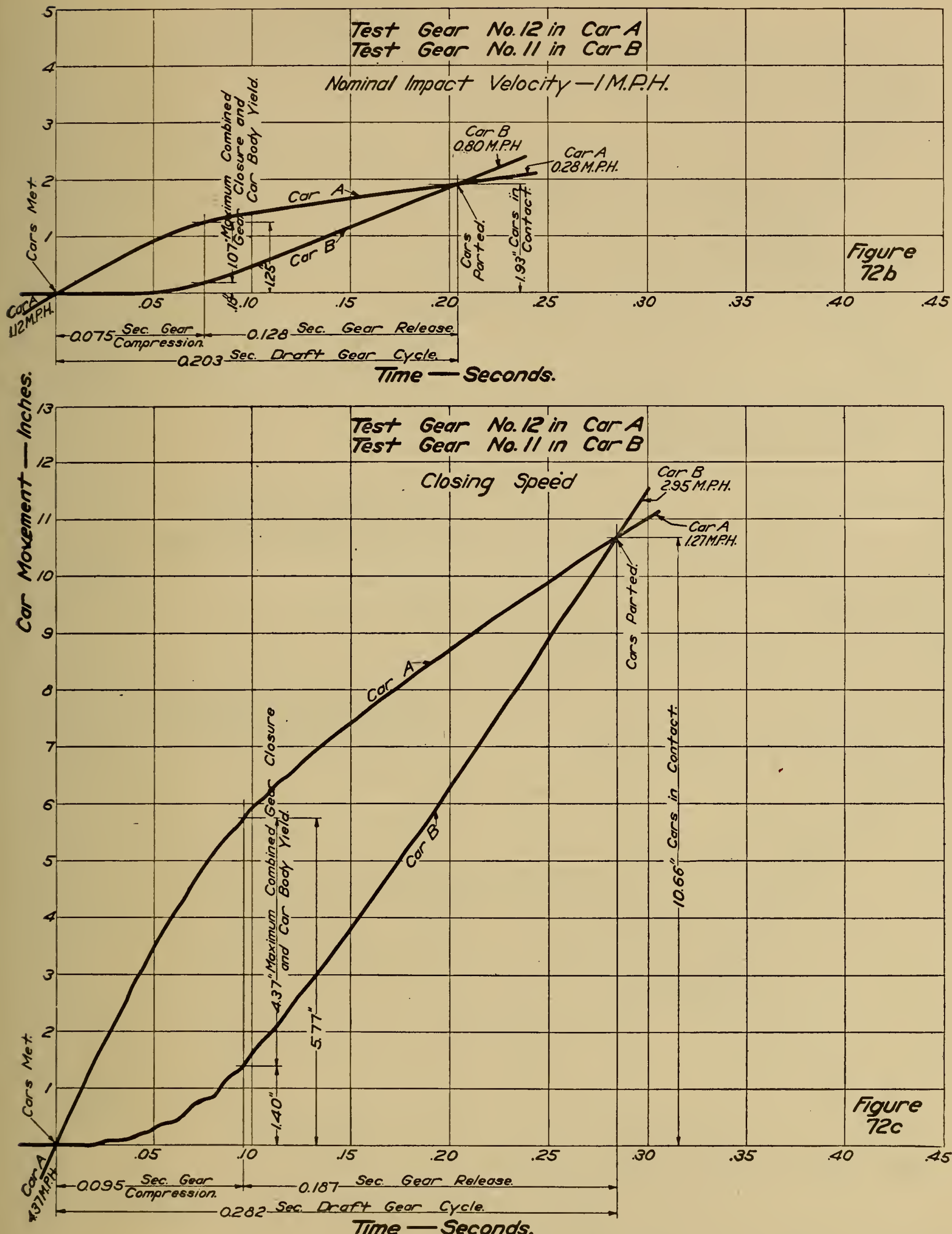
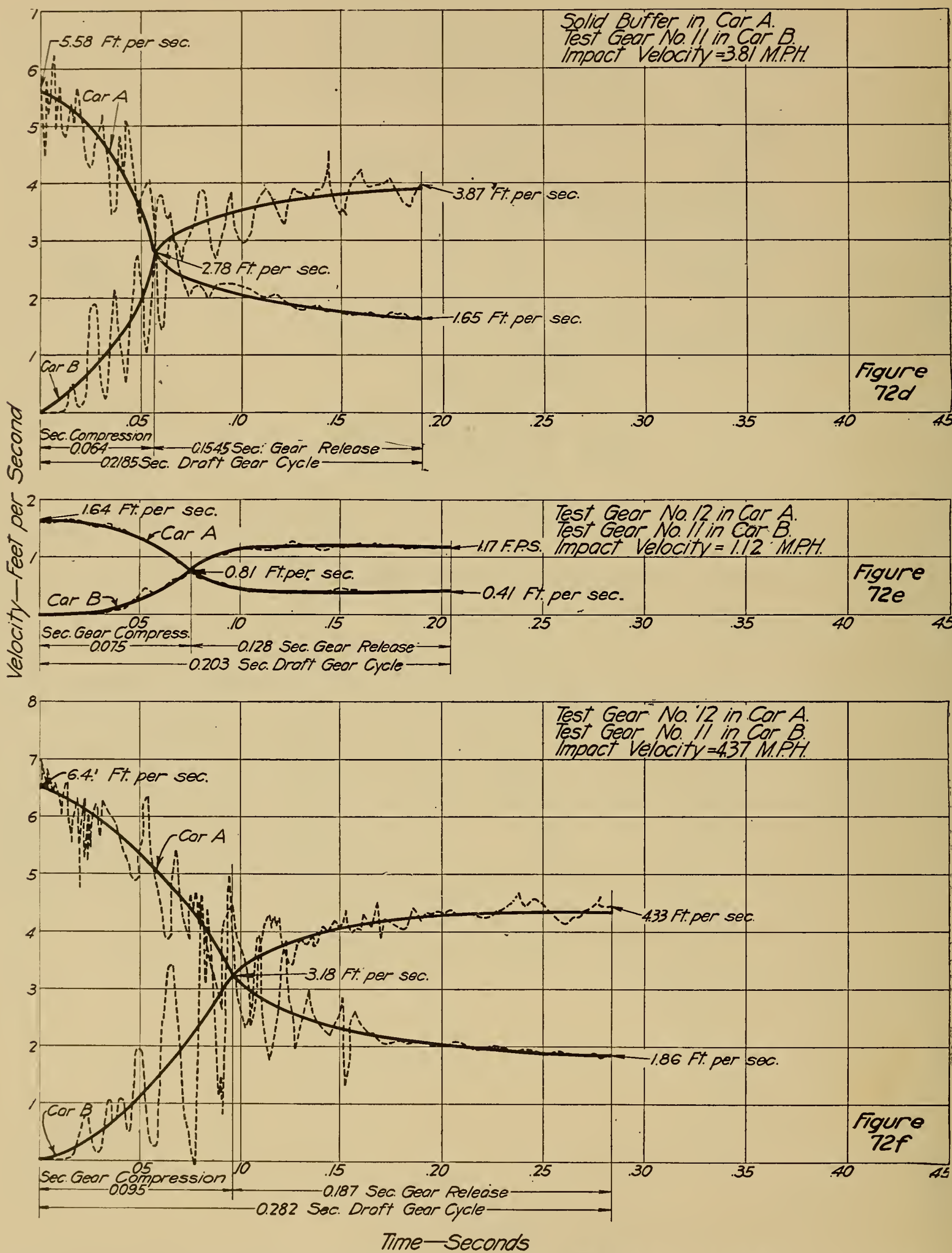


Fig. 72a—Car-Movement Curves, Superimposed. Sessions K Gears. These Curves Drawn by Cars in Test



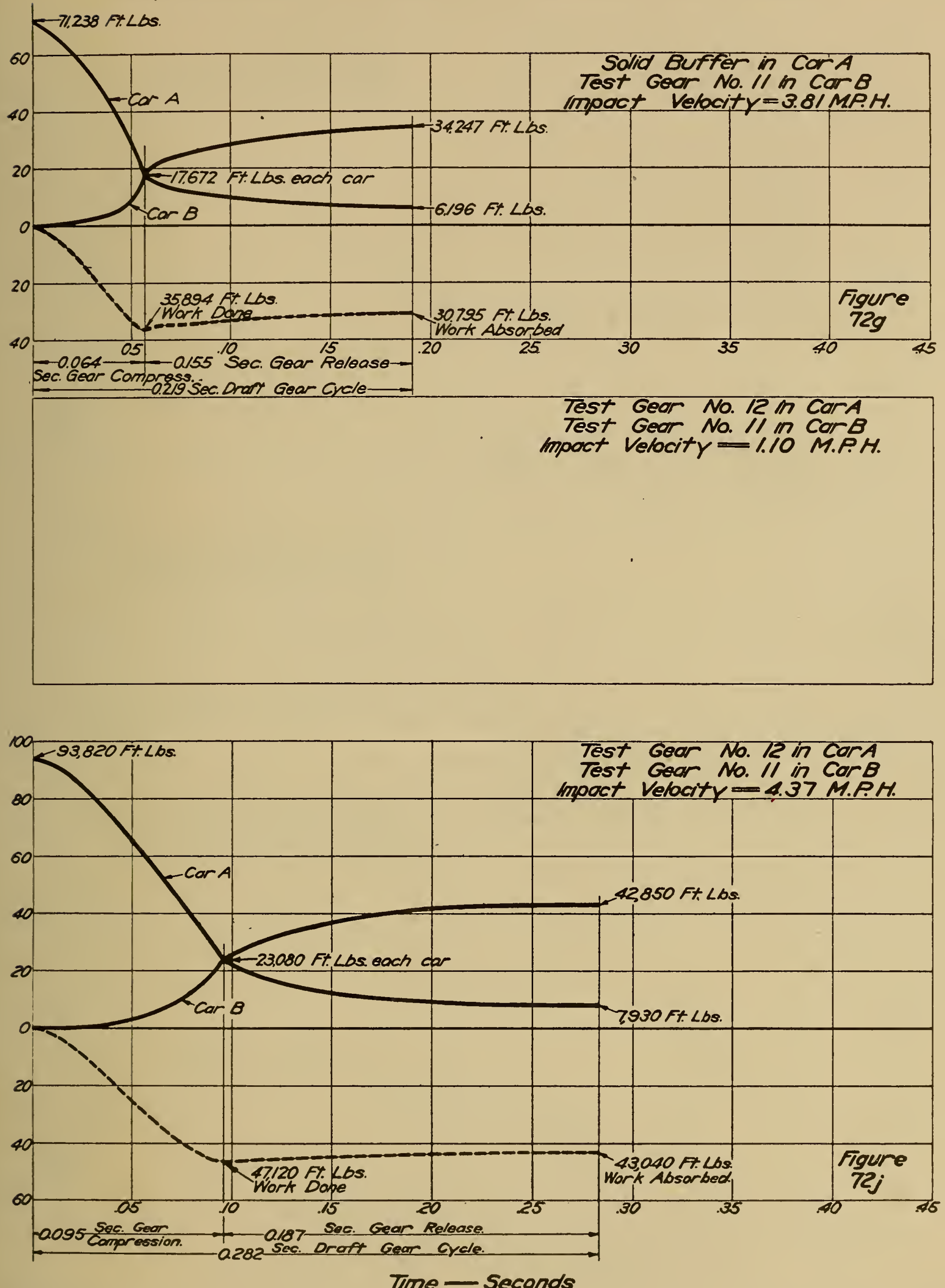
Figs. 72b and 72c—Car-Movement Curves, Superimposed. Sessions K Gears. These Curves Drawn by Cars in Tests



Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

The irregularities are due in general to vibrations of the car structure induced by draft gear action.
Full lines represent the mean velocity curves.

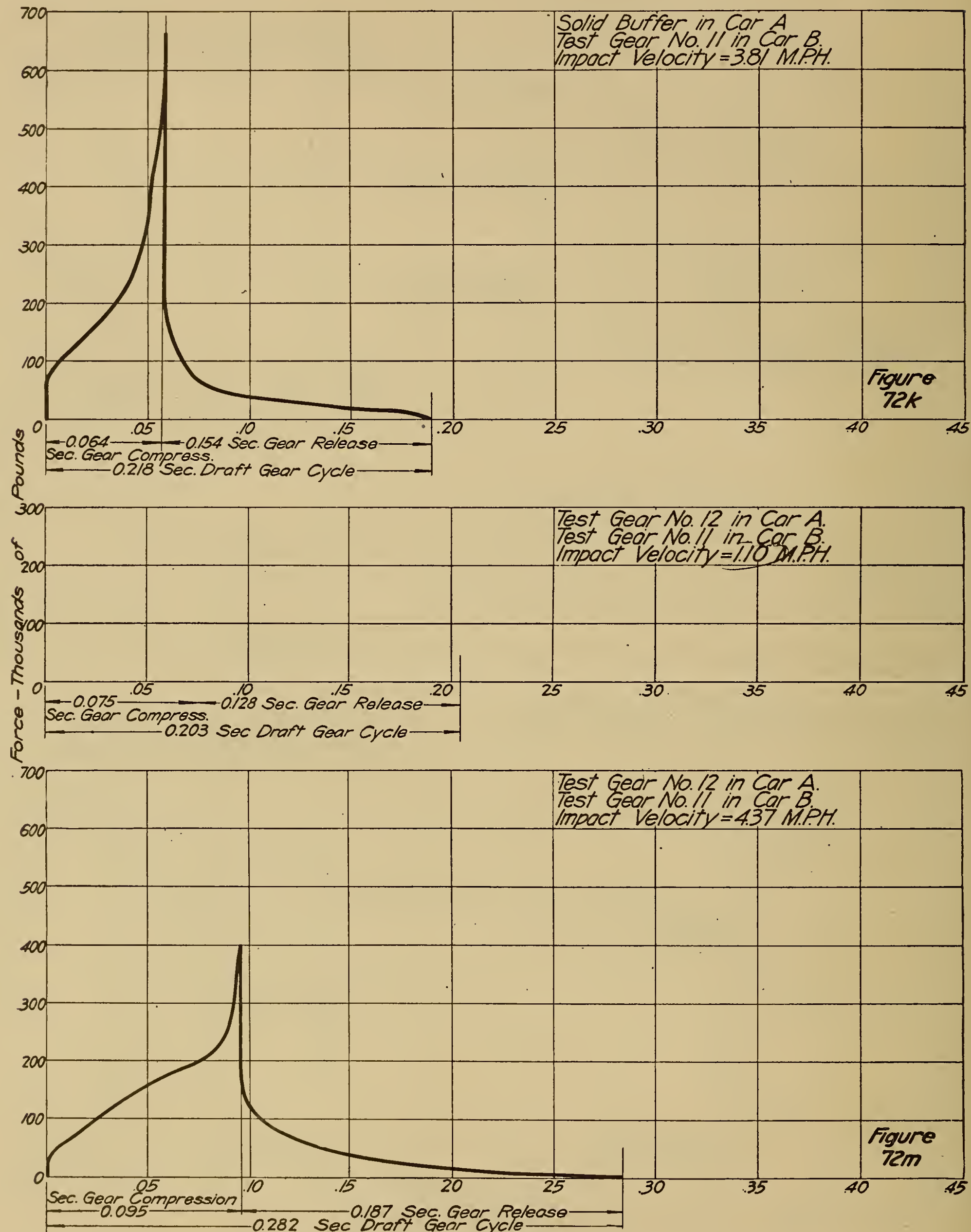
FIGS. 72D, 72E AND 72F—VELOCITY CURVES, SESSIONS K GEARS



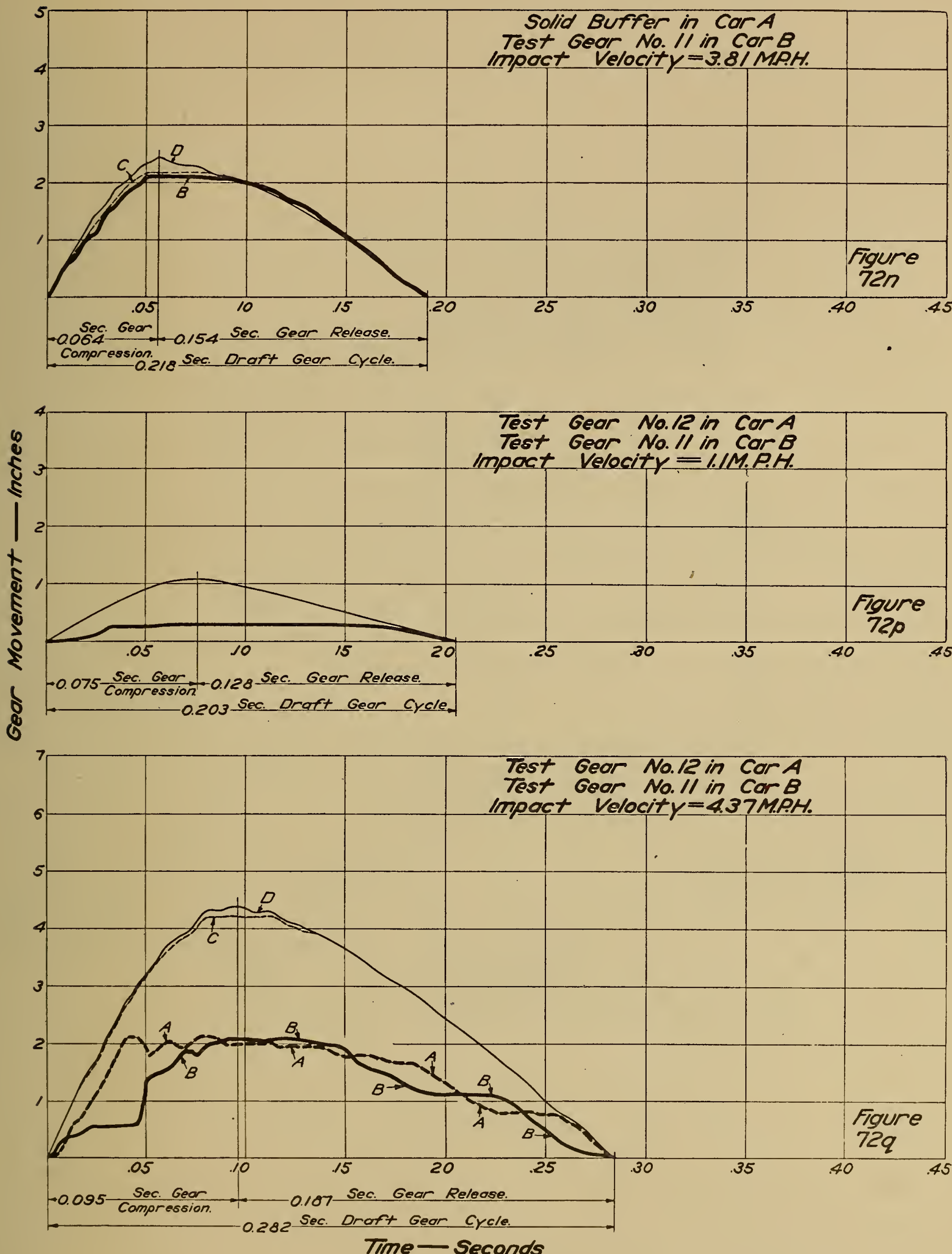
Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft gear cycle.

FIGS. 72G AND 72J—ENERGY CURVES, SESSIONS K GEARS



FIGS. 72K AND 72M—TIME-FORCE CURVES, SESSIONS K GEARS



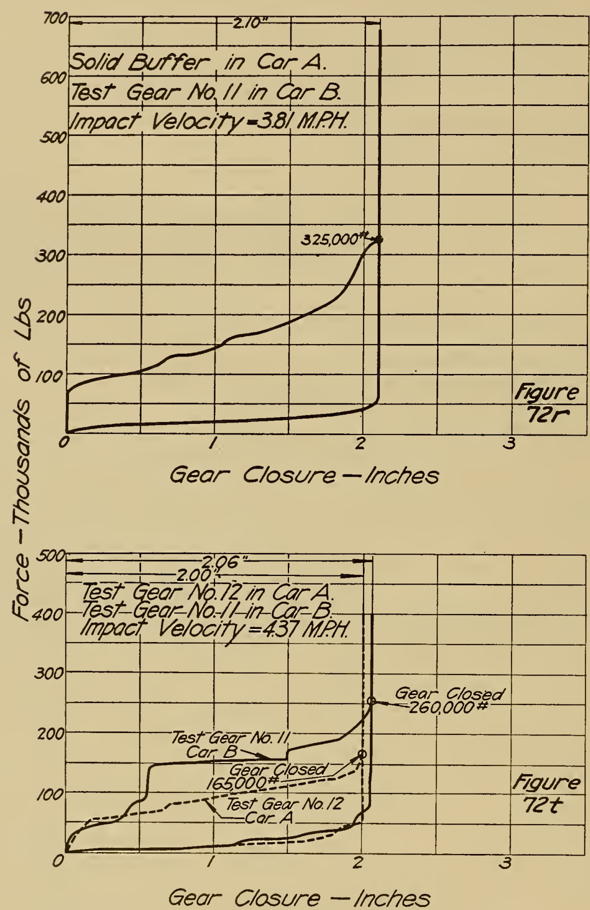
Curve D, determined from superimposed car-movement curves, represents combined draft gear movement and yield of car bodies.

Curve C, obtained by eliminating car body yield from curve D, represents true combined movement of both gears.

Curve B, traced on small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

FIGS. 72N, 72P AND 72Q—TIME-CLOSURE CURVES, SESSIONS K GEARS



Figs. 72r and 72t—Force-Closure Diagrams
Sessions K Gears

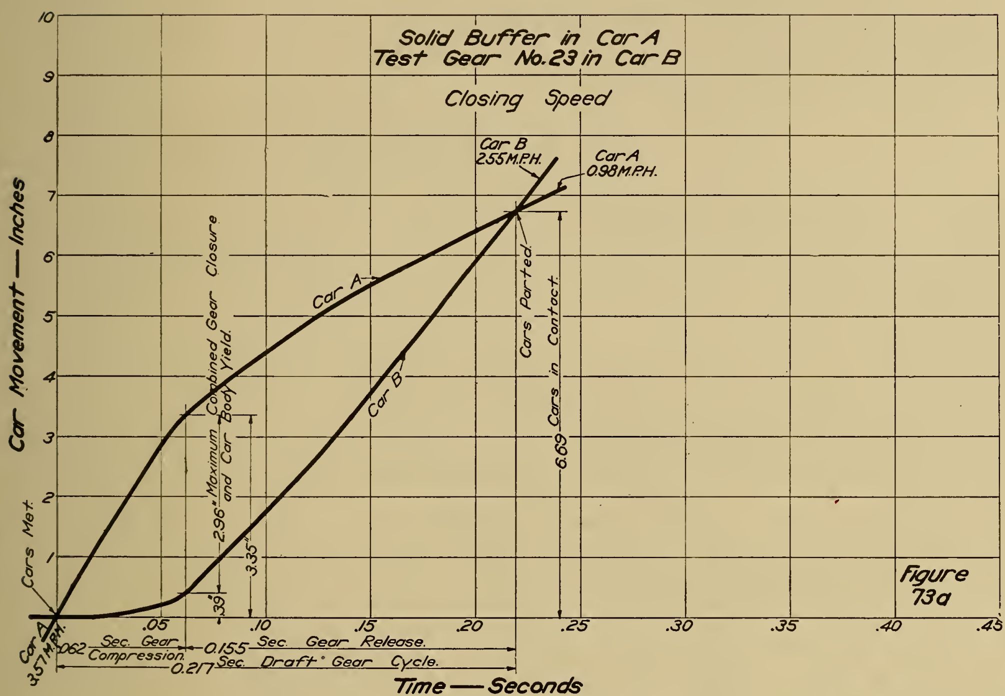
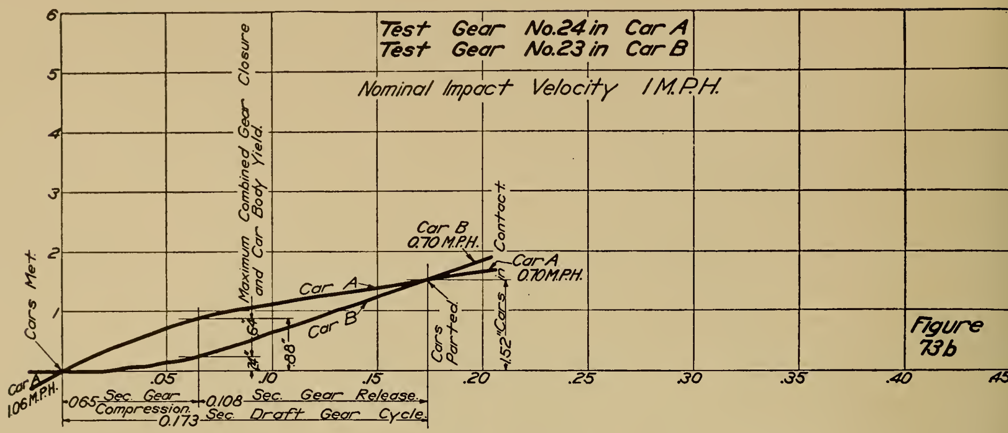
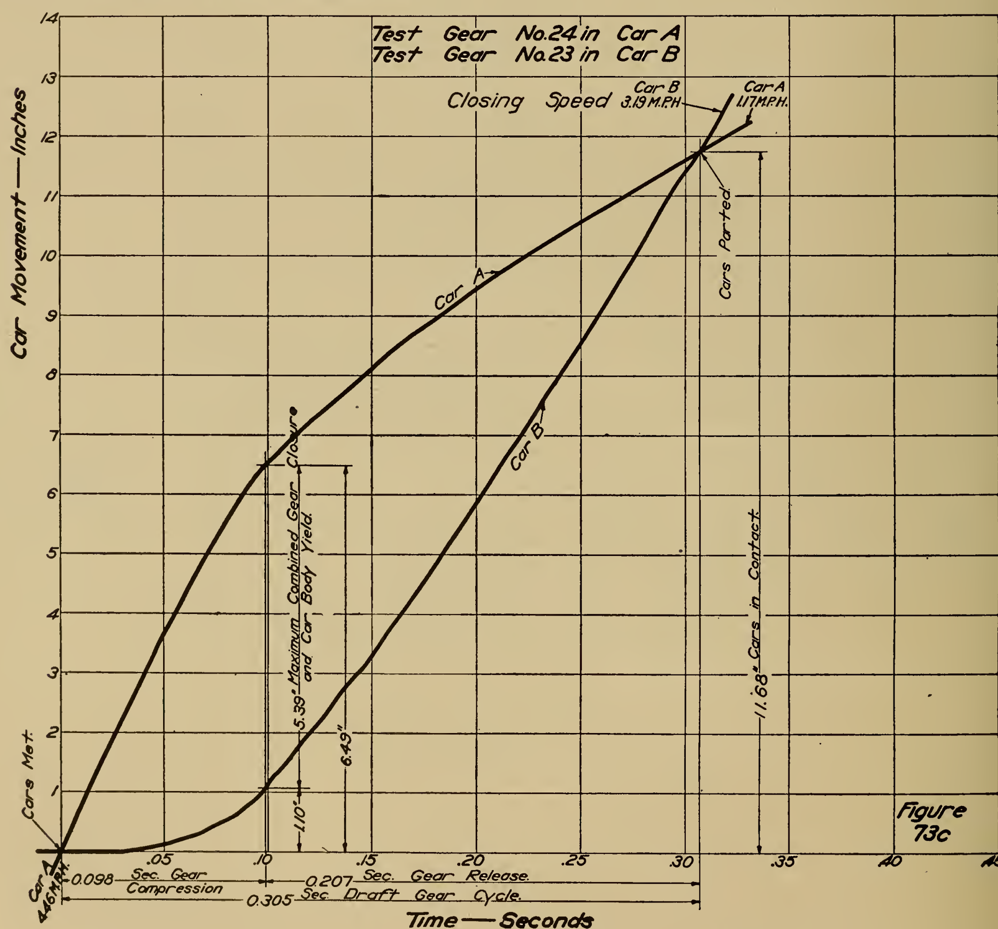
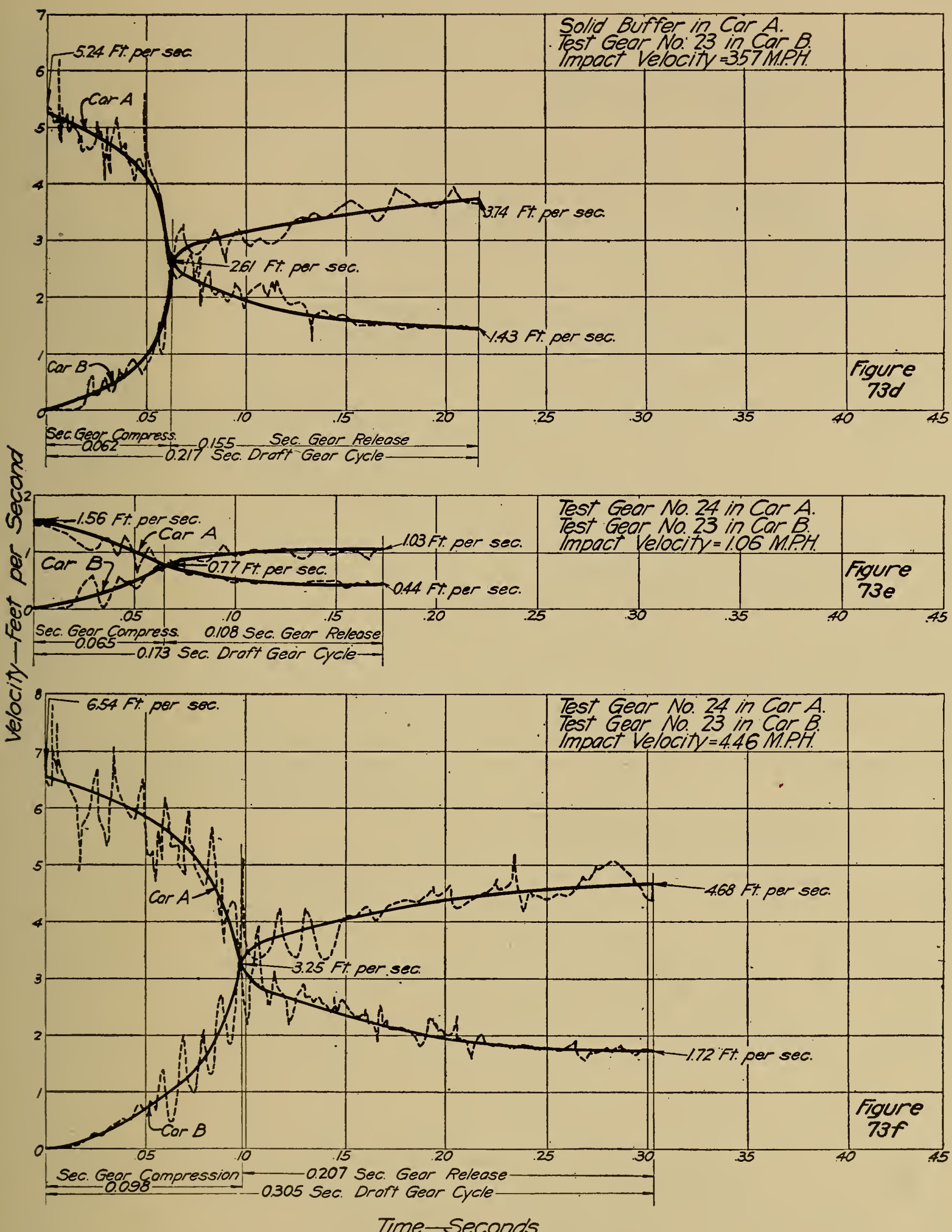


Fig. 73a—Car-Movement Curves, Superimposed. Miner A-18-S Gears
These Curves Drawn by Cars in Test





Figs. 73b and 73c—Car-Movement Curves, Superimposed. Miner A-18-S Gears
These Curves Drawn by Cars in Test

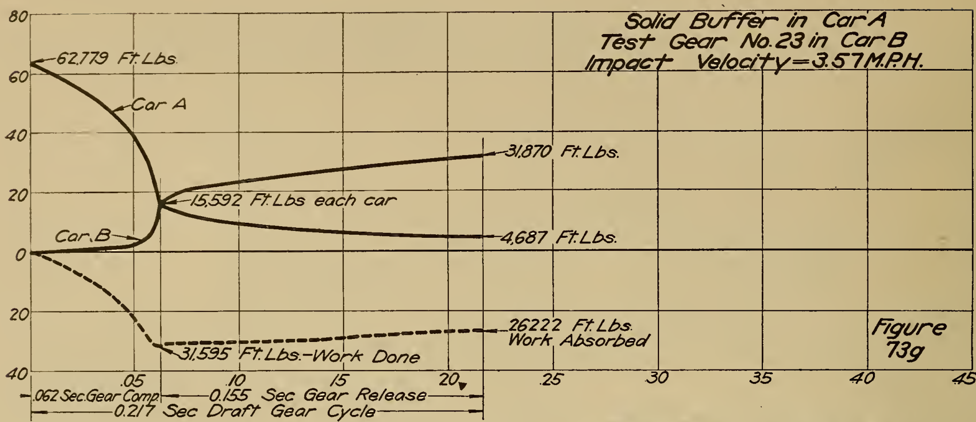


Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

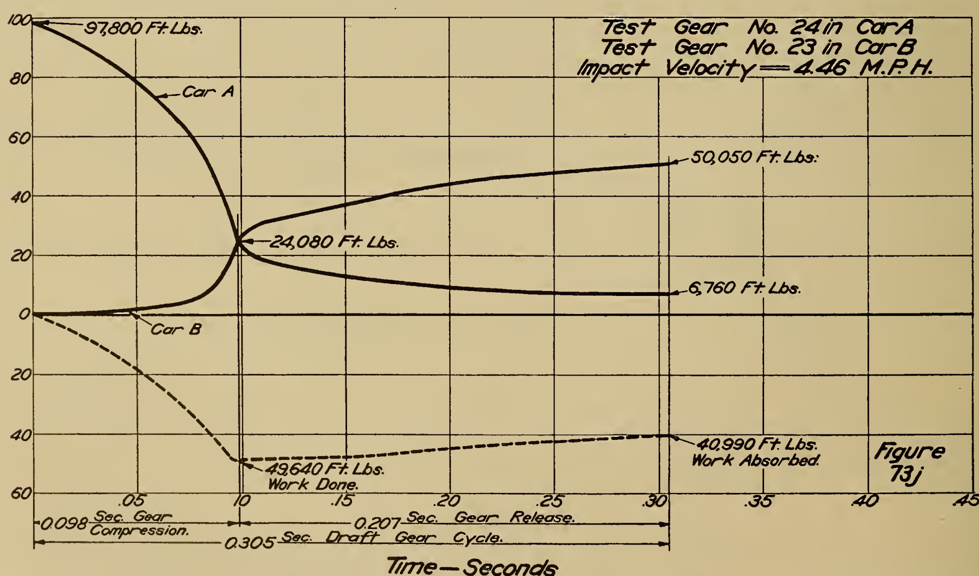
The irregularities are due in general to vibrations of the car structure induced by draft gear action.

Full lines represent the mean velocity curves.

FIGS. 73D, 73E AND 73F—VELOCITY CURVES, MINER A-18-S GEARS



Test Gear No. 24 in Car A
Test Gear No. 23 in Car B
Impact Velocity = 1.05 M.P.H.



Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft gear cycle.

FIGS. 73G AND 73J—ENERGY CURVES, MINER A-18-S GEARS

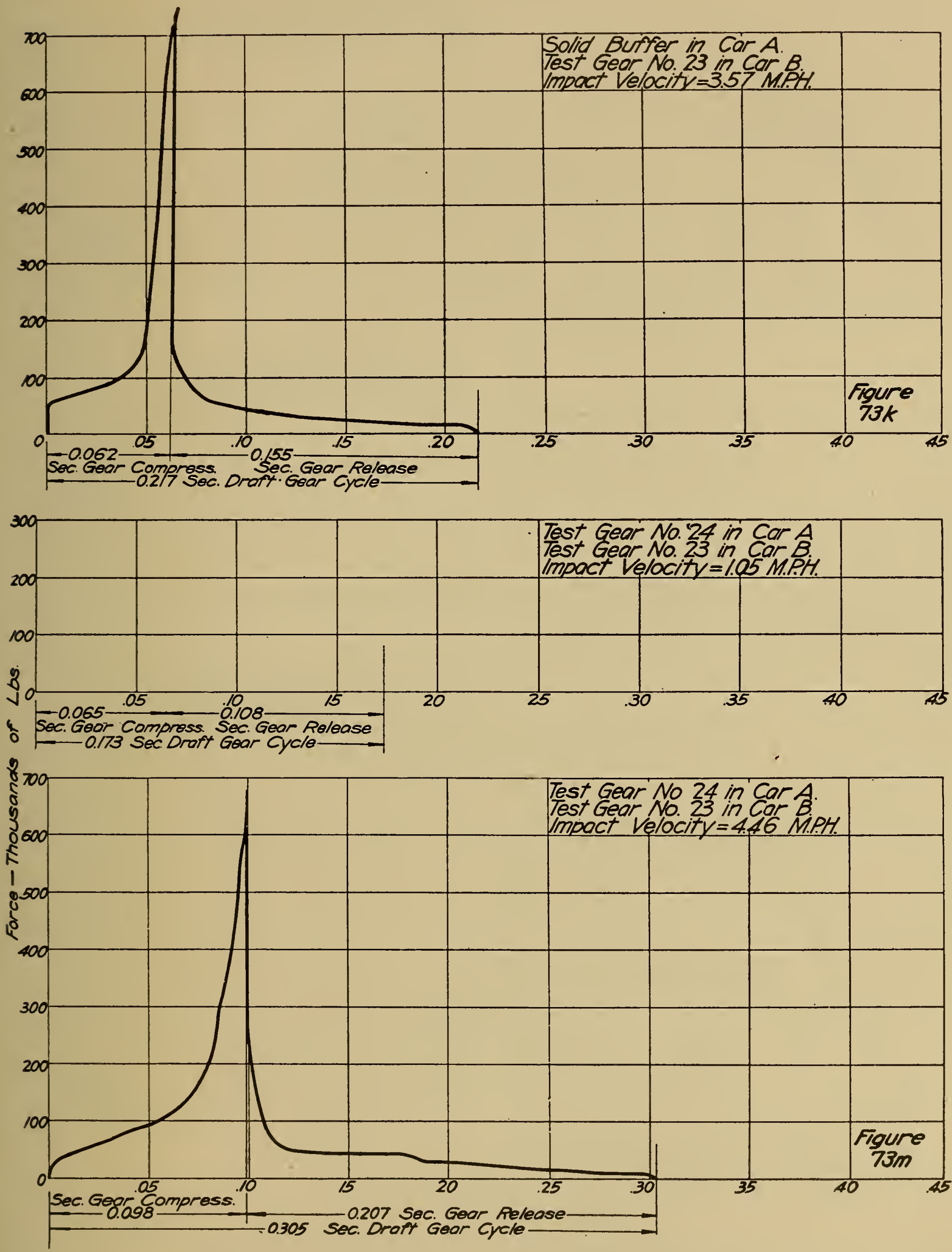
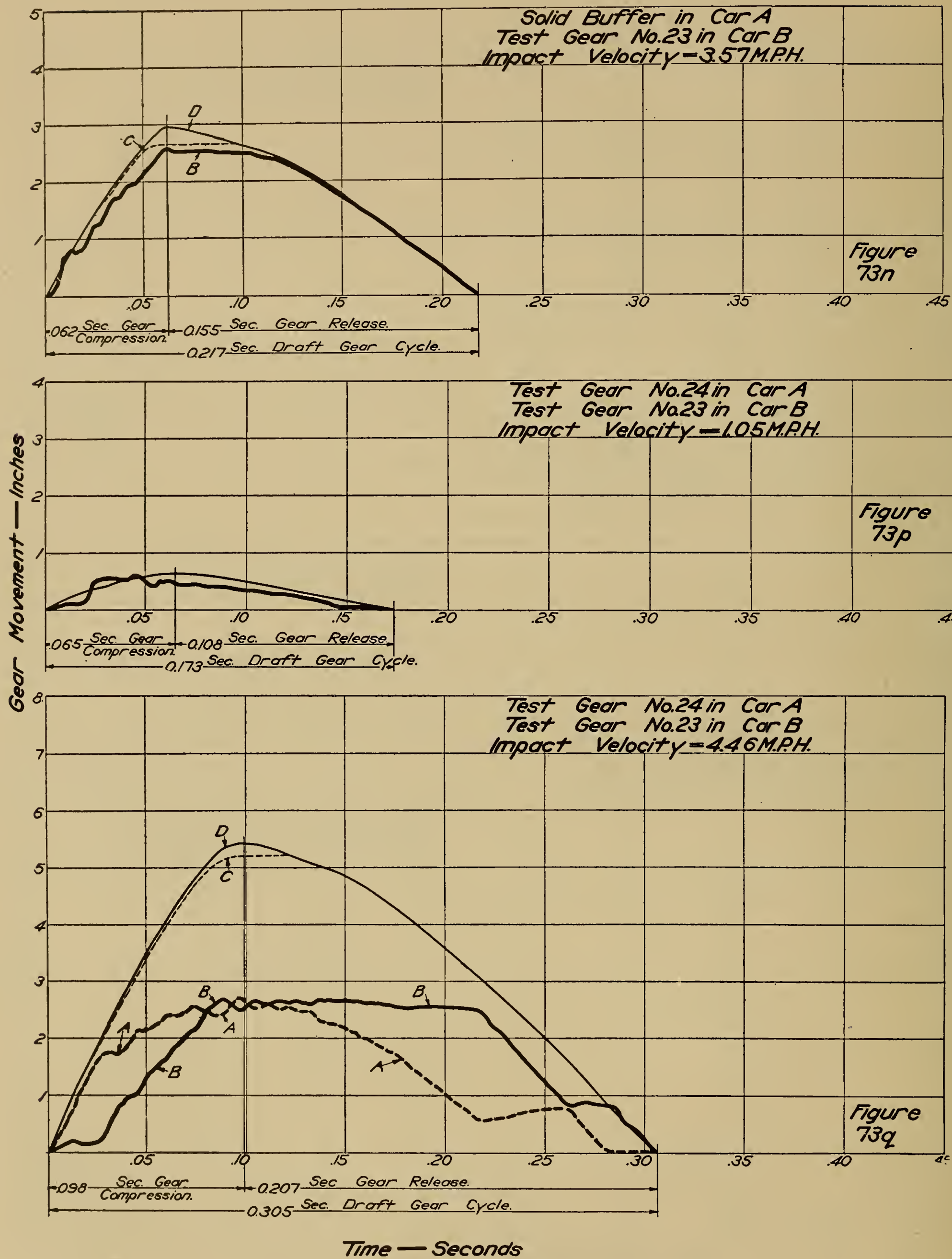


Fig. 73k and 73m—Time-Force Curves, Miner A-18-S Gears



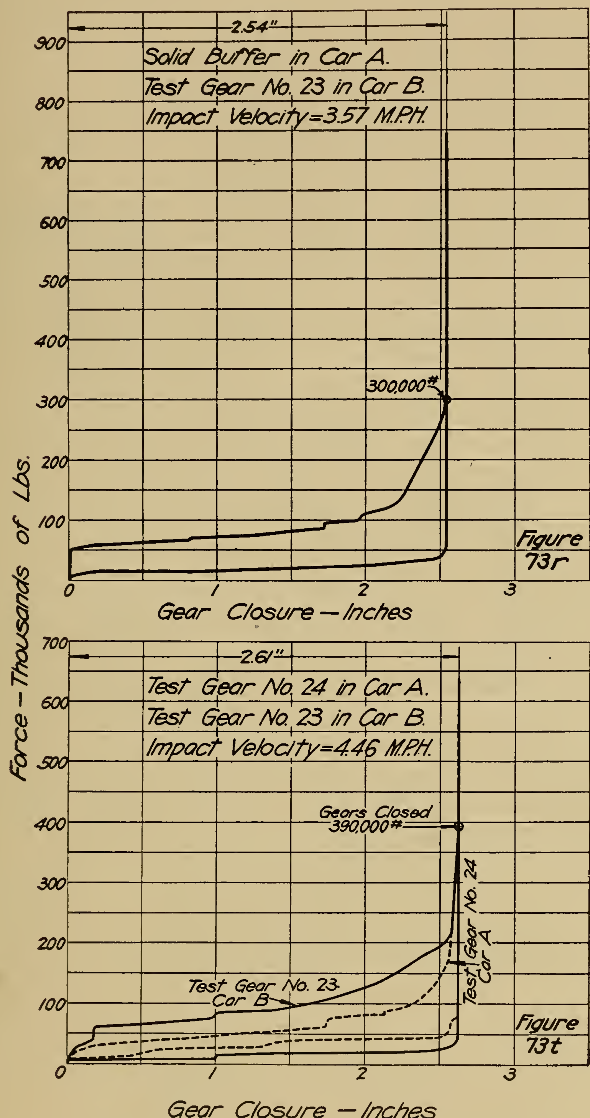
Curve D, determined from superimposed car-movement curves, represents combined draft gear movement and yield of car bodies.

Curve C, obtained by eliminating car body yield from Curve D, represents true combined movement of both gears.

Curve B, traced on small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

FIGS. 73N, 73P AND 73Q—TIME-CLOSURE CURVES, MINER A-18-S GEARS



FIGS. 73R AND 73T—FORCE-CLOSURE DIAGRAMS
MINER A-18-S GEARS

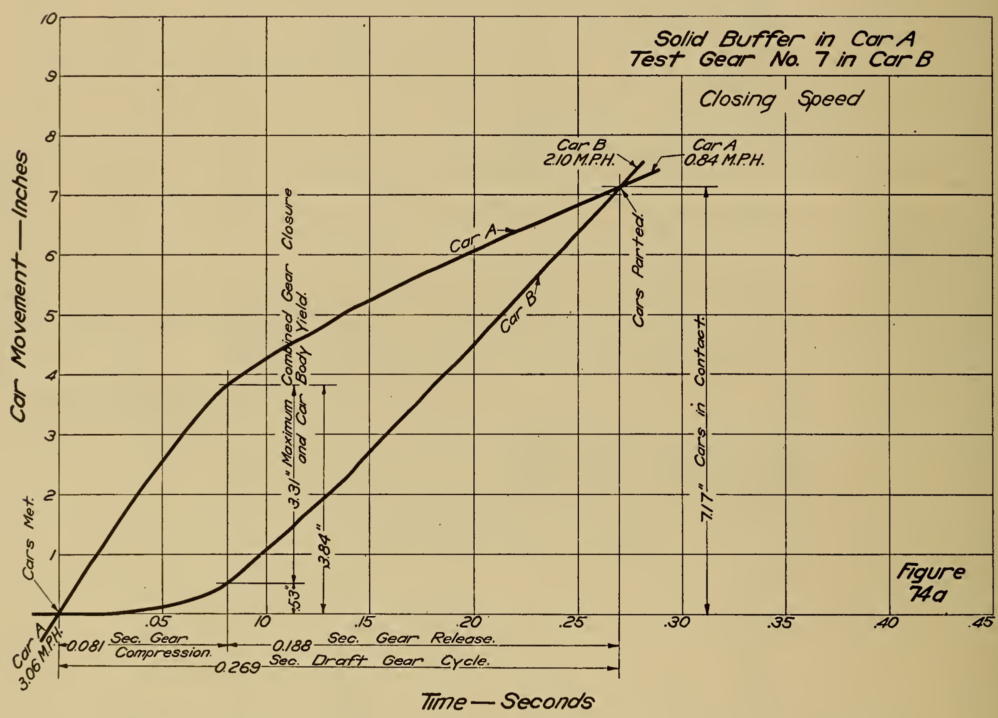
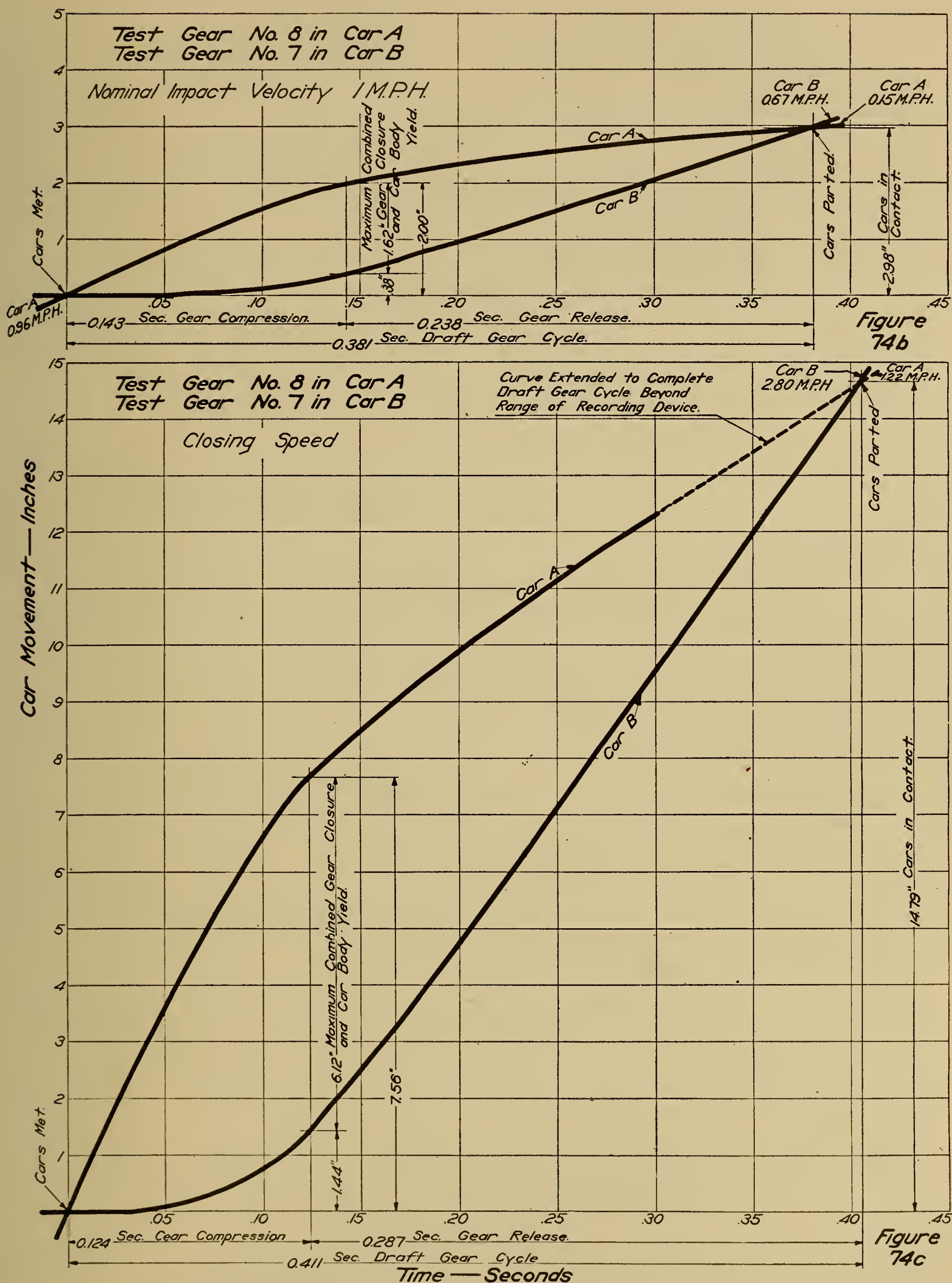
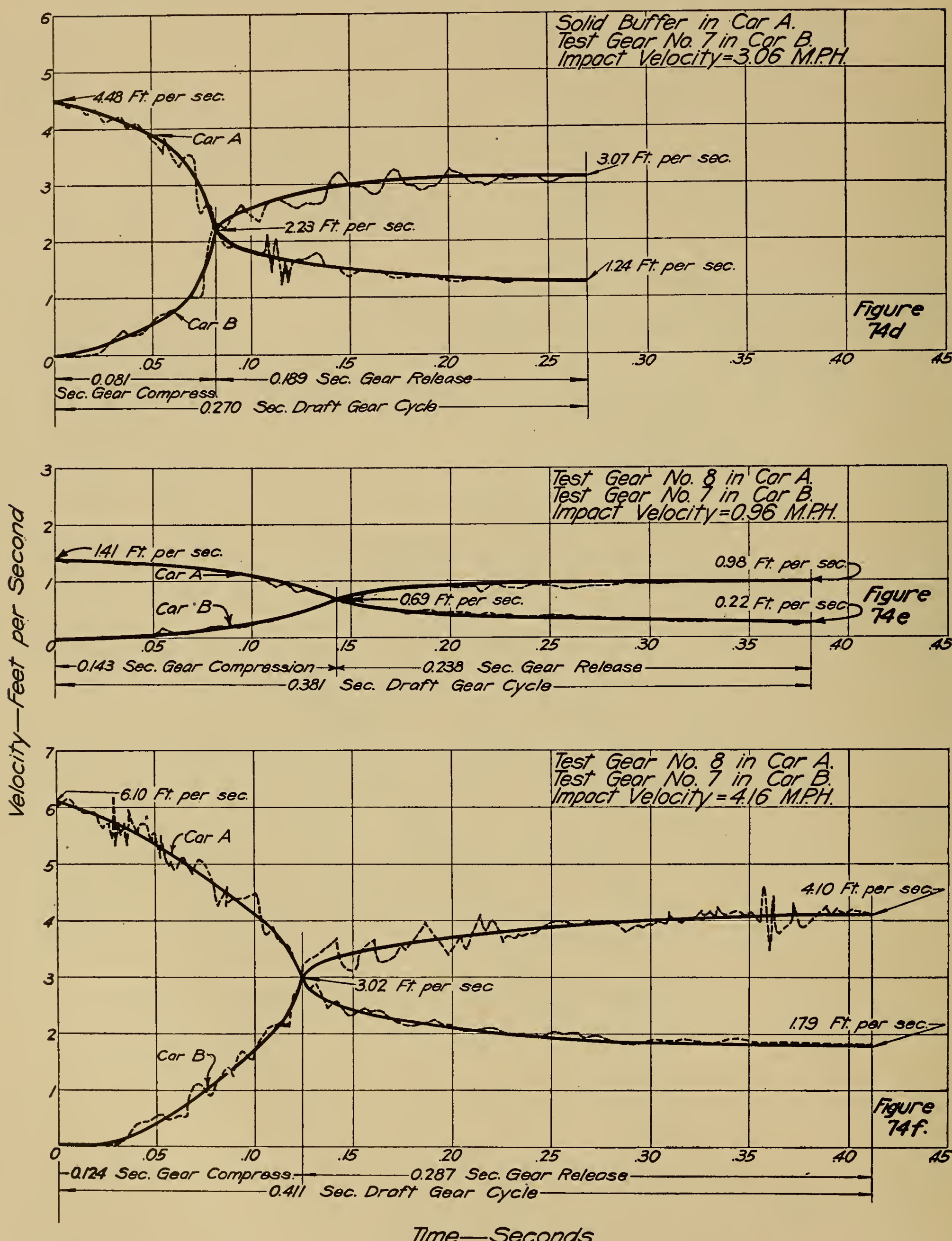


FIG. 74A—CAR-MOVEMENT CURVES, SUPERIMPOSED. WESTINGHOUSE NA-1 GEARS. THESE CURVES DRAWN BY CARS IN TEST



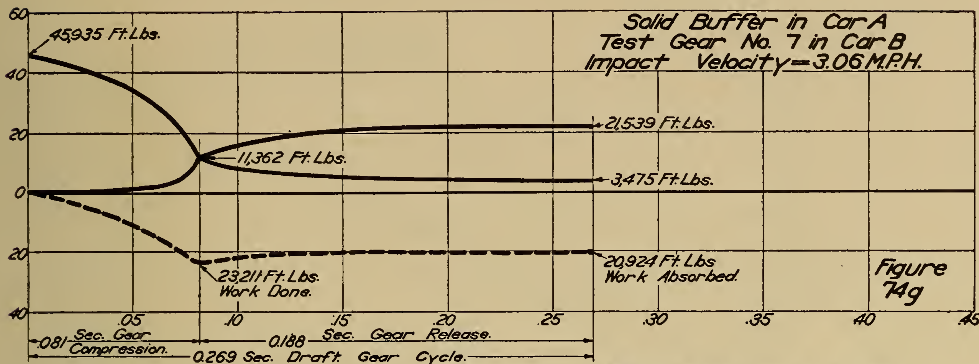
FIGS. 74B AND 74C—CAR-MOVEMENT CURVES, SUPERIMPOSED. WESTINGHOUSE NA-1 GEARS
THESE CURVES DRAWN BY CARS IN TEST

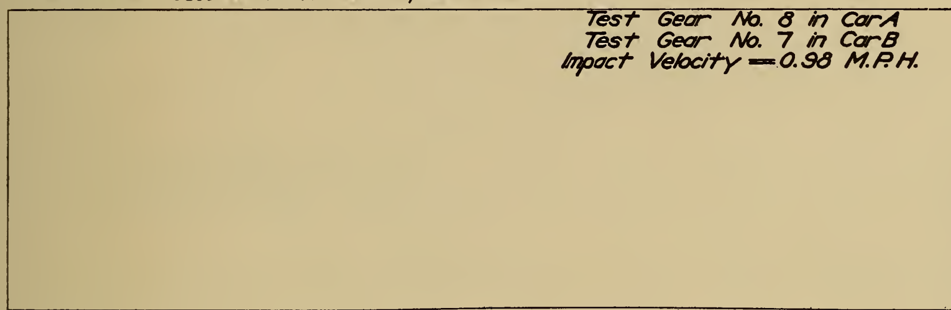


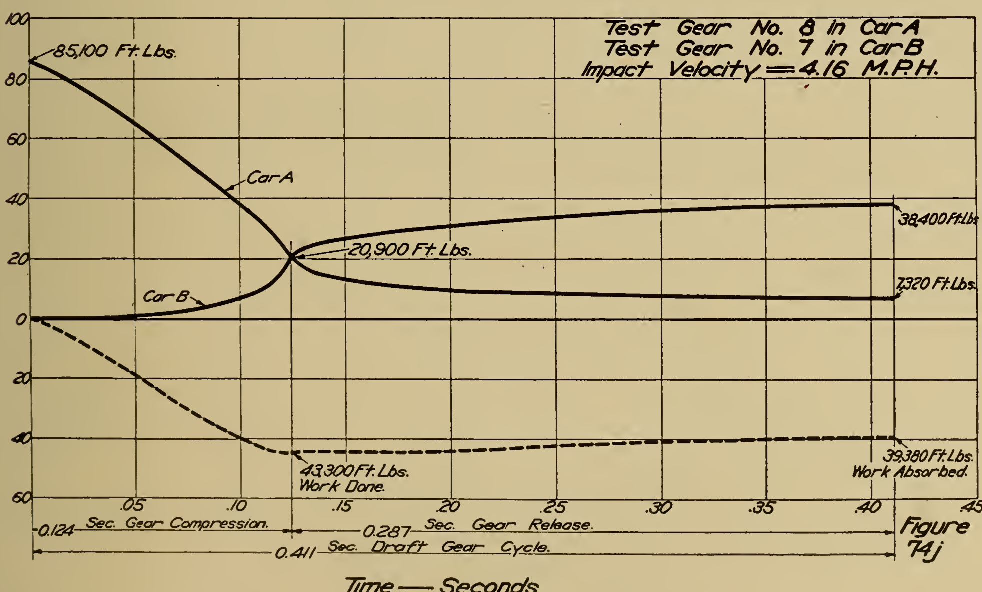
Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

The irregularities are due in general to vibrations of the car structure induced by draft gear action.
Full lines represent the mean velocity curves.

Figs. 74d, 74e and 74f—Velocity Curves, Westinghouse NA-1 Gears



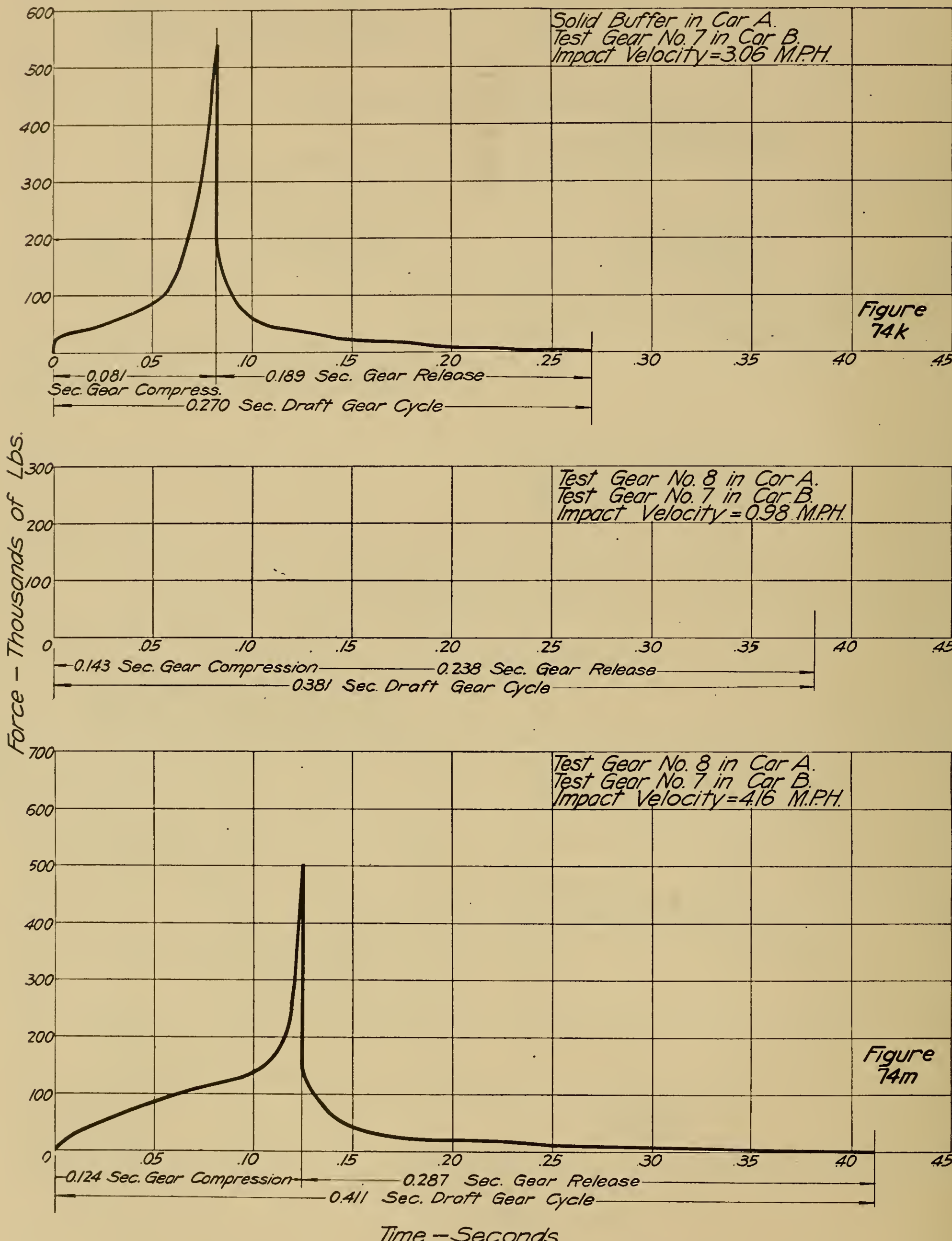




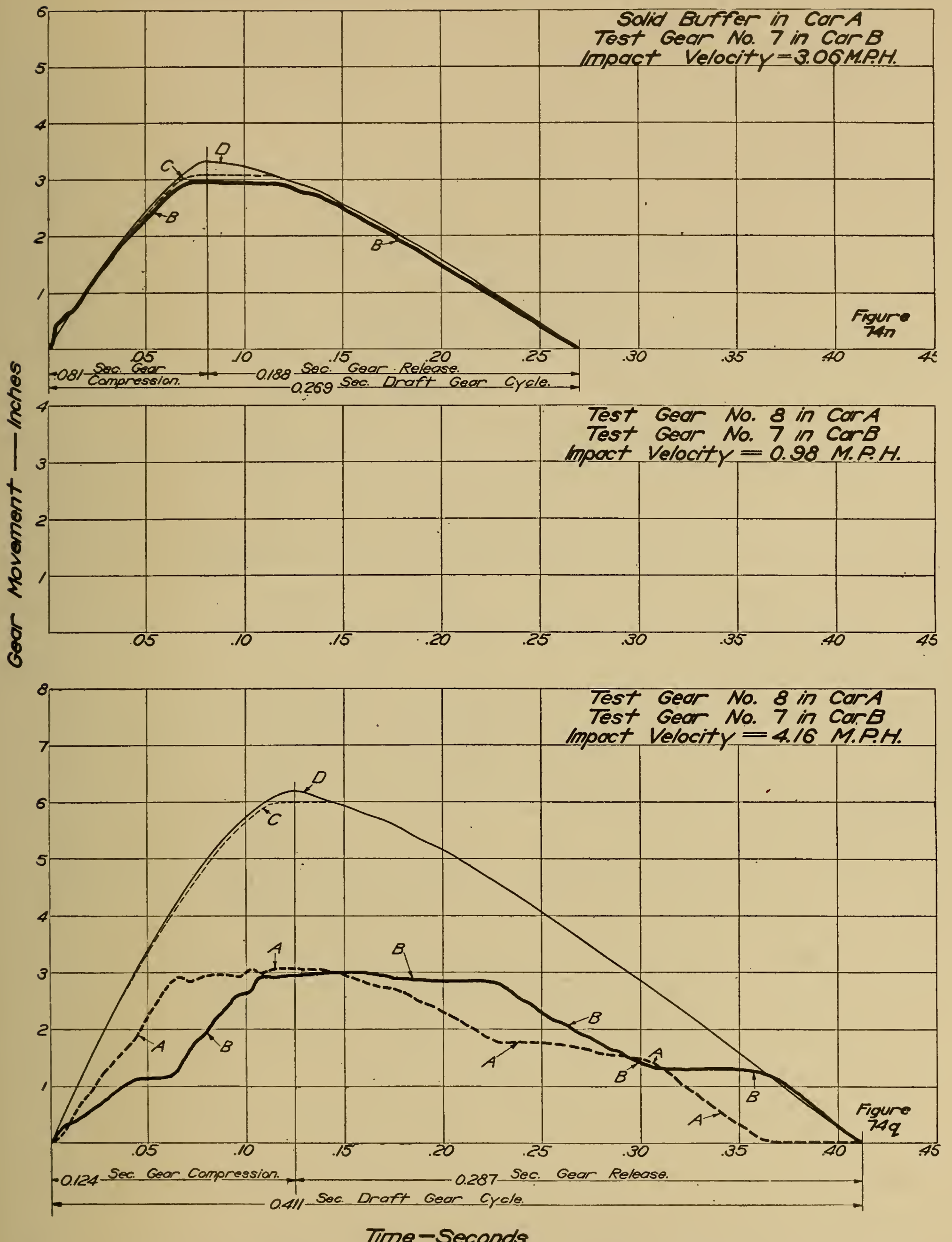
Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft gear cycle.

FIGS. 74G AND 74J—ENERGY CURVES, WESTINGHOUSE NA-1 GEARS



Figs. 74k and 74m—Time-Force Curves, Westinghouse NA-1 Gears



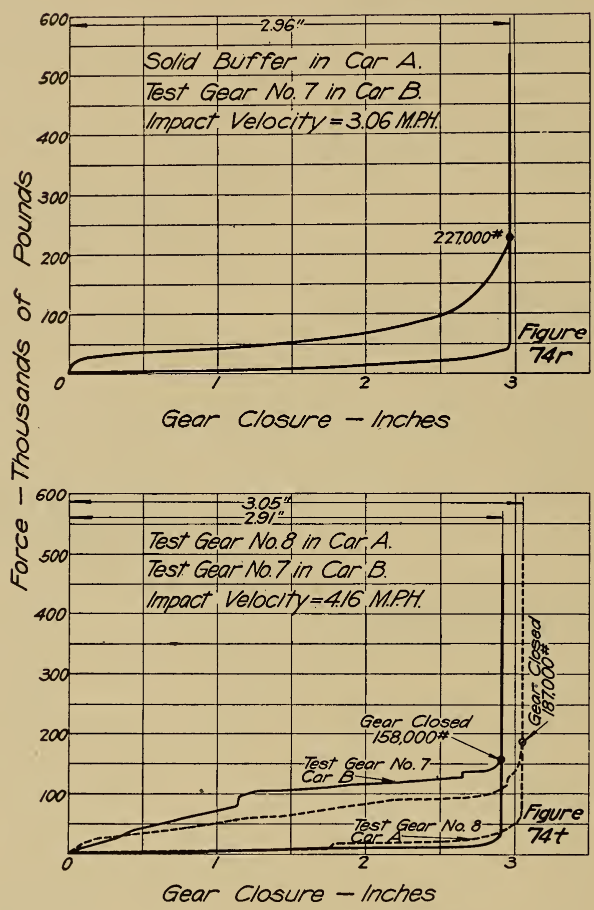
Curve D, determined from superimposed car-movement curves, represents combined draft-gear movement and yield of car bodies.

Curve C, obtained by eliminating car-body yield from curve D, represents true combined movement of both gears.

Curve B, traced on small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

FIGS. 74N AND 74Q—TIME-CLOSURE CURVES, WESTINGHOUSE NA-1 GEARS



FIGS. 74R AND 74T—FORCE-CLOSURE DIAGRAMS, WESTINGHOUSE NA-1 GEARS

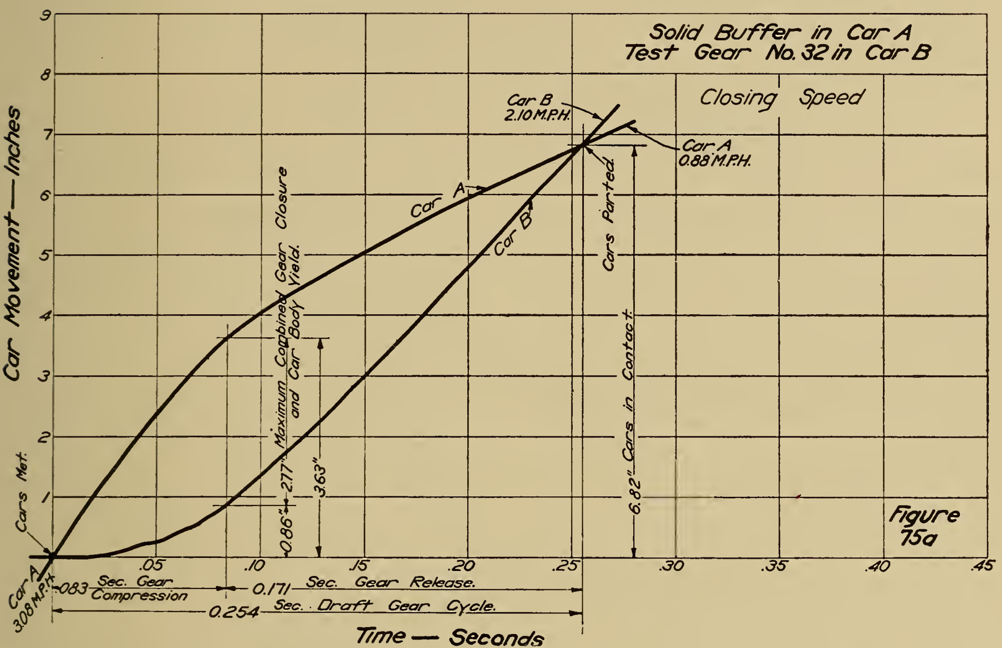
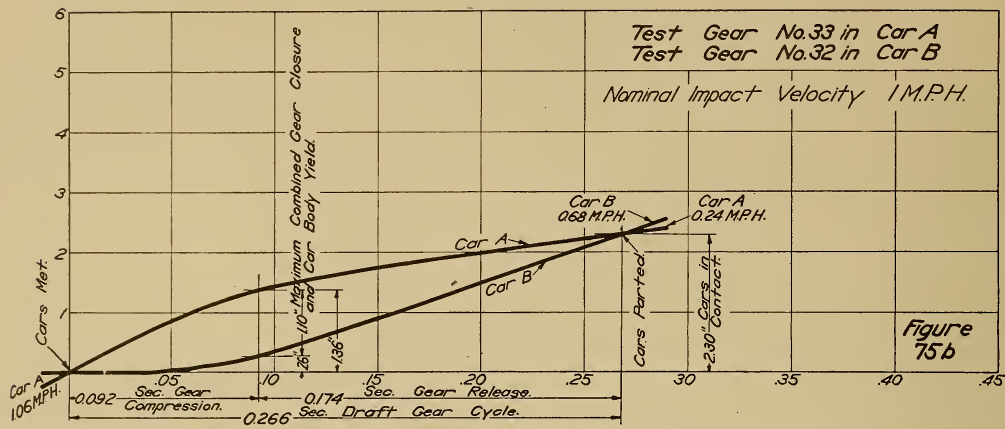
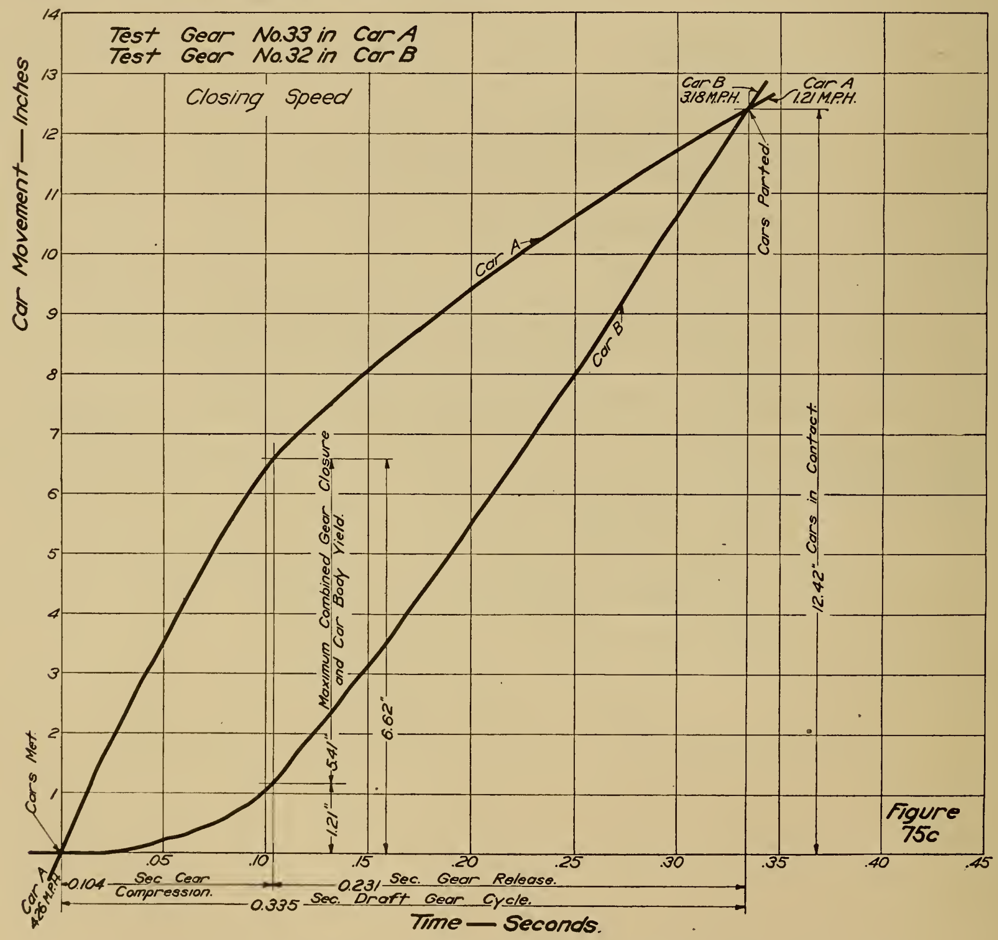
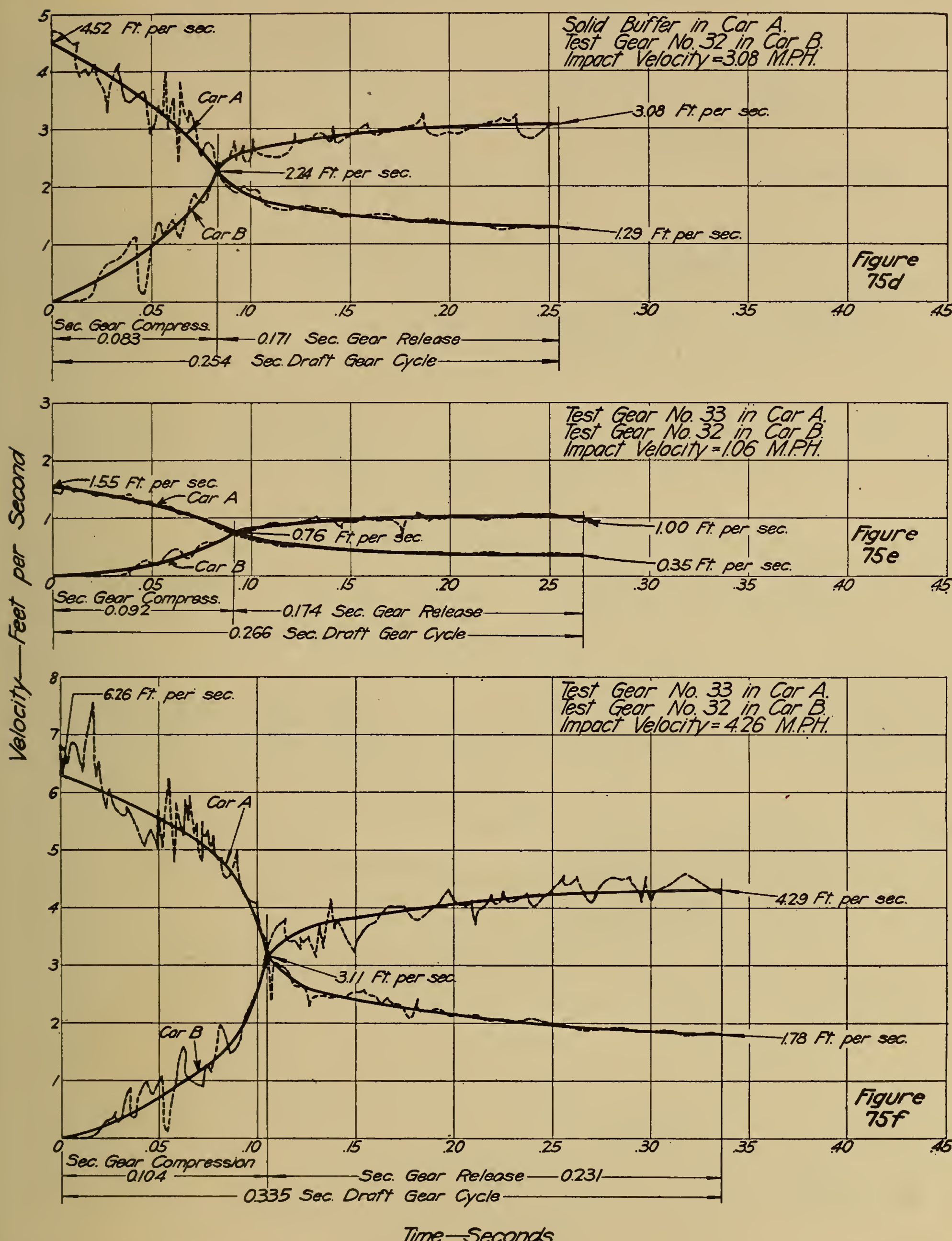


FIG. 75A—CAR-MOVEMENT CURVES, SUPERIMPOSED. NATIONAL M-1 GEARS
THESE CURVES DRAWN BY CARS IN TEST





Figs. 75b and 75c—Car-Movement Curves, Superimposed. National M-1 Gears
These Curves Drawn by Cars in Test

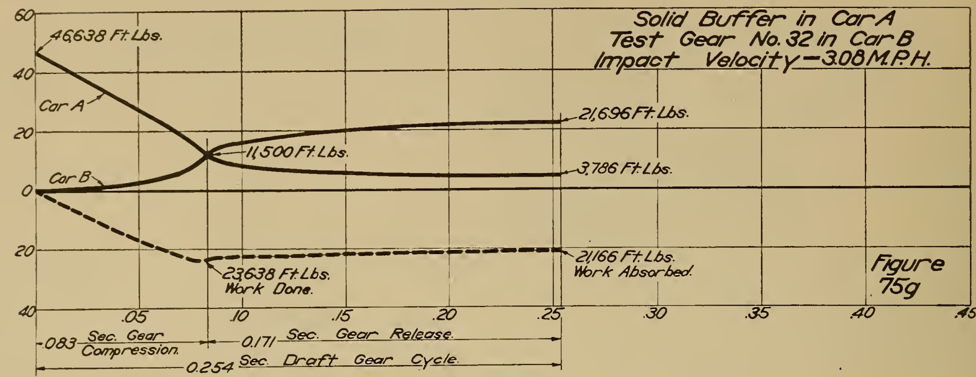


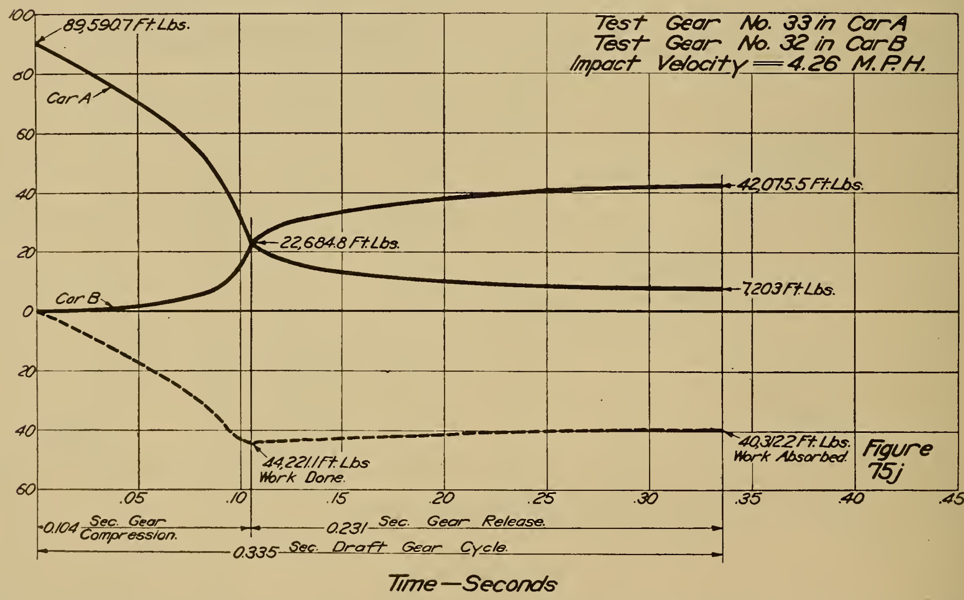
Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

The irregularities are due in general to vibrations of the car structure induced by draft-gear action.

Full lines represent the mean velocity curves.

FIGS. 75D, 75E AND 75F—VELOCITY CURVES, NATIONAL M-1 GEARS

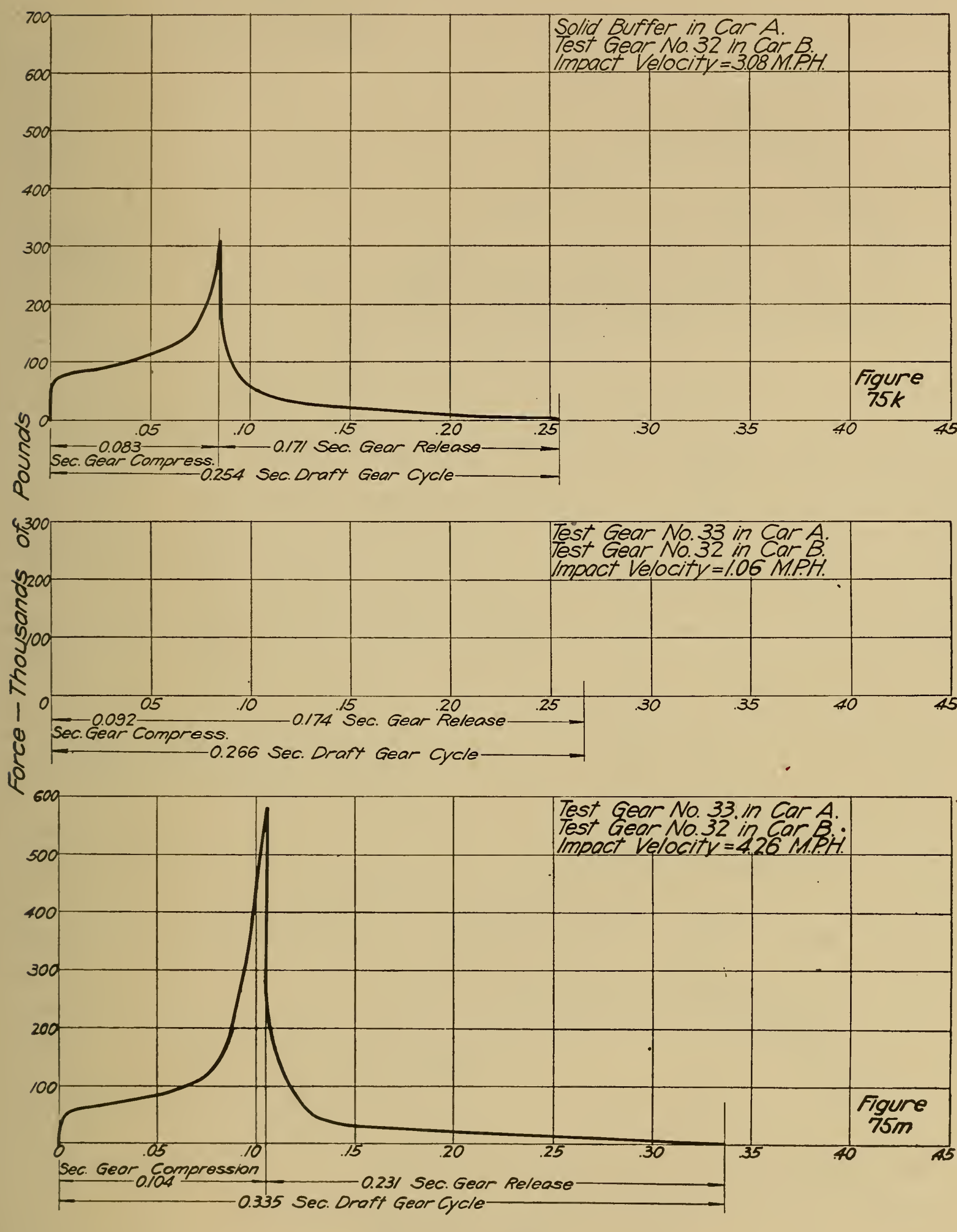




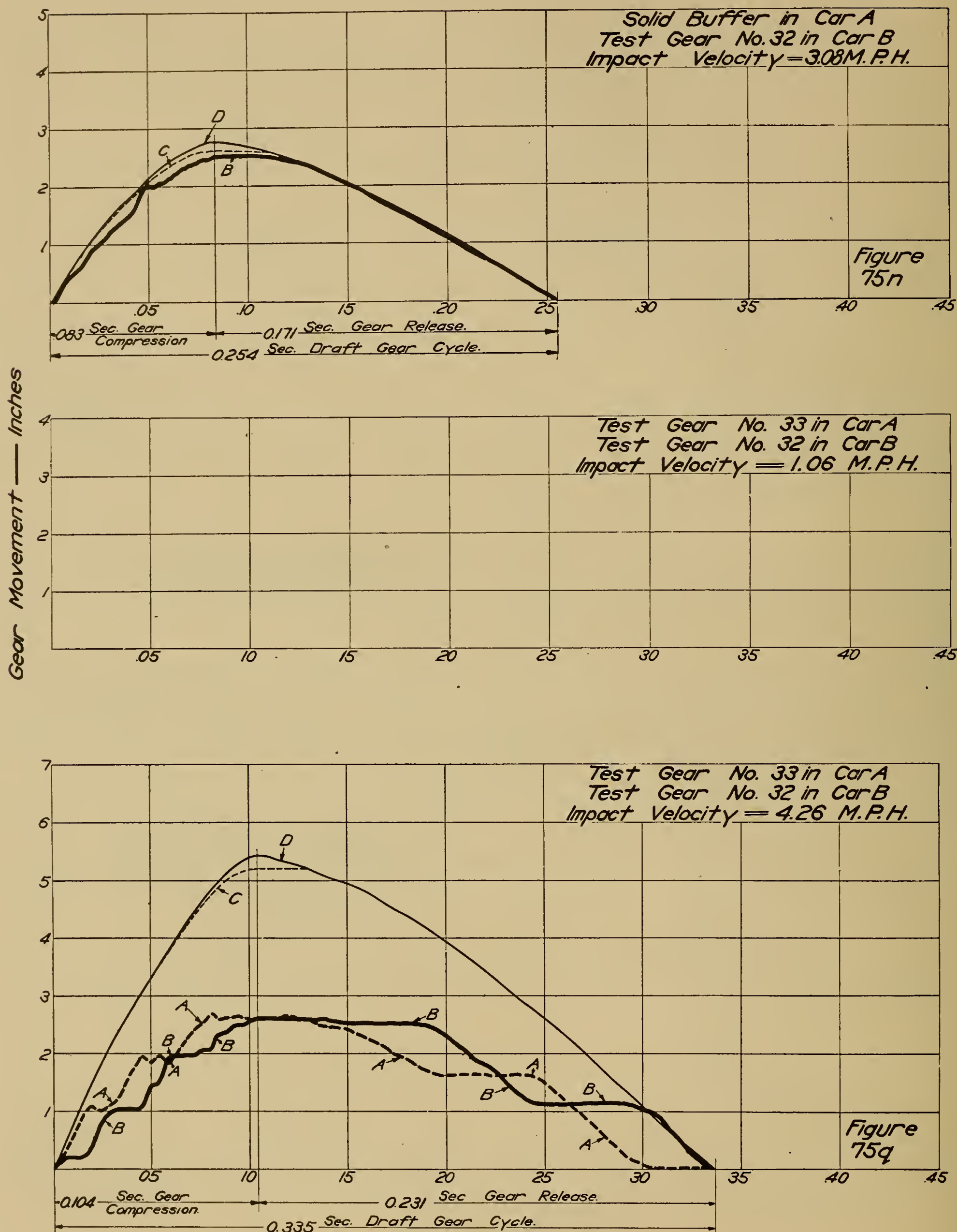
Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft-gear cycle.

Figs. 75g and 75j—Energy Curves, National M-1 Gears



Figs. 75k and 75m—Time-Force Curves, National M-1 Gears



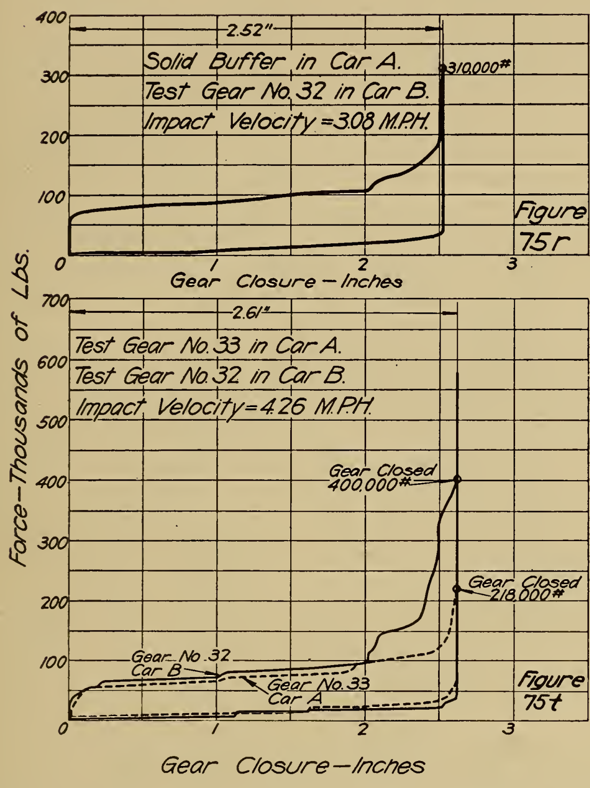
Curve D, determined from superimposed car-movement curves, represents combined draft-gear movement and yield of car bodies.

Curve C, obtained by eliminating car-body yield from curve D, represents true combined movement of both gears.

Curve B, traced on small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

Figs. 75n and 75q—Time-Closure Curves, National M-1 Gears



FIGS. 75R AND 75T—FORCE-CLOSURE DIAGRAMS, NATIONAL M-1 GEARS

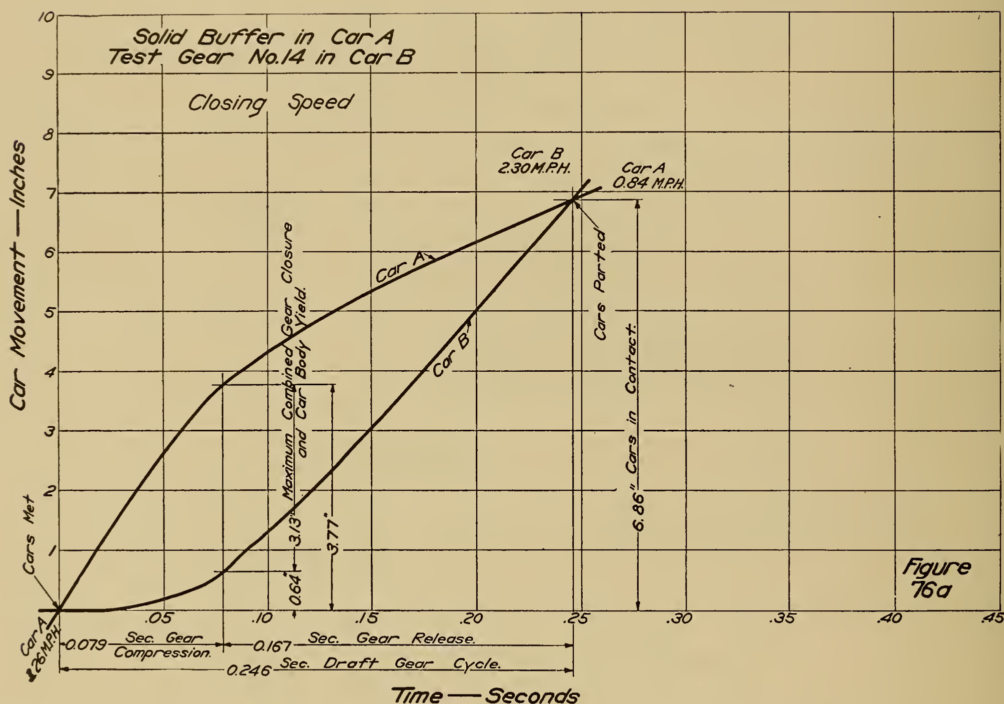
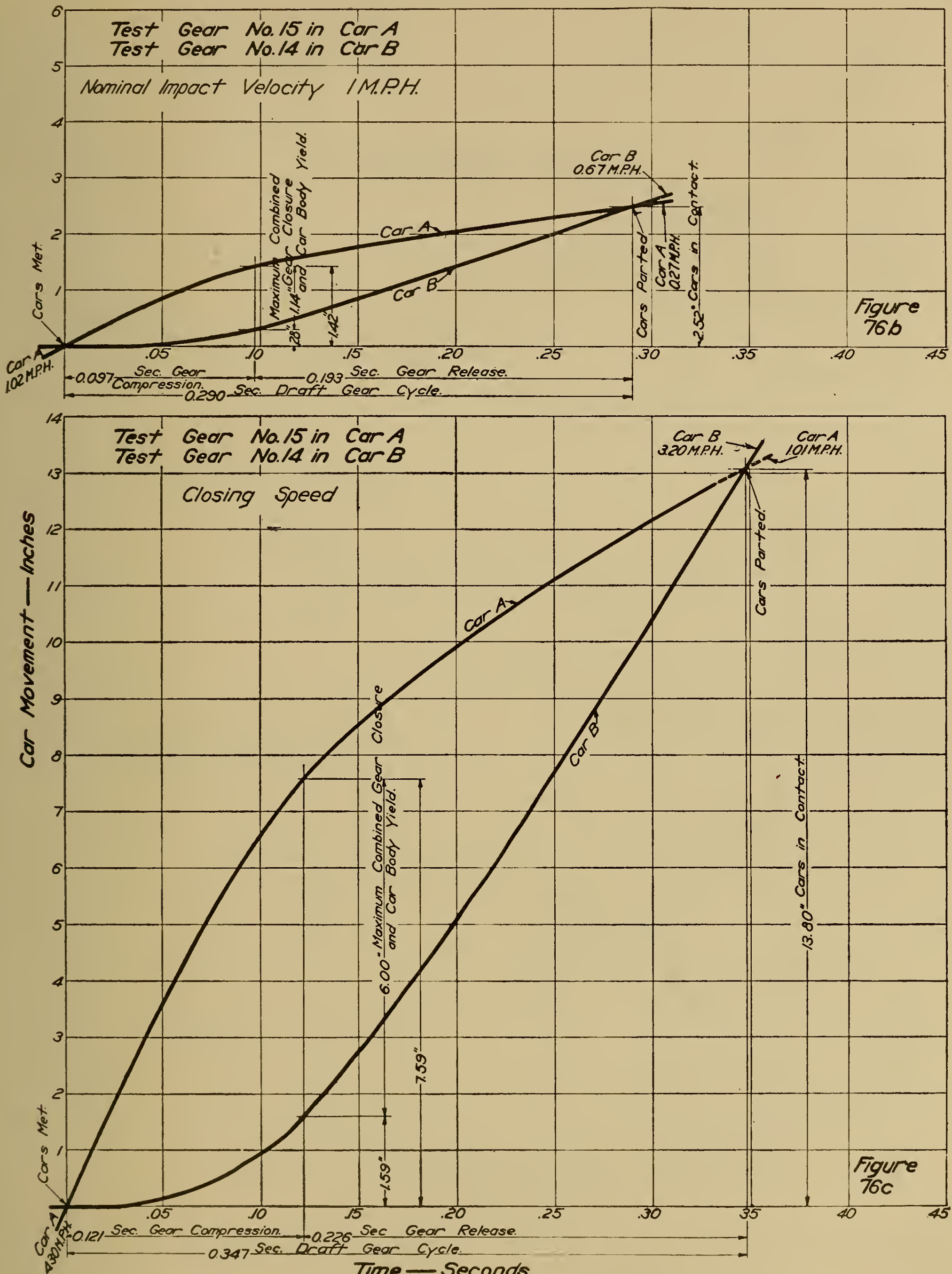
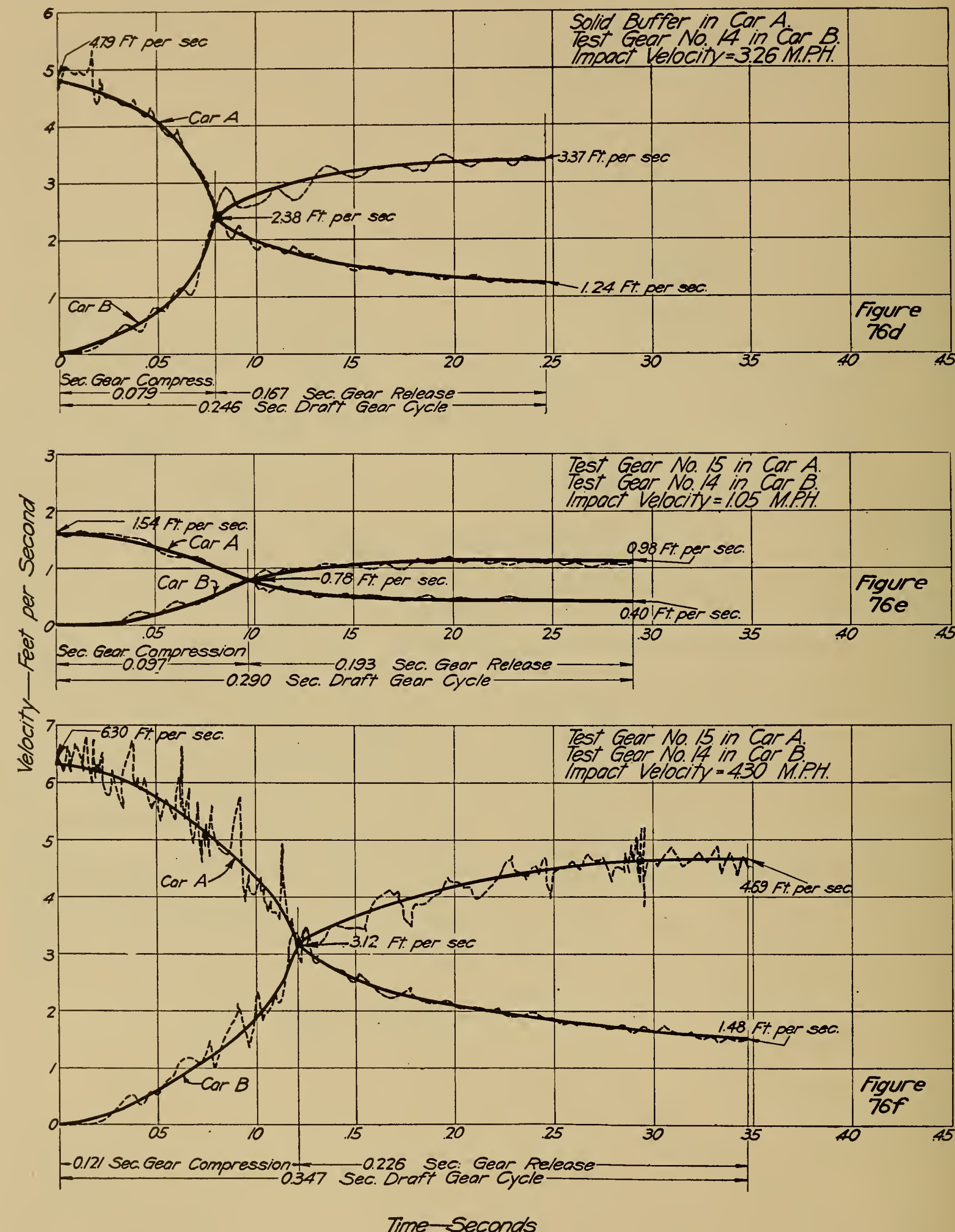


FIG. 76A—CAR-MOVEMENT CURVES, SUPERIMPOSED. SESSIONS JUMBO GEARS. THESE CURVES DRAWN BY CARS IN TEST



Figs. 76b and 76c—Car-Movement Curves, Superimposed. Sessions Jumbo Gears
These Curves Drawn by Cars in Test

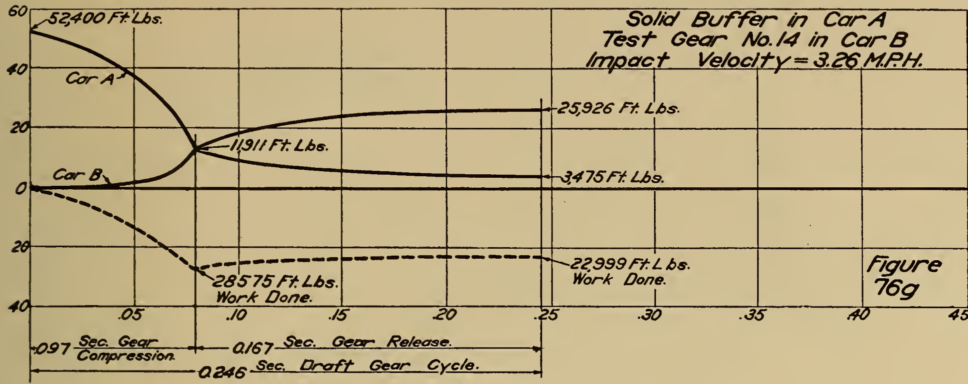


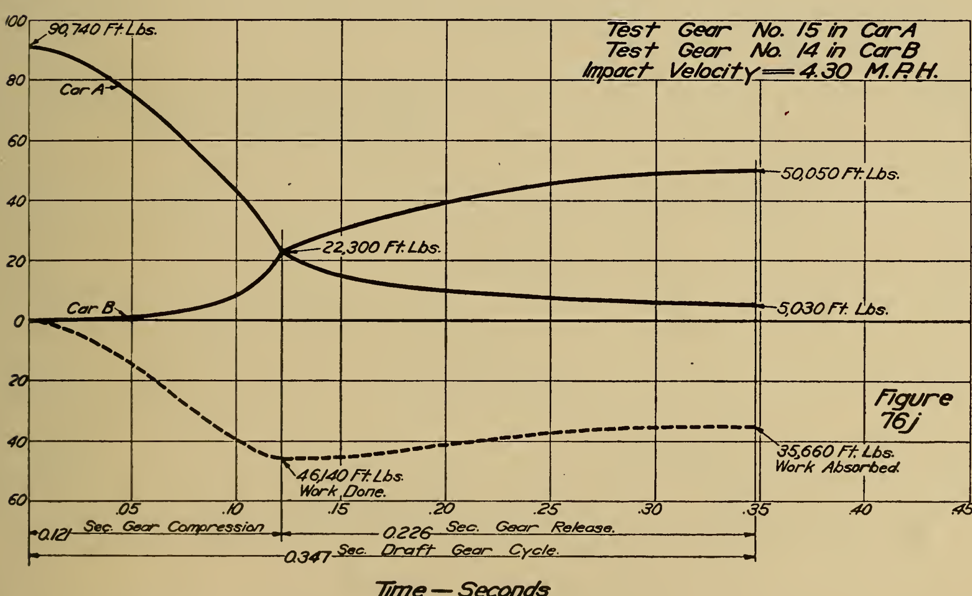
Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

The irregularities are due in general to vibrations of the car structure induced by draft-gear action.

Full lines represent the mean velocity curves.

FIGS. 76D, 76E AND 76F—VELOCITY CURVES, SESSIONS JUMBO GEARS

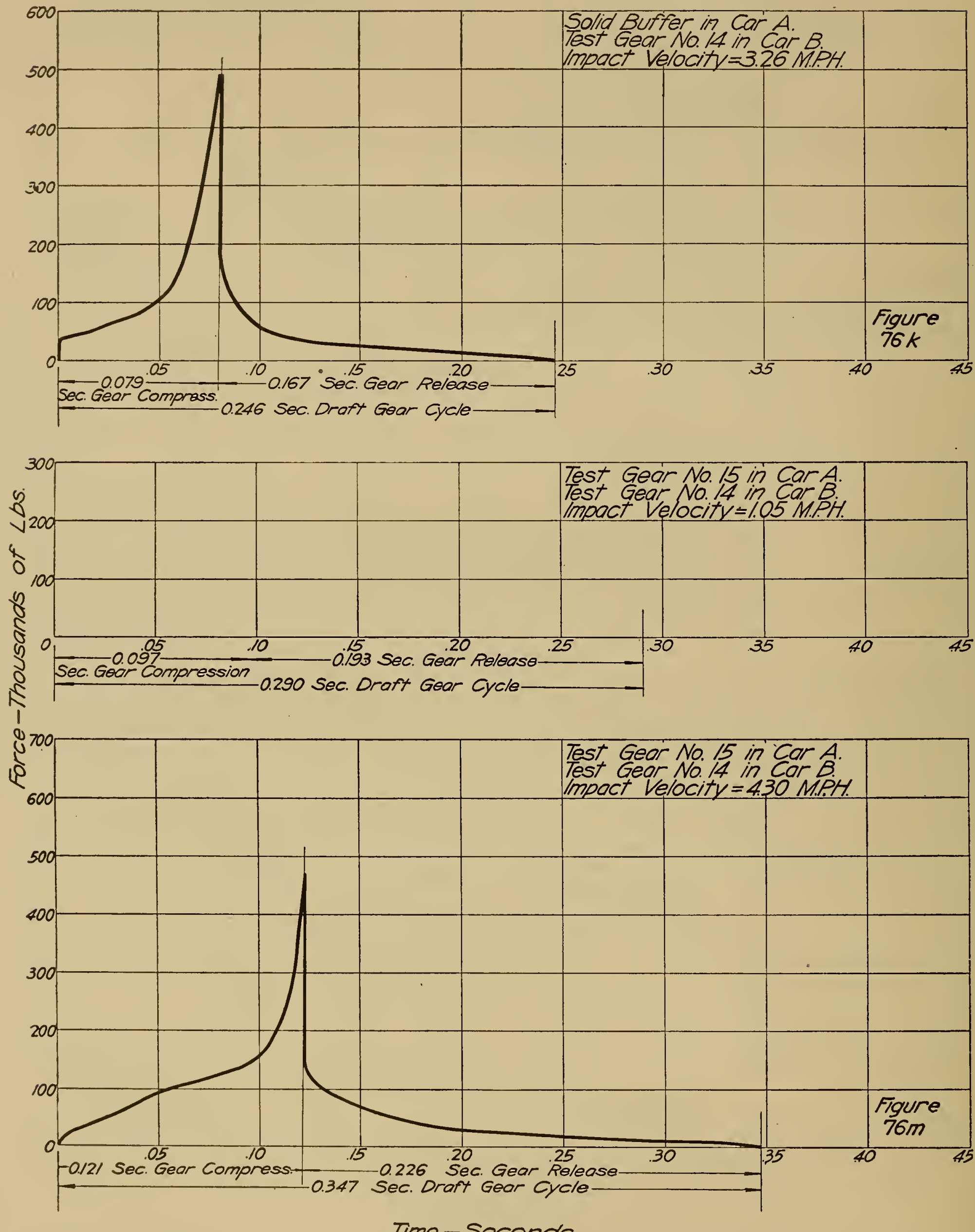




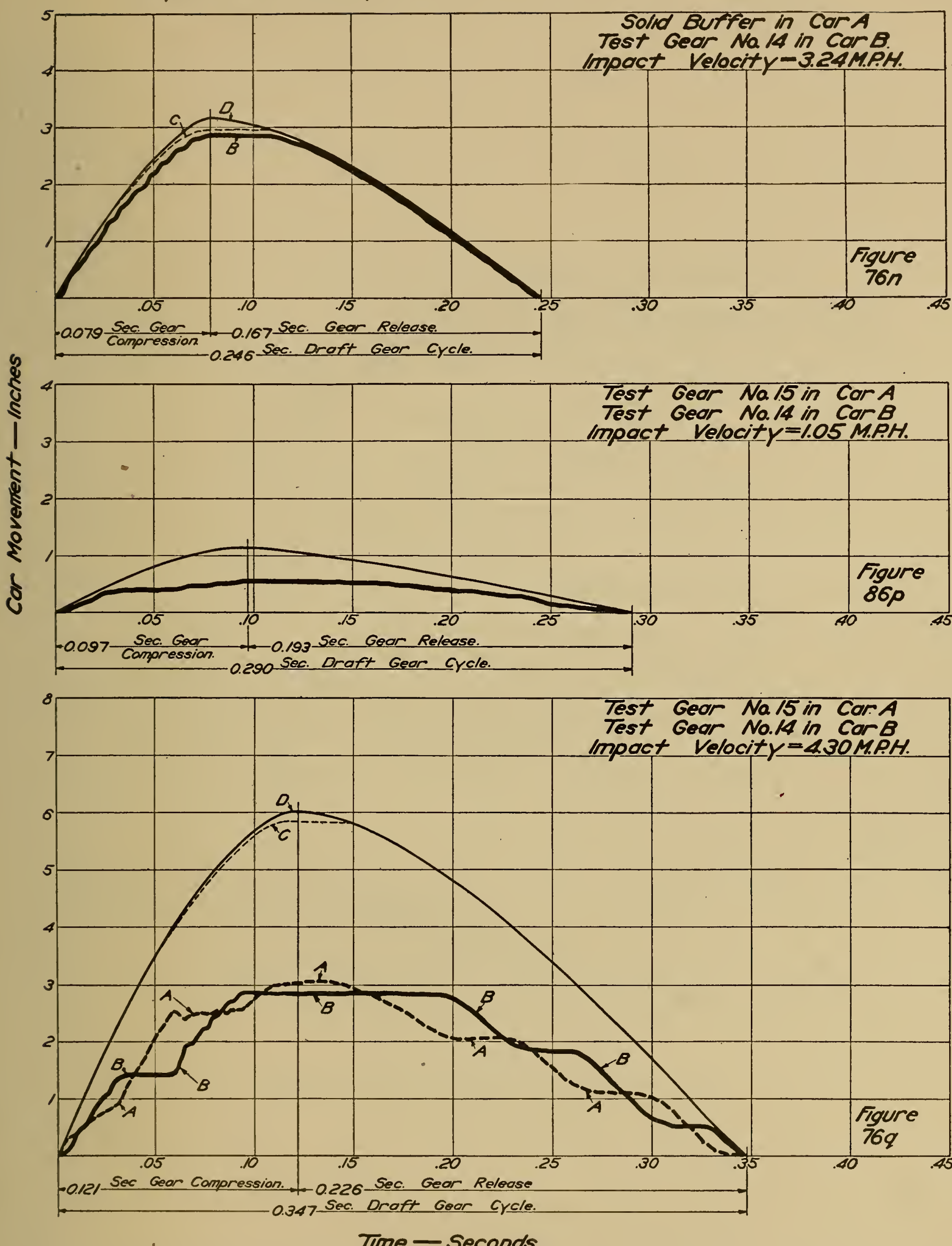
Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft-gear cycle.

FIGS. 76G AND 76J—ENERGY CURVES, SESSIONS JUMBO GEARS



FIGS. 76K AND 76M—TIME-FORCE CURVES, SESSIONS JUMBO GEARS



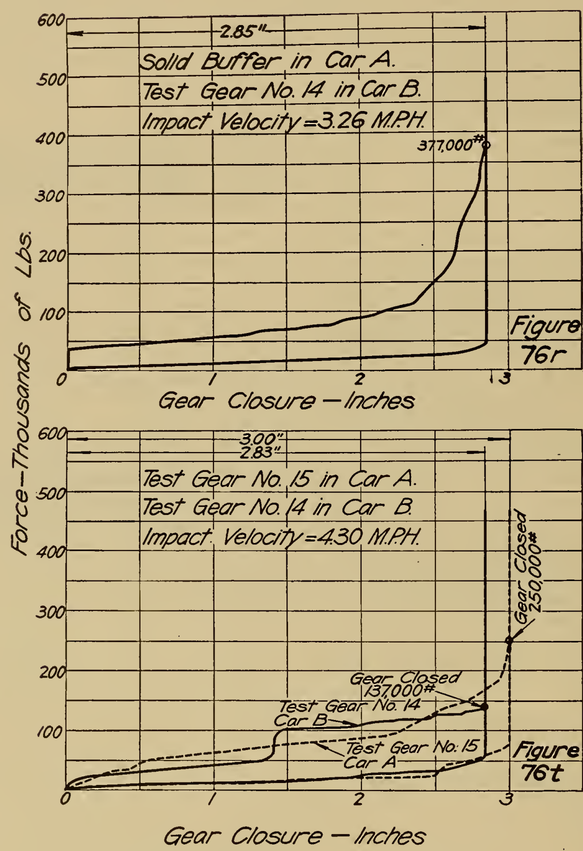
Curve D, determined from superimposed car-movement curves, represents combined draft-gear movement and yield of car bodies.

Curve C, obtained by eliminating car-body yield from curve D, represents true combined movement of both gears.

Curve B, traced on small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

Figs. 76n, 76p and 76q—Time-Closure Curves, Sessions Jumbo Gears



Figs. 76r and 76t—Force-Closure Diagrams Sessions Jumbo Gears

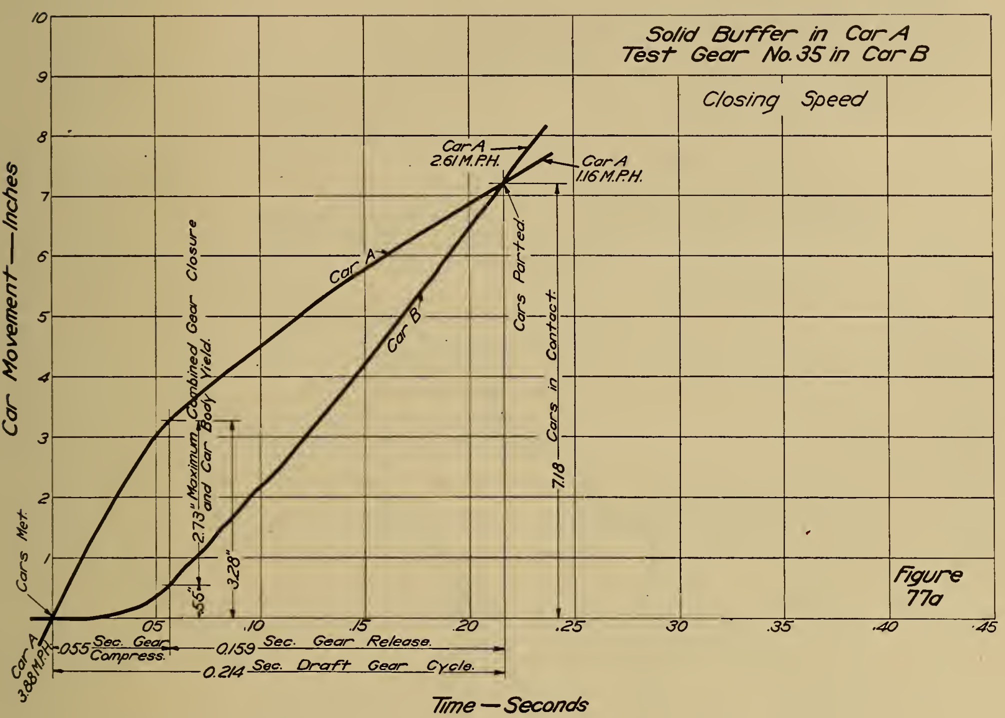
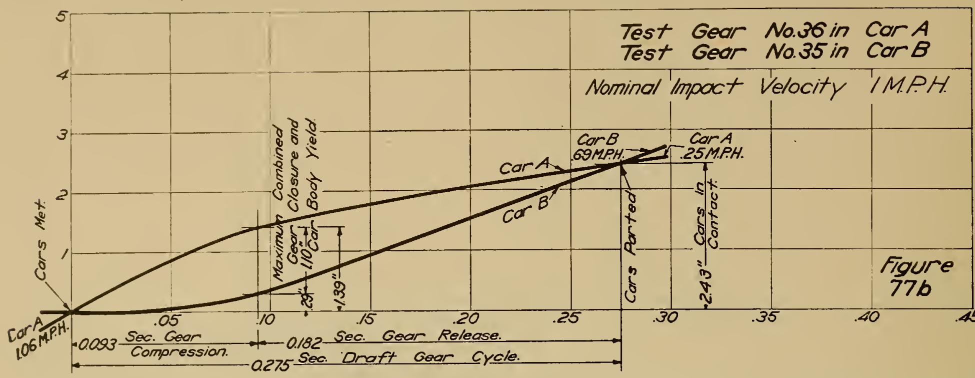
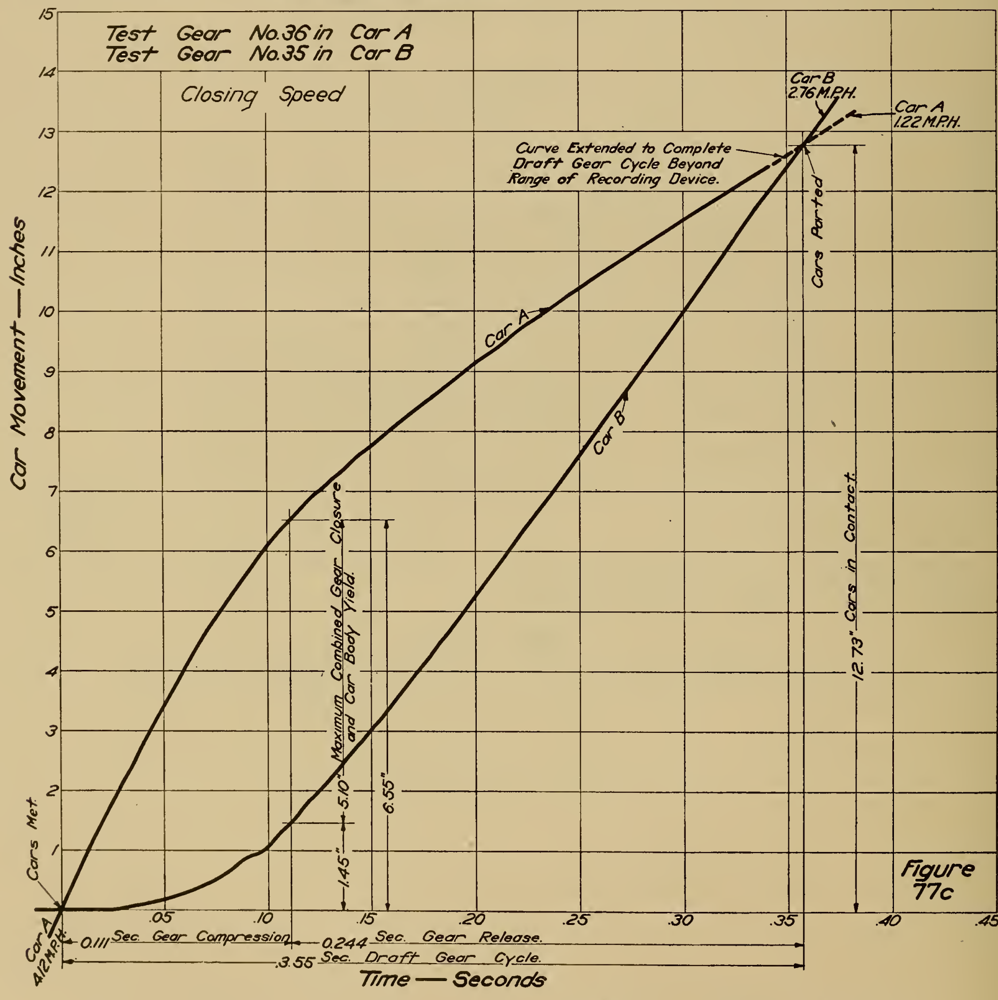
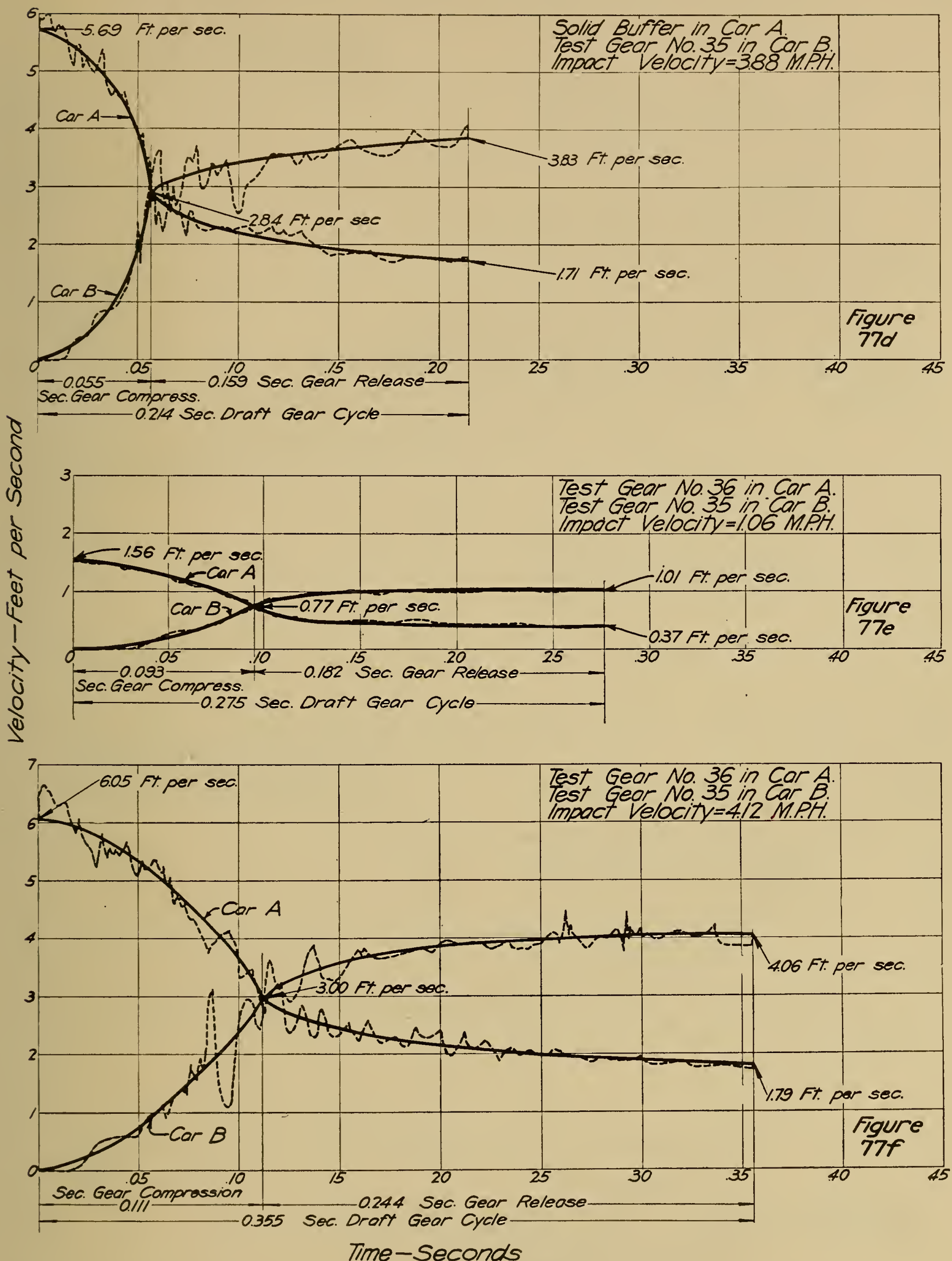


Fig. 77a—Car-Movement Curves, Superimposed. National M-4 Gears
These Curves Drawn by Cars in Test





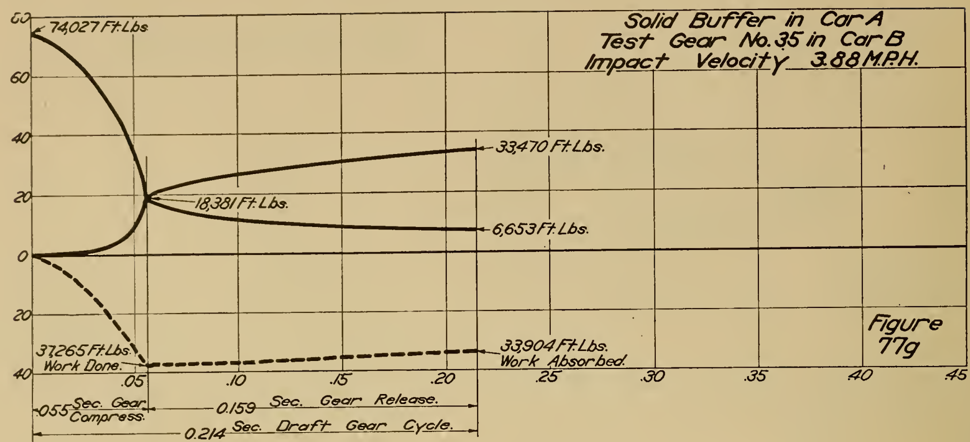
Figs. 77b and 77c—Car-Movement Curves, Superimposed. National M-4 Gears These Curves Drawn by Cars in Test

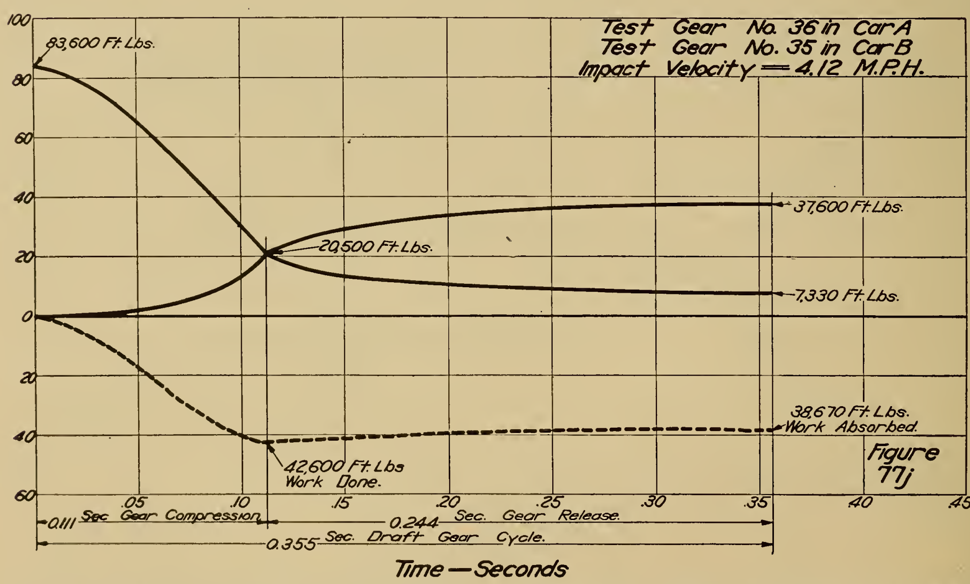


Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

The irregularities are due in general to vibrations of the car structure induced by draft-gear action.
Full lines represent the mean velocity curves.

Figs. 77d, 77e and 77f—Velocity Curves, National M-4 Gears

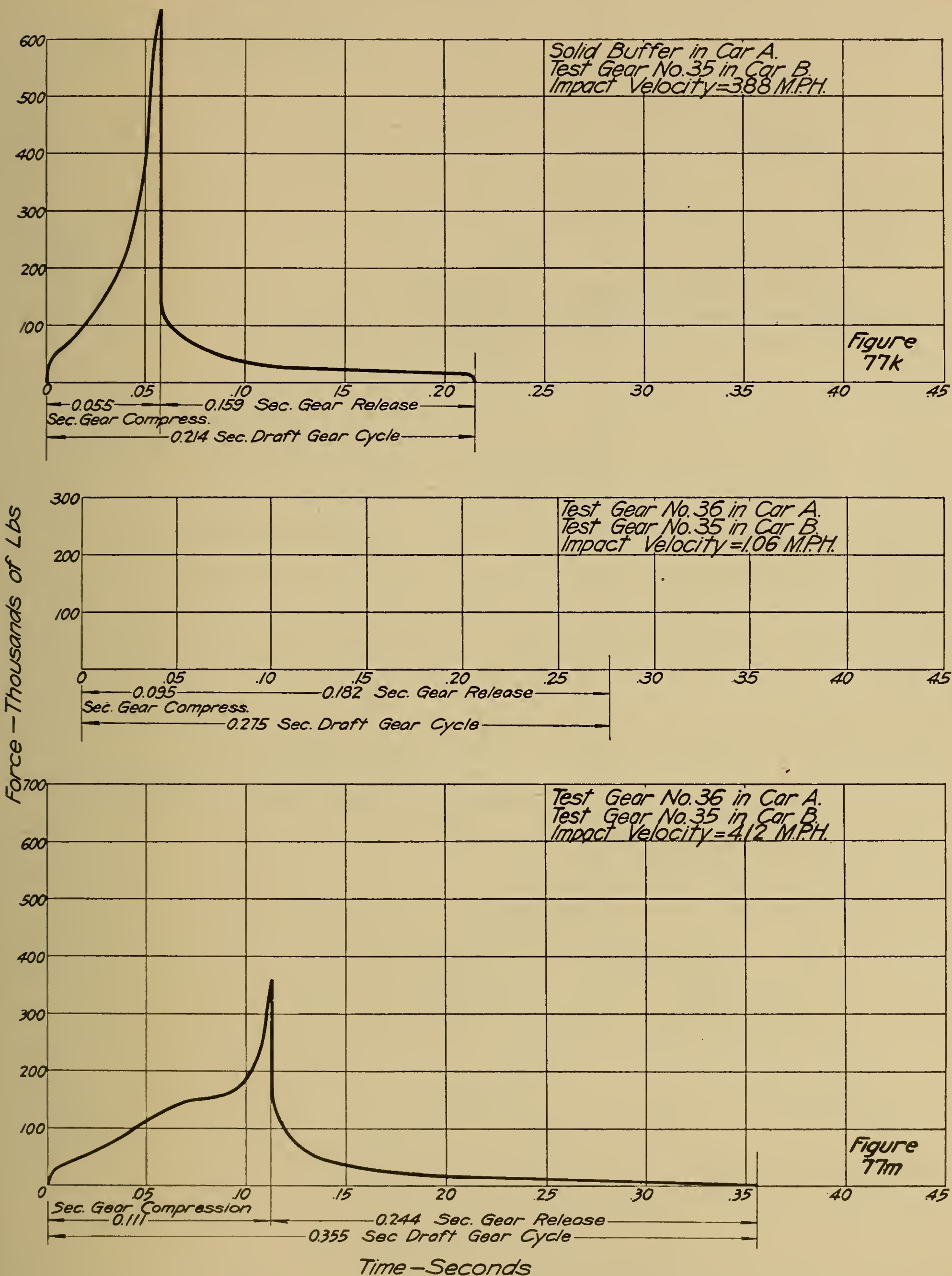




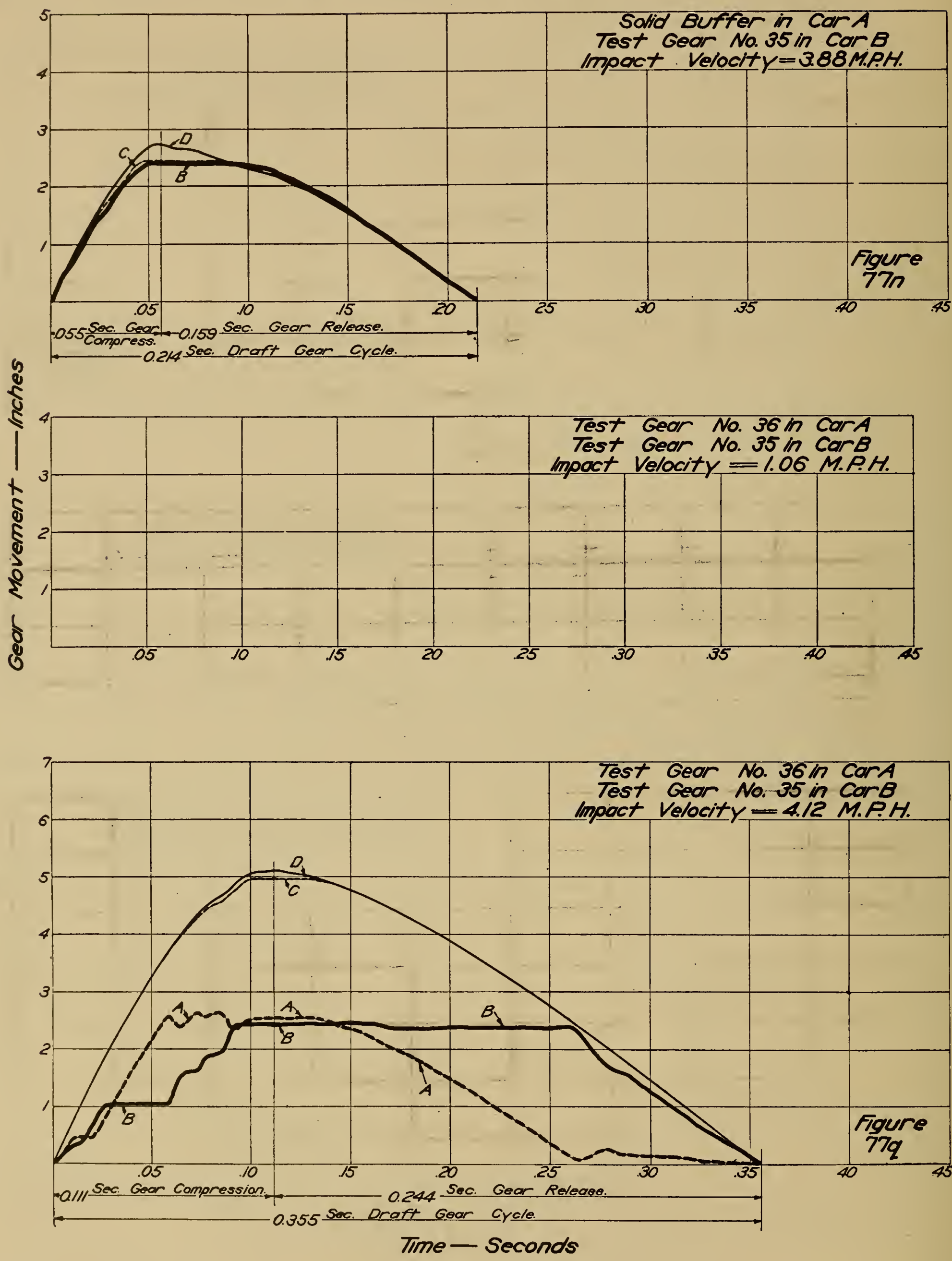
Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during draft-gear cycle.

FIGS. 77G AND 77J—ENERGY CURVES, NATIONAL M-4 GEARS



Figs. 77k and 77m—Time-Force Curves, National M-4 Gears



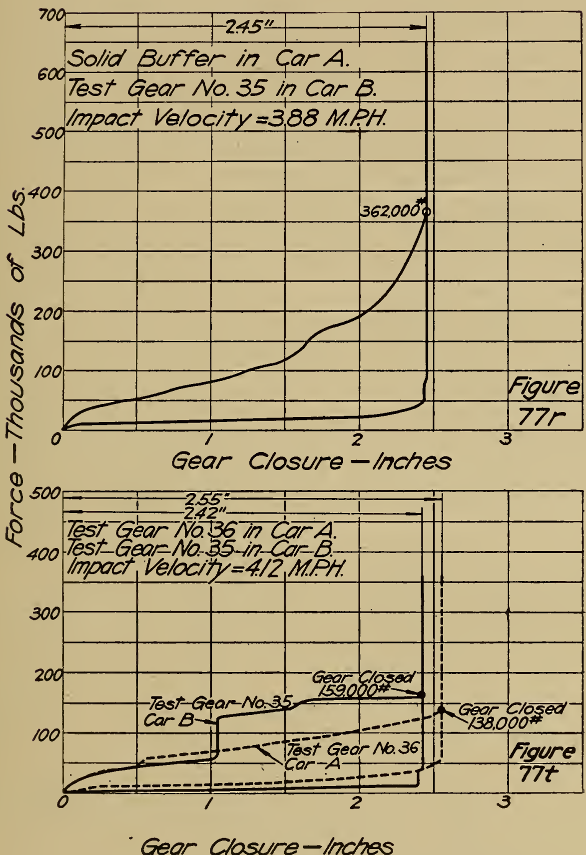
Curve D, determined from superimposed car-movement curves, represents combined draft gear movement and yield of car bodies.

Curve C, obtained by eliminating car body yield from curve D, represents true combined movement of both gears.

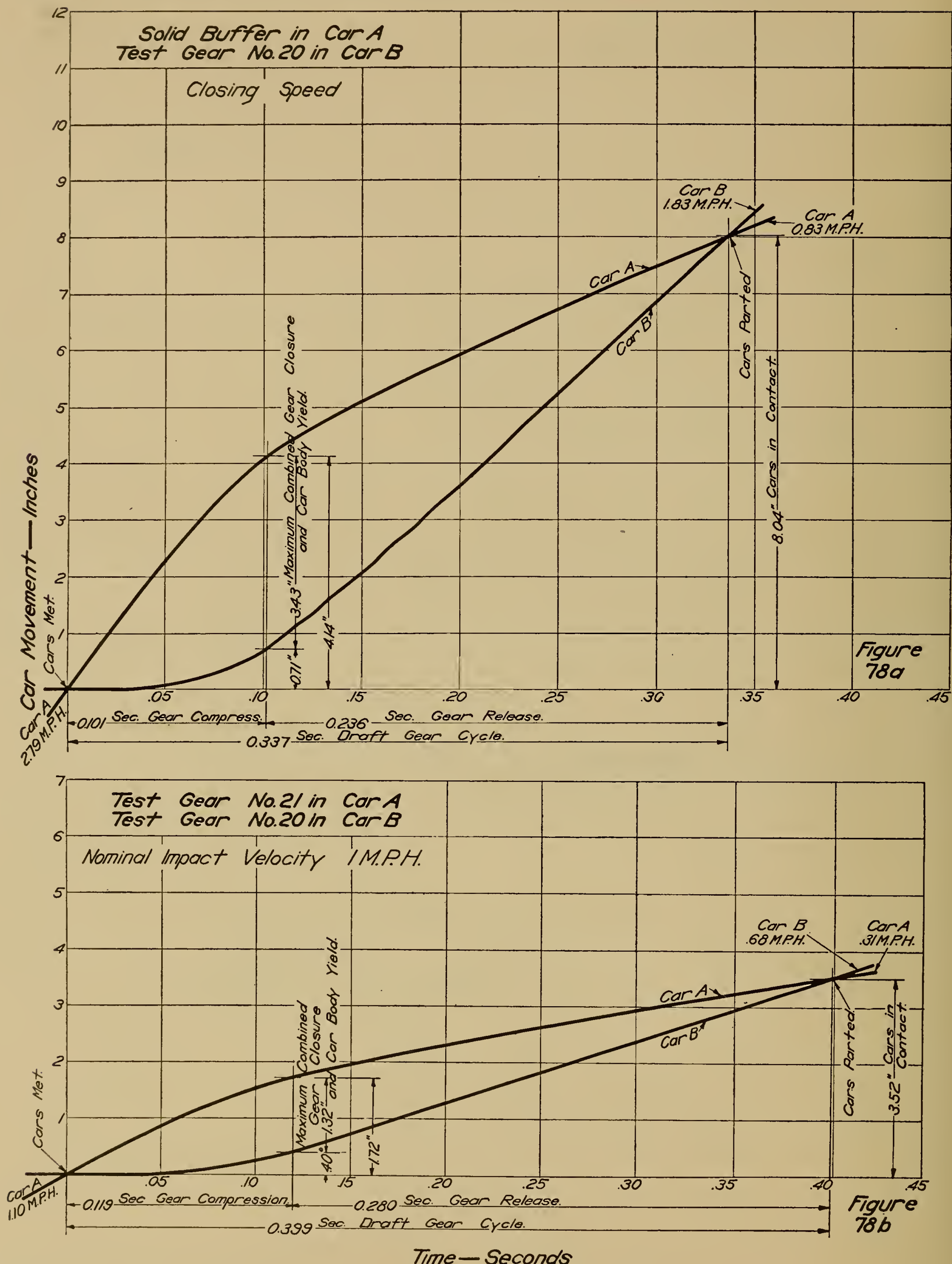
Curve B, traced on small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

FIGS. 77N AND 77Q—TIME-CLOSURE CURVES, NATIONAL M-4 GEARS



Figs. 77r and 77t—Force-Closure Diagrams
National M-4 Gears



Figs. 78a and 78b—Car-Movement Curves, Superimposed. Cardwell G-18-A Gears These Curves Drawn by Cars in Test

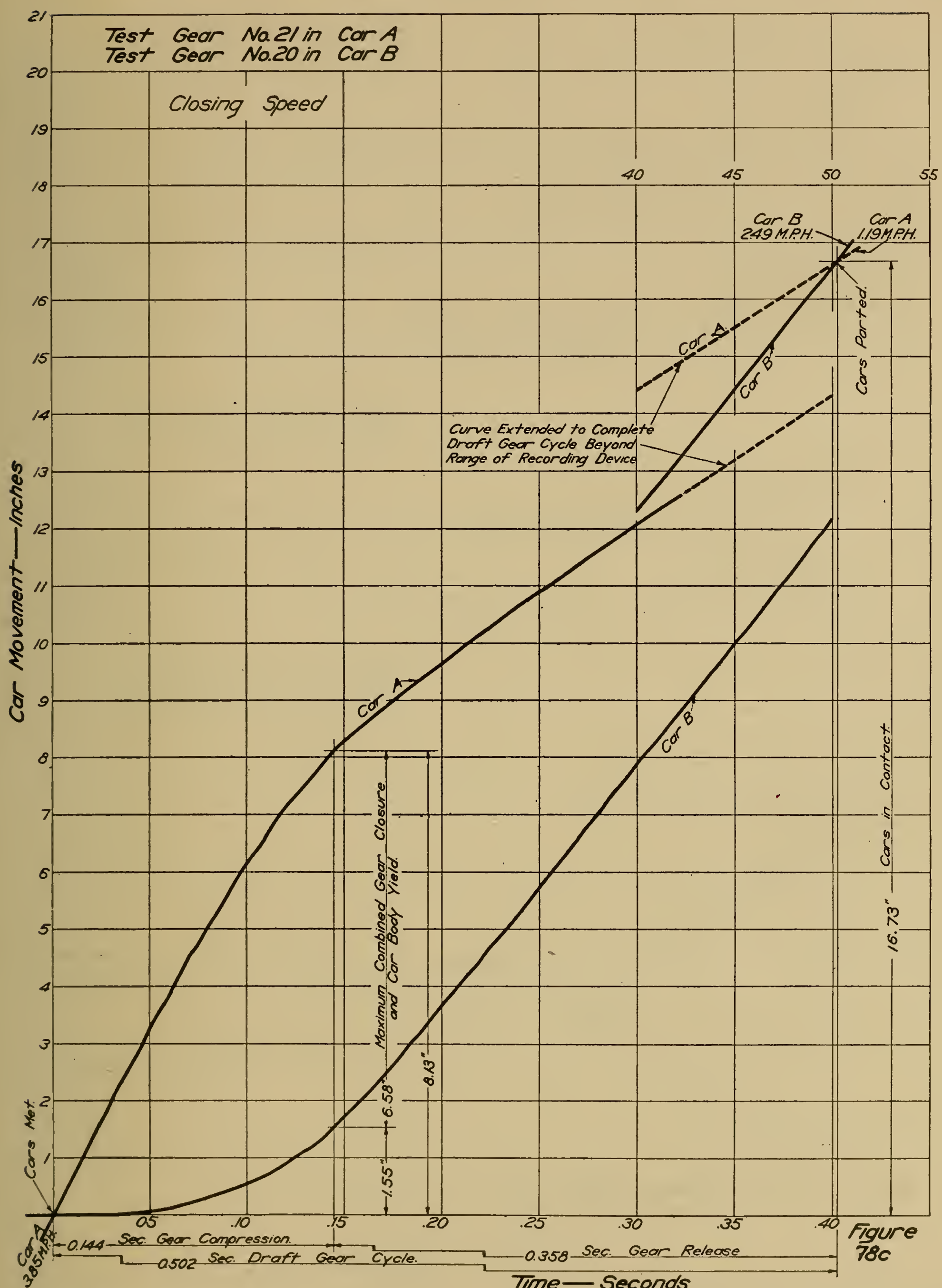
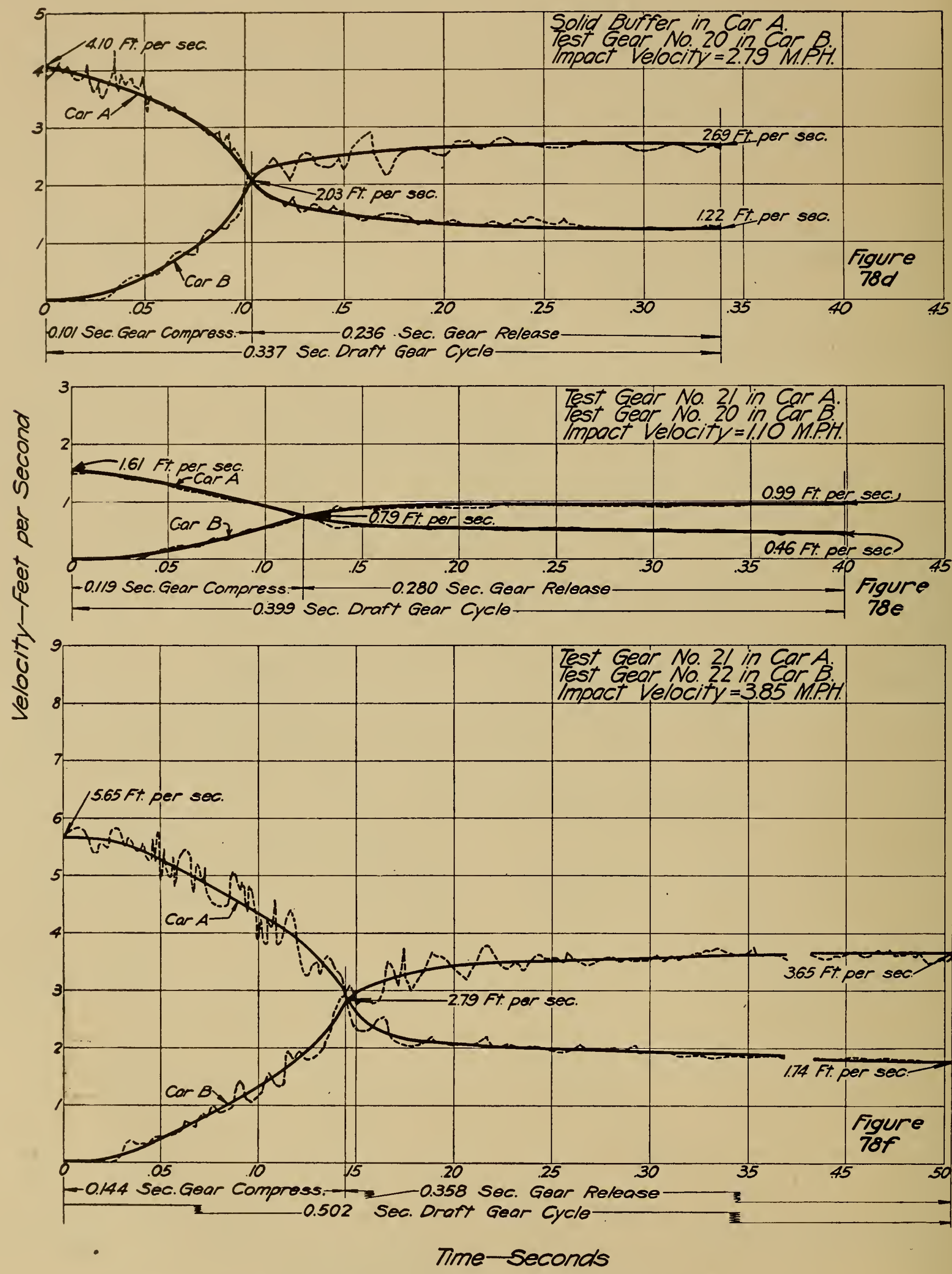


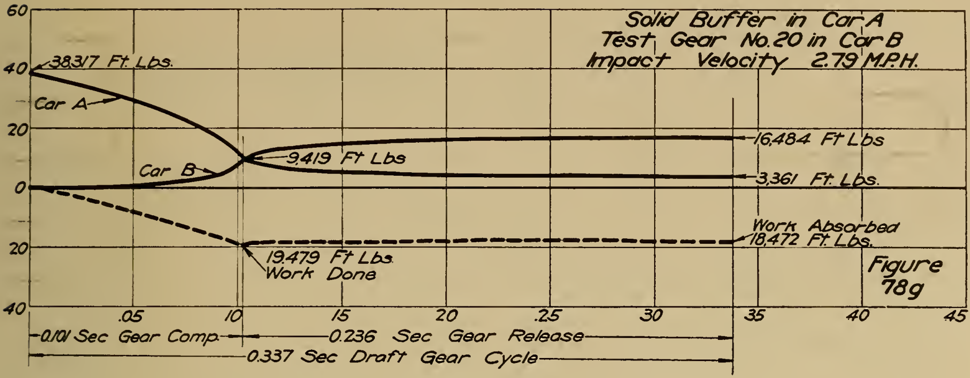
Fig. 78c—Car-Movement Curves, Superimposed. Cardwell G-18-A Gears These Curves Drawn by Cars in Test

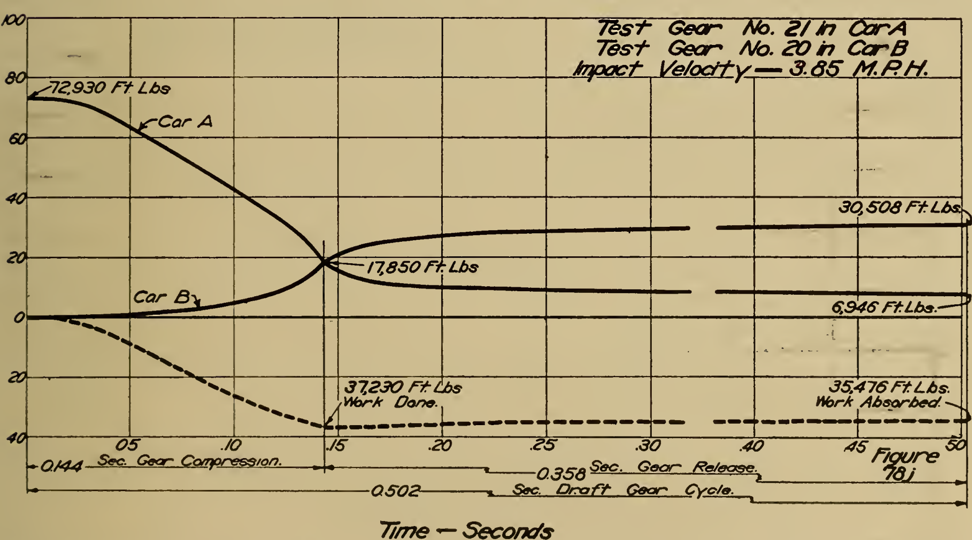


Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

The irregularities are due in general to vibrations of the car structure induced by draft gear action.
Full lines represent the mean velocity curves.

Figs. 78d, 78e and 78f—Velocity Curves, Cardwell G-18-A Gears

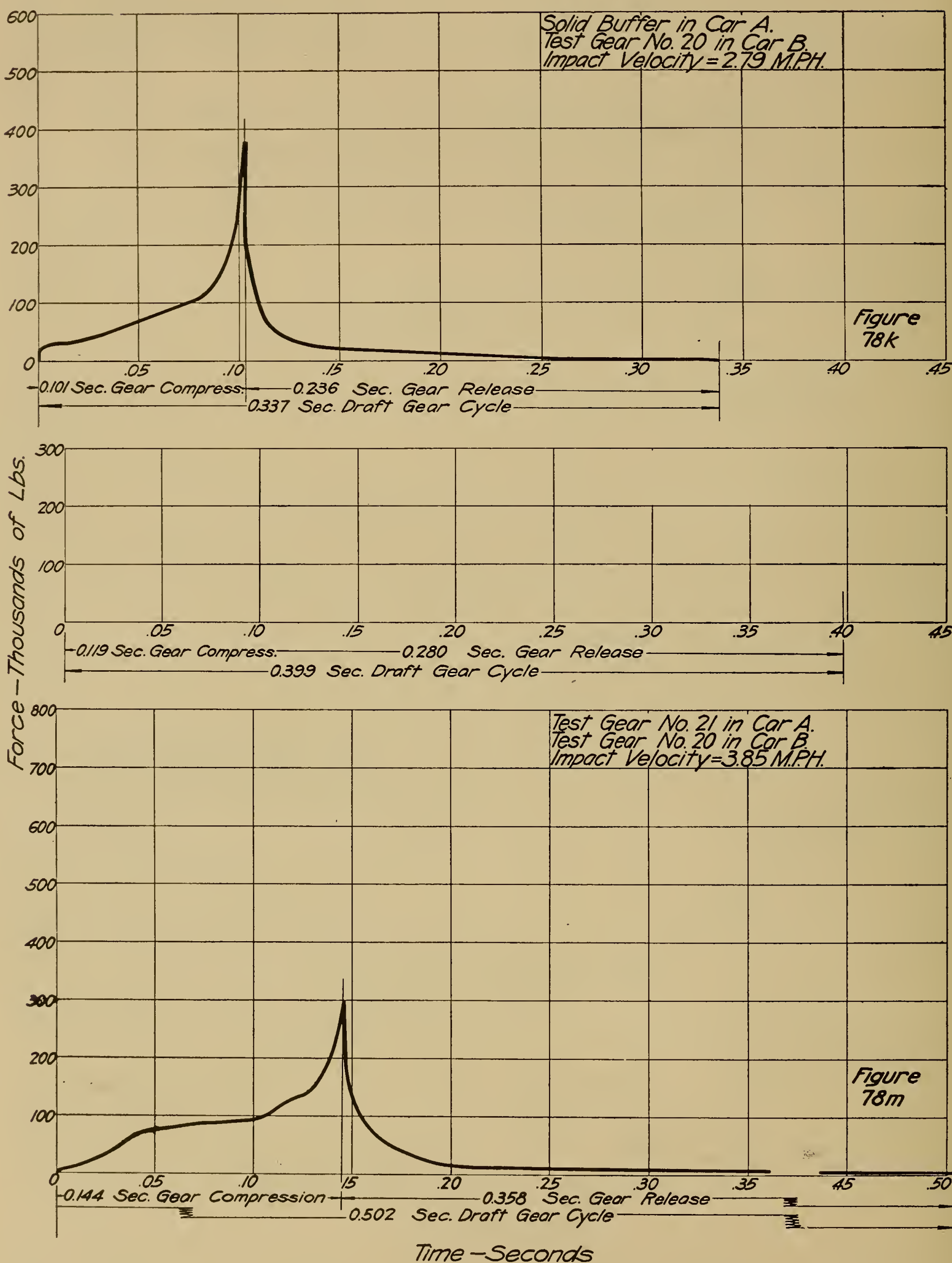




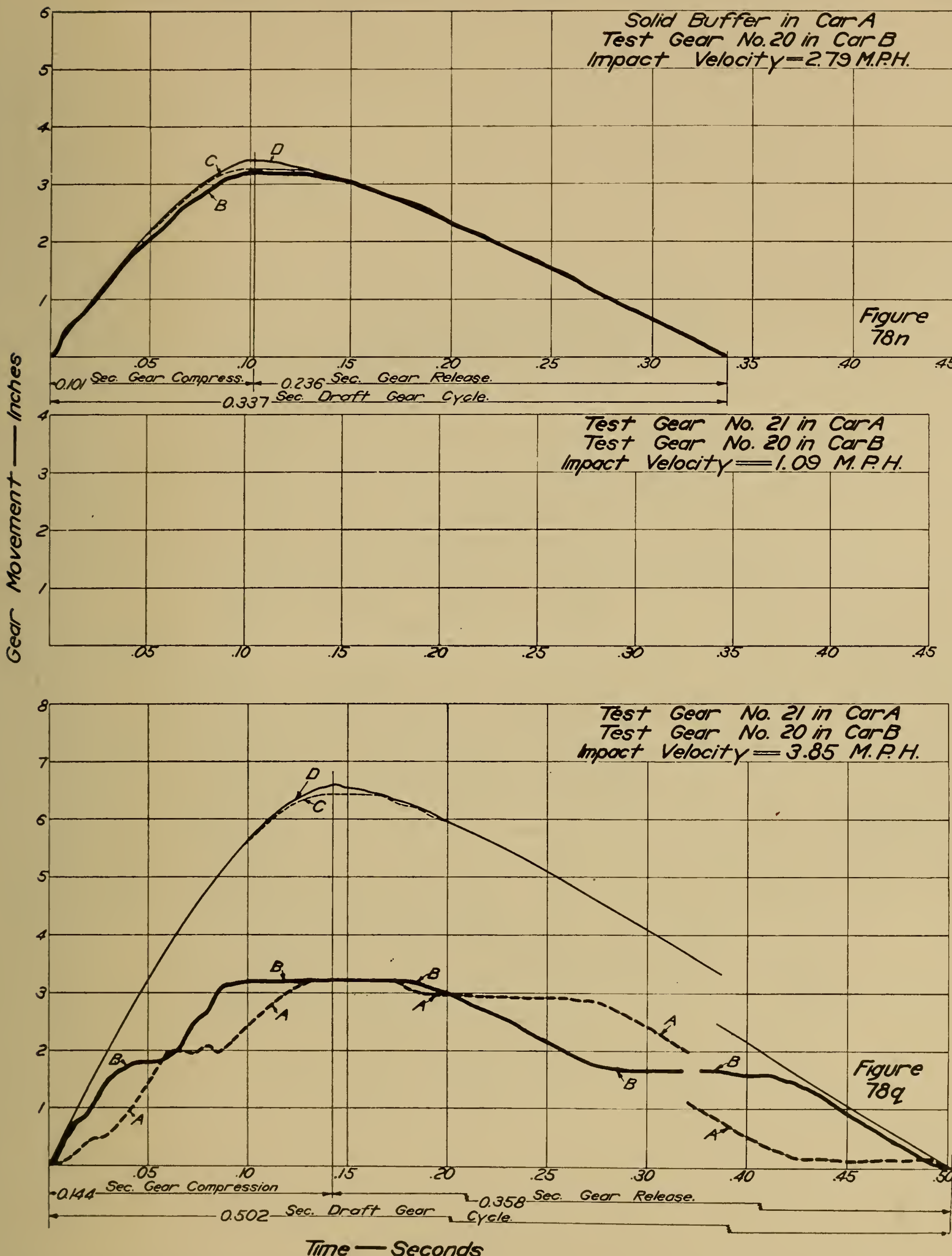
Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft gear cycle.

FIGS. 78G AND 78J—ENERGY CURVES, CARDWELL G-18-A GEARS



FIGS. 78K AND 78M—TIME-FORCE CURVES, CARDWELL G-18-A GEARS



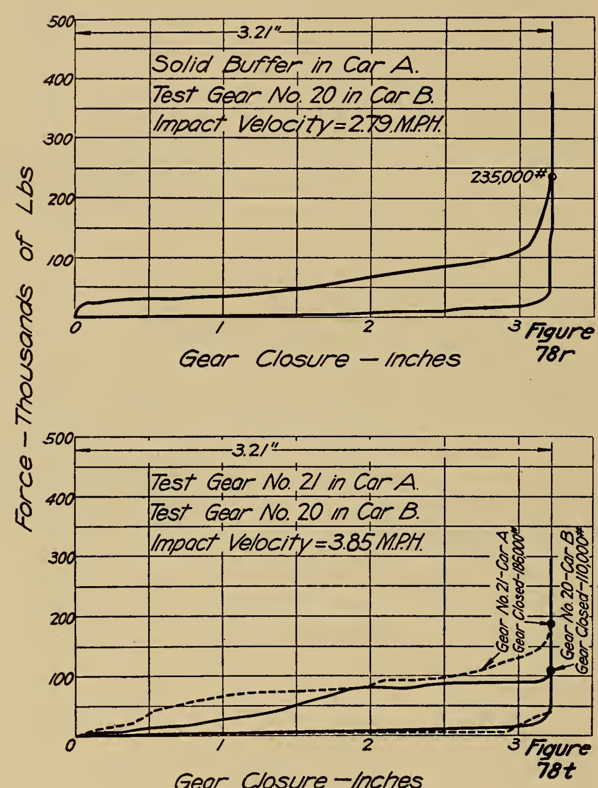
Curve D, determined from superimposed car-movement curves, represents combined draft gear movement and yield of car bodies.

Curve C, obtained by eliminating car body yield from curve D, represents true combined movement of both gears.

Curve B, traced on small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

Figs. 78n and 78q—Time-Closure Curves, Cardwell G-18-A Gears



Figs. 78r and 78t—Force-Closure Diagrams
Cardwell G-18-A Gears

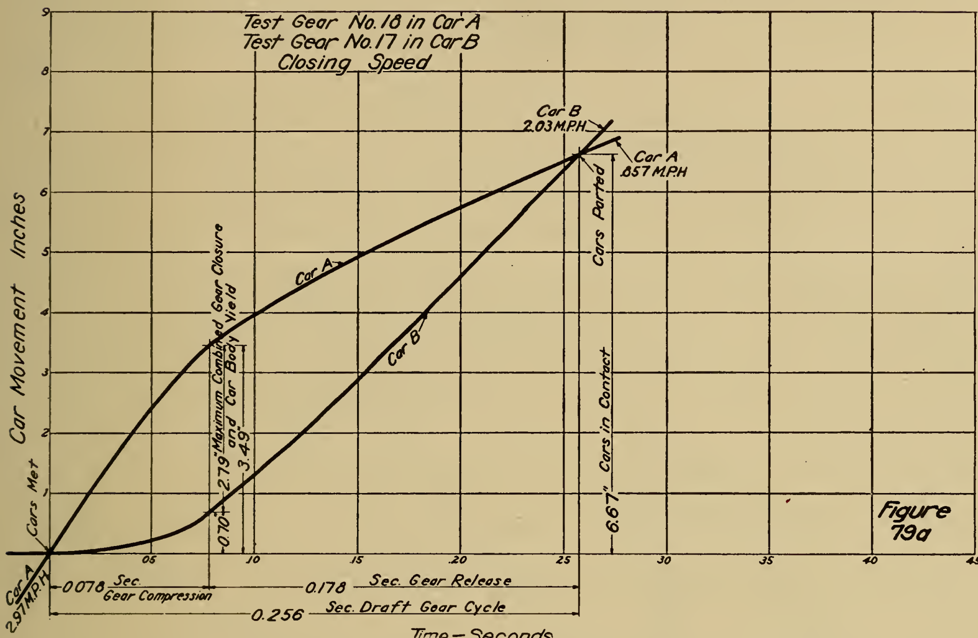
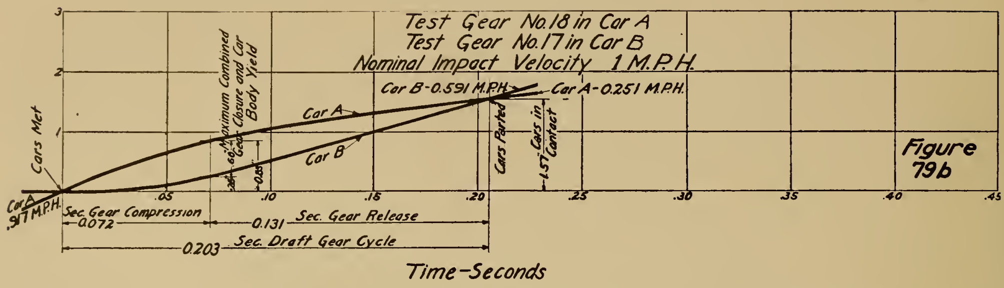
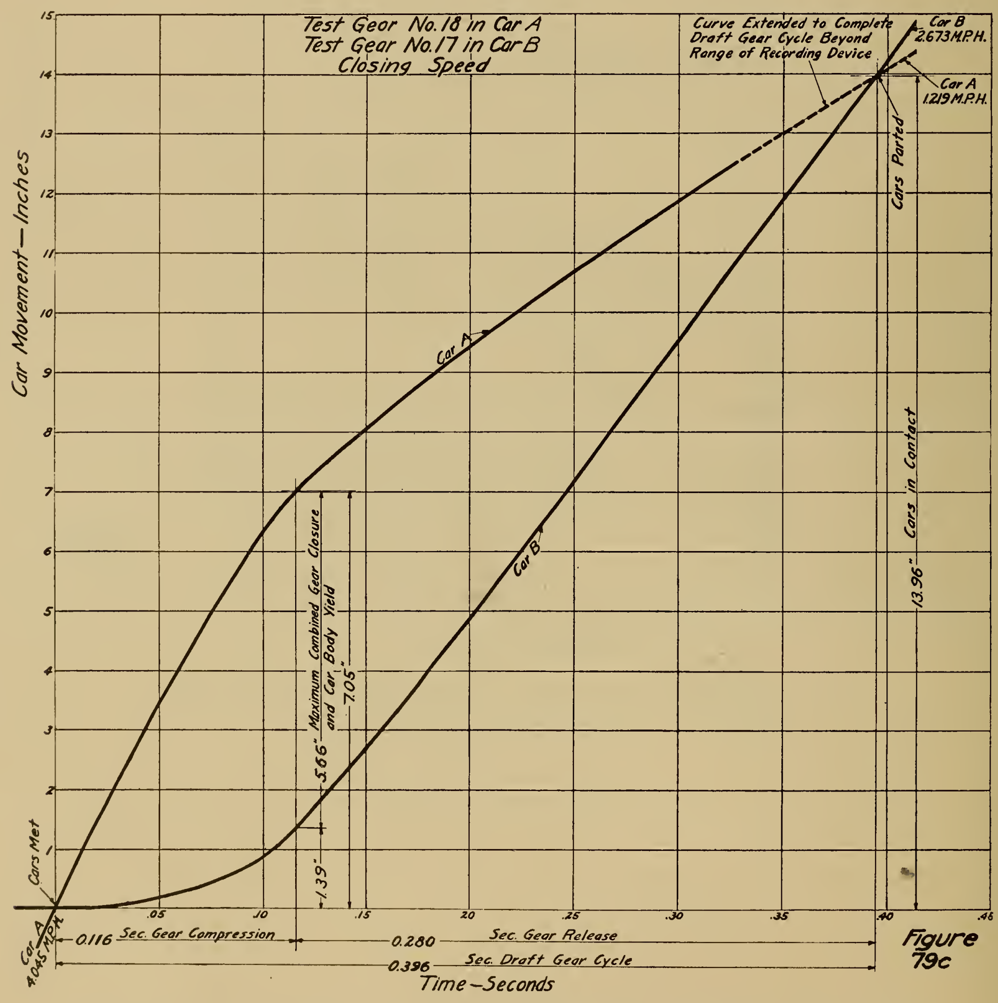
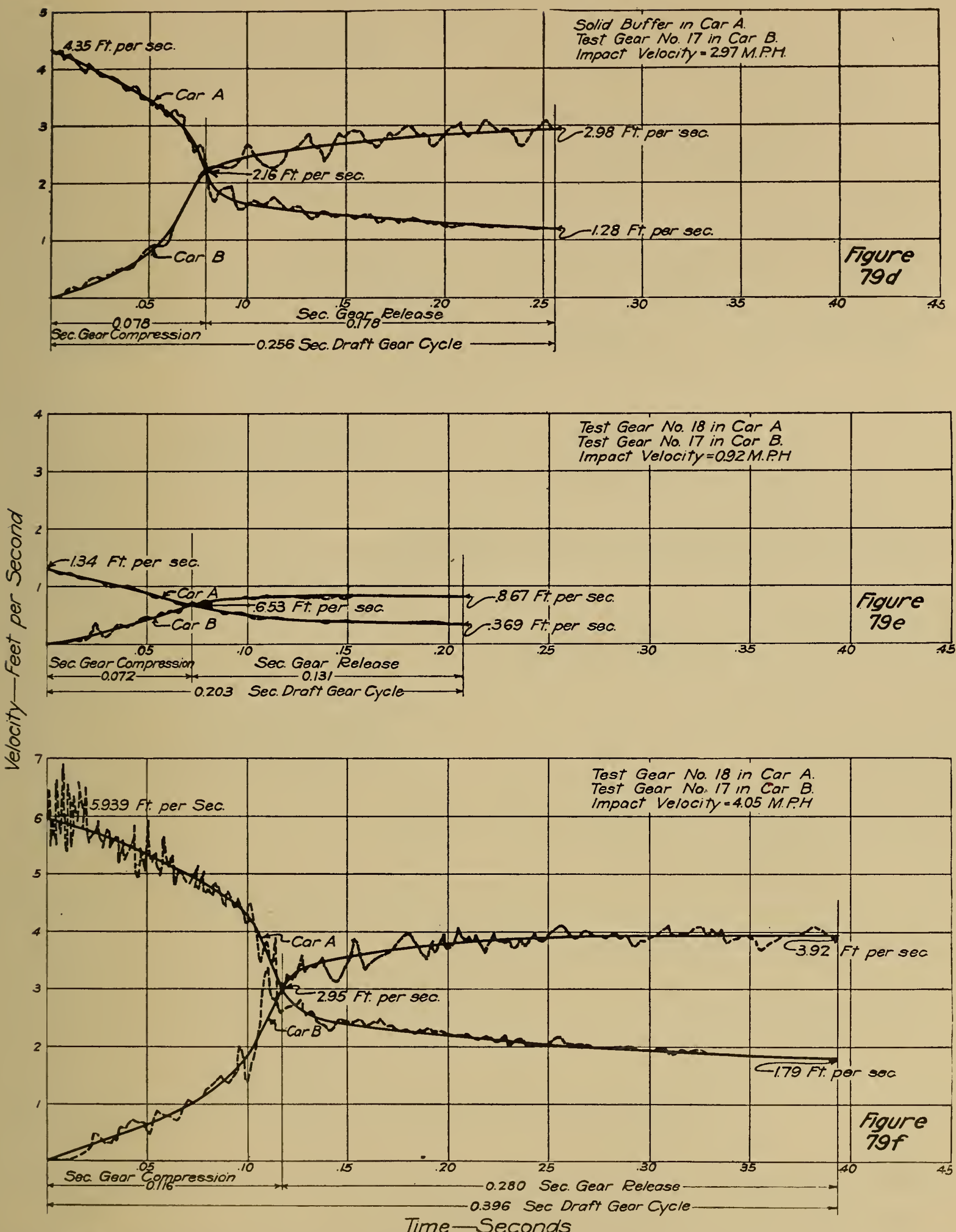


FIG. 79A—CAR-MOVEMENT CURVES, SUPERIMPOSED. CARDWELL G-25-A
THESE CURVES DRAWN BY CARS IN TESTS





Figs. 79b and 79c—Car-Movement Curves, Superimposed. Cardwell G-25-A Gears
These Curves Drawn by Cars in Tests

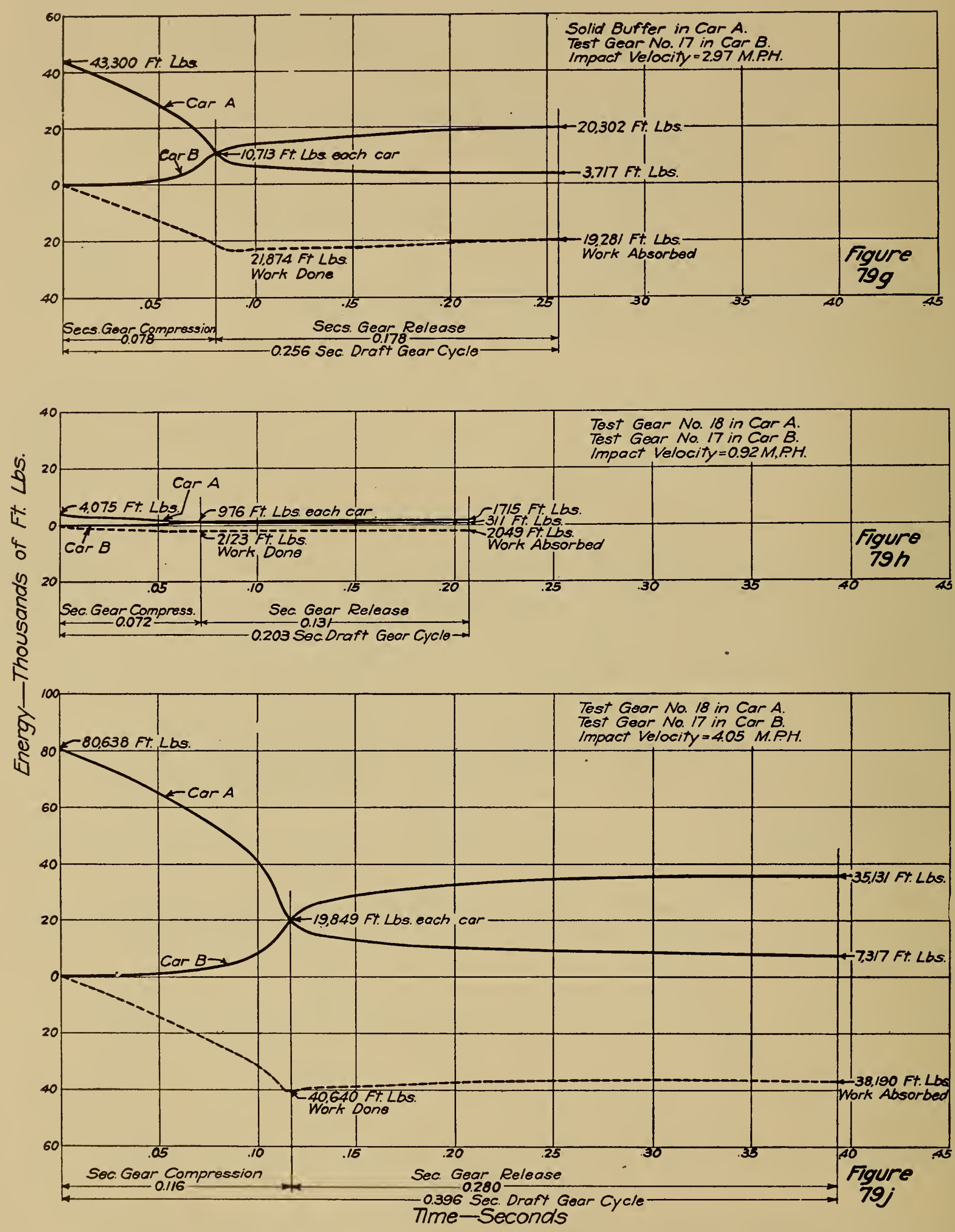


Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

The irregularities are due in general to vibrations of the car structures induced by draft gear action.

Full lines represent the mean velocity curves.

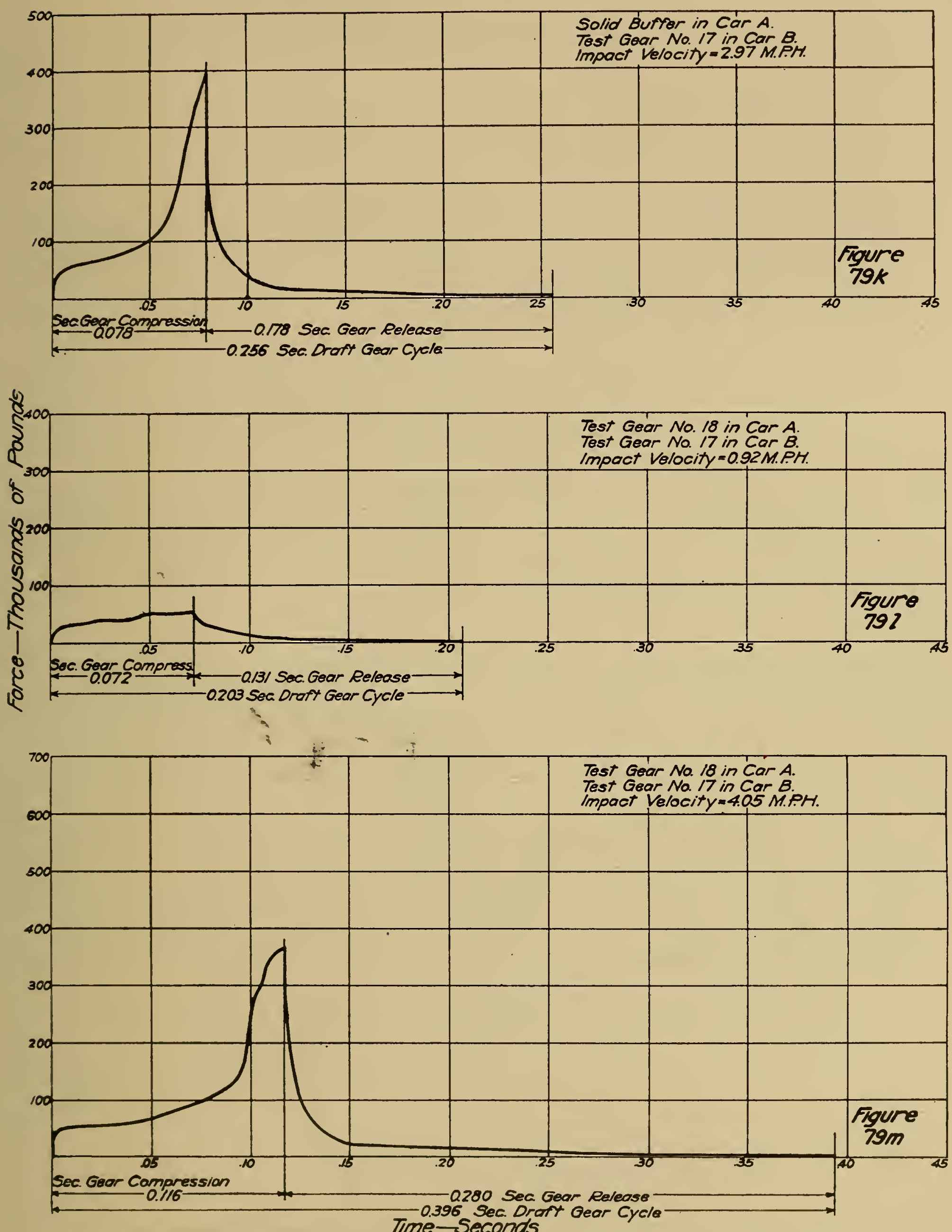
Figs. 79d, 79e and 79f—Velocity Curves, Cardwell G-25-A Gears



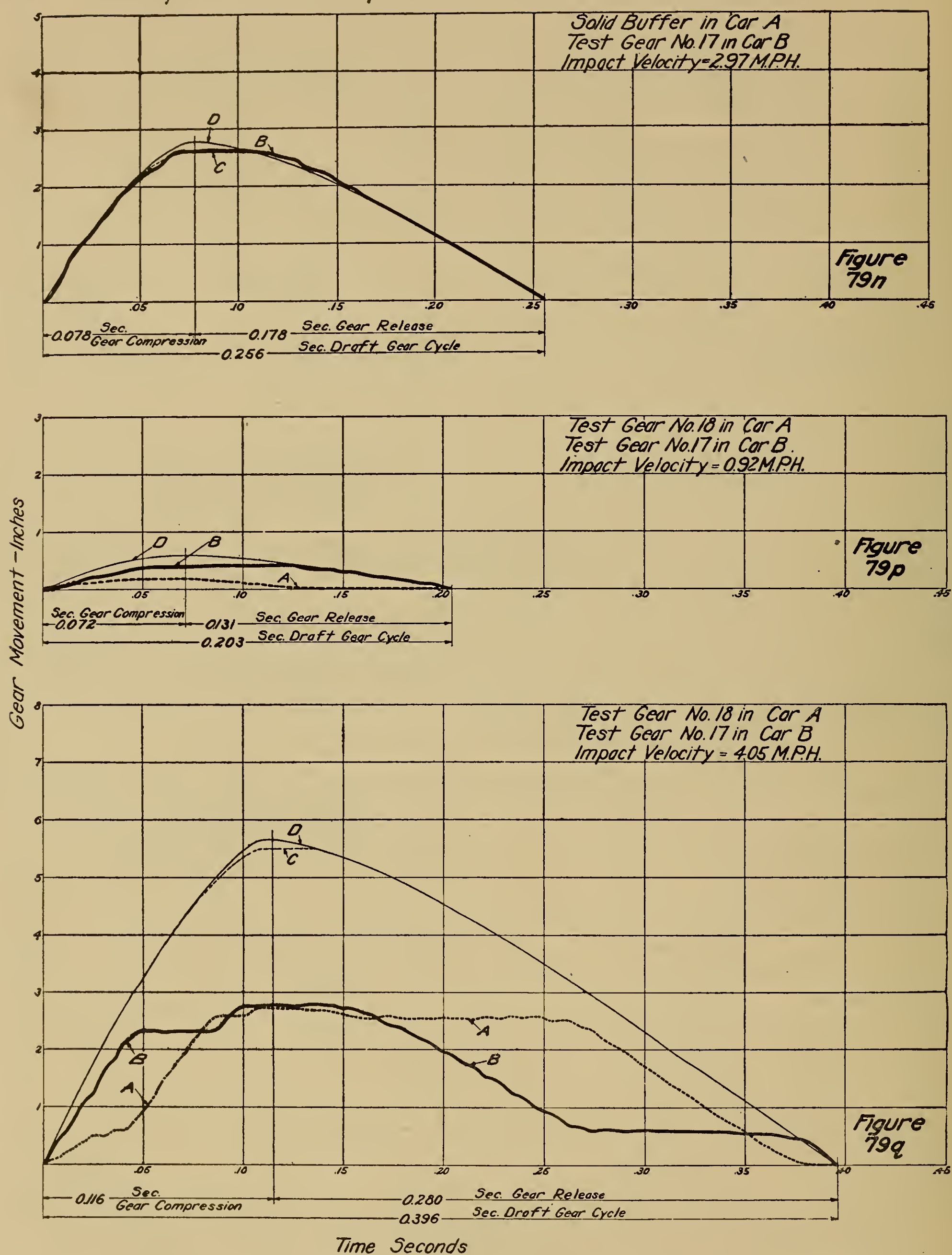
Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft gear cycle.

Figs. 79g, 79h and 79j—Energy Curves, Cardwell G-25-A Gears



FIGS. 79K, 79L AND 79M—TIME-FORCE CURVES, CARDWELL G-25-A GEARS



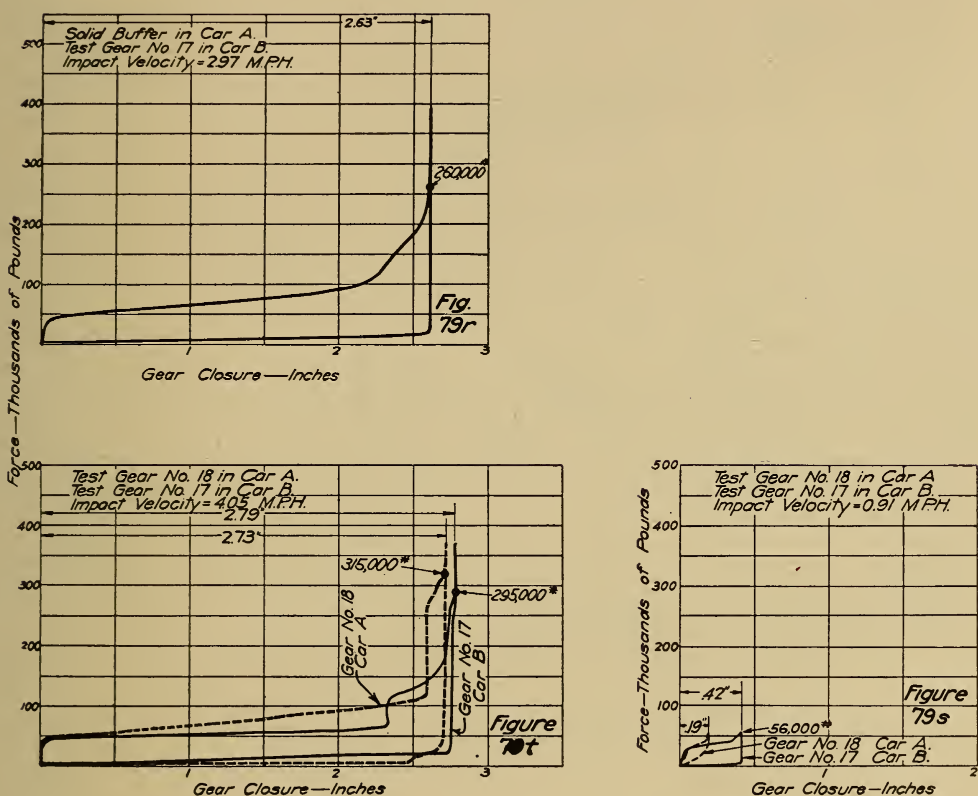
Curve D, determined from superimposed car-movement curves, represents combined draft gear movement and yield of car bodies.

Curve C, obtained by eliminating car body yield from curve D, represents true combined movement of both gears.

Curve B, traced on small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

Figs. 79n, 79p and 79q—Time-Closure Curves, Cardwell G-25-A Gears



Figs. 79r, 79s and 79t—Force-Closure Diagrams, Cardwell G-25-A Gears

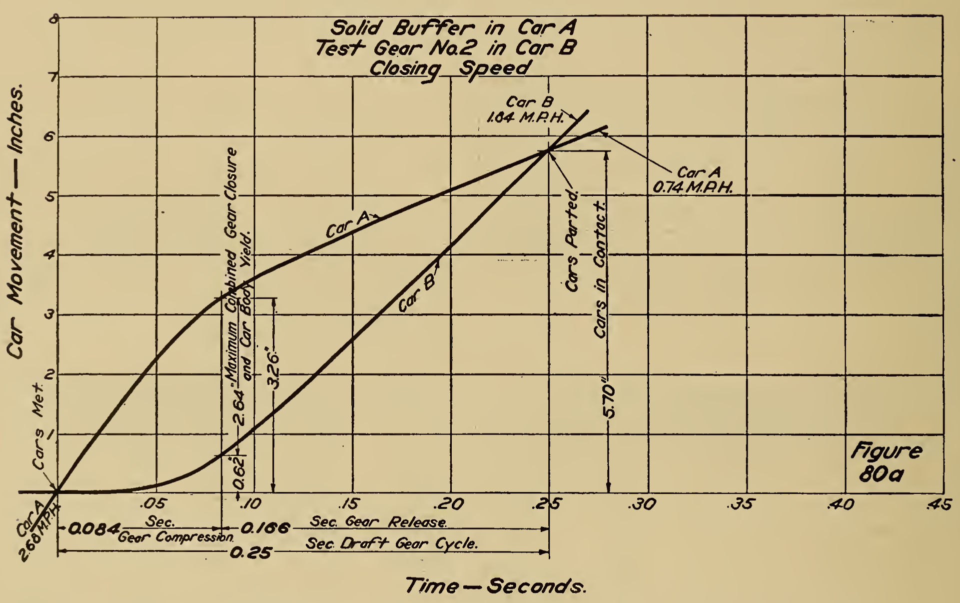
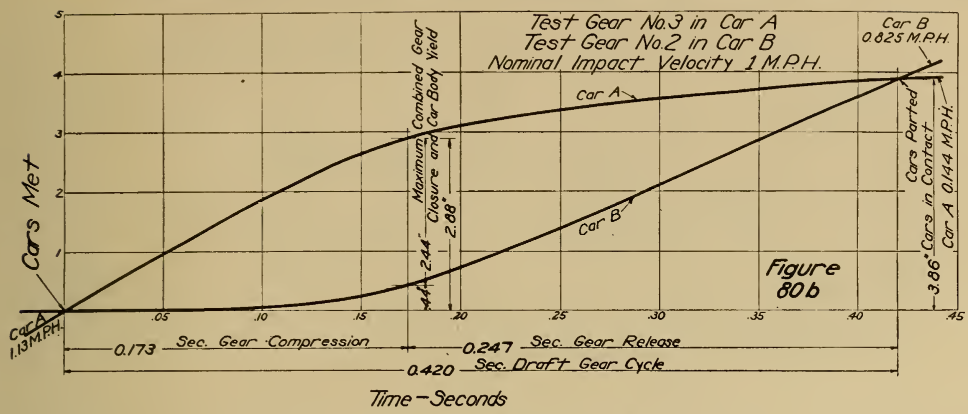
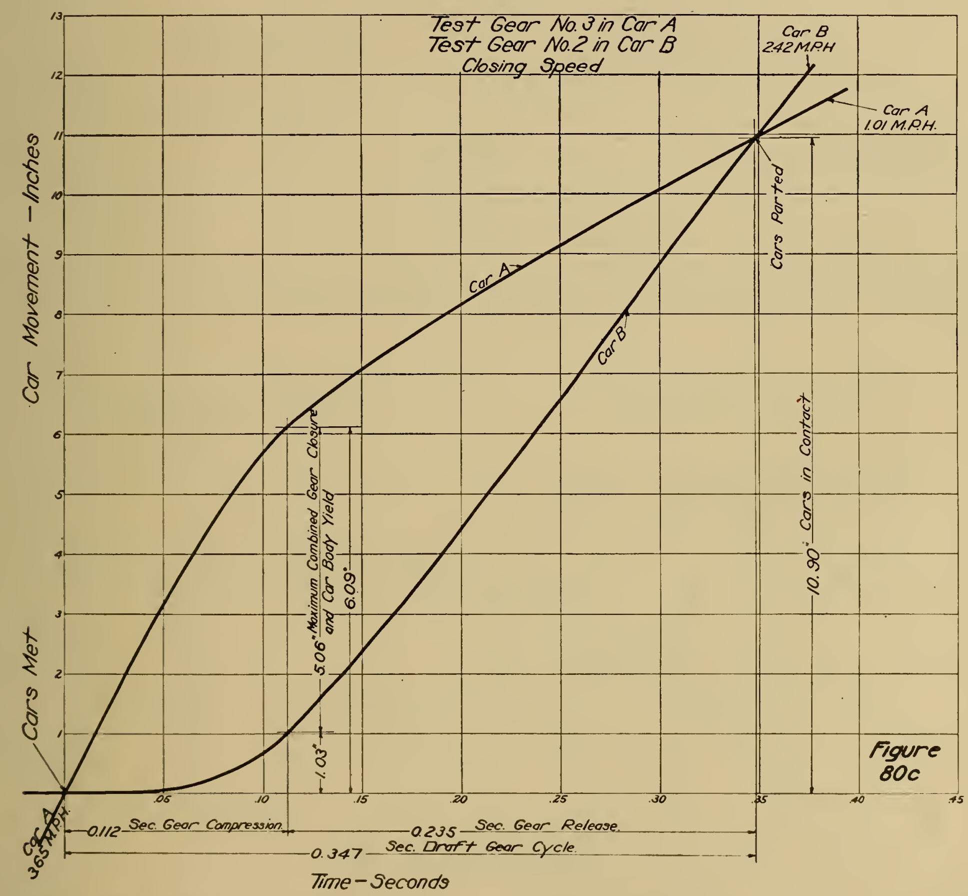
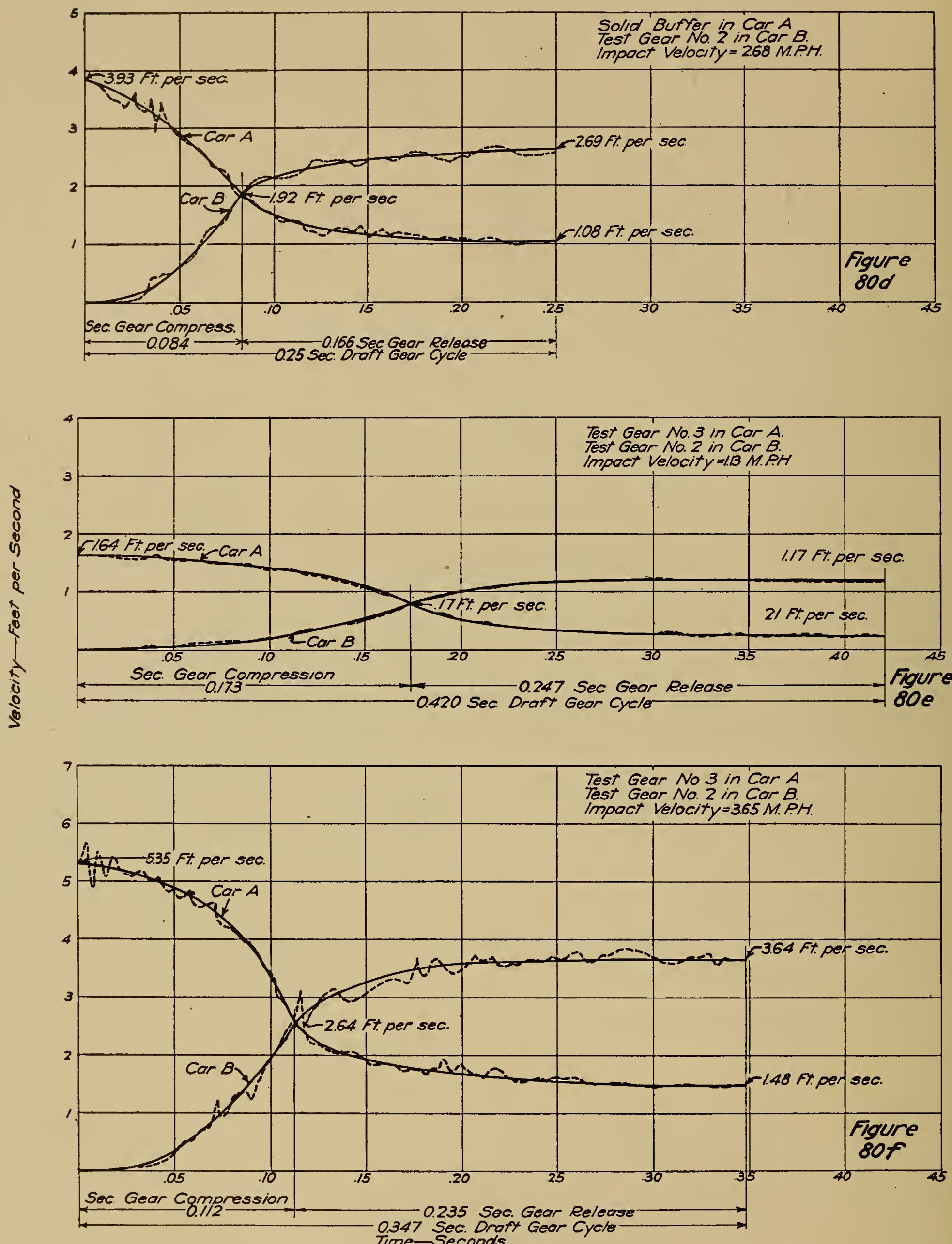


FIG. 80A—CAR-MOVEMENT CURVES, SUPERIMPOSED. WESTINGHOUSE D-3 GEARS THESE CURVES DRAWN BY CARS IN TESTS





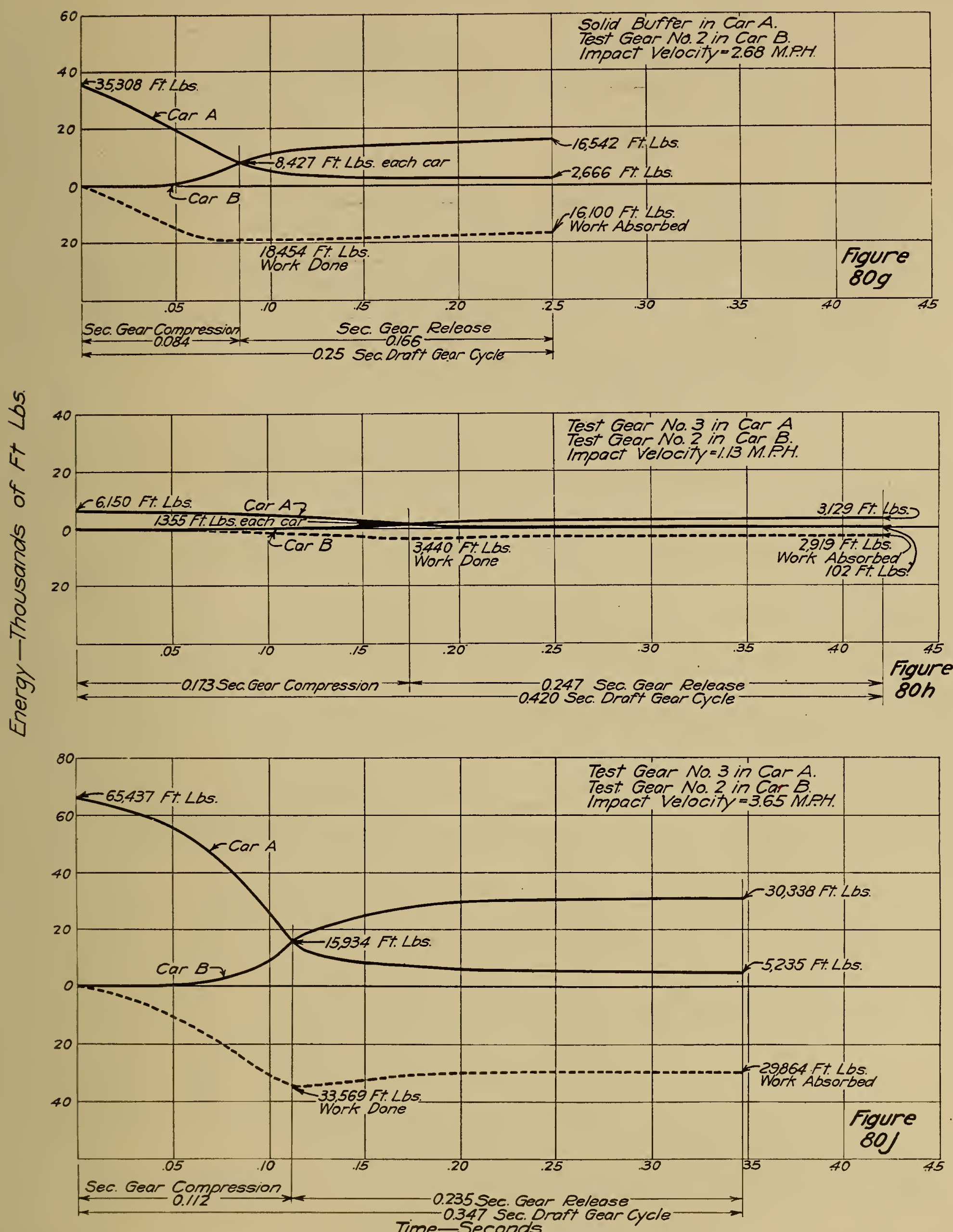
Figs. 80b and 80c—Car-Movement Curves, Superimposed. Westinghouse D-3 Gears These Curves Drawn by Cars in Tests



Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

The irregularities are due in general to vibrations of the car structure induced by draft gear action.
Full lines represent the mean velocity curves.

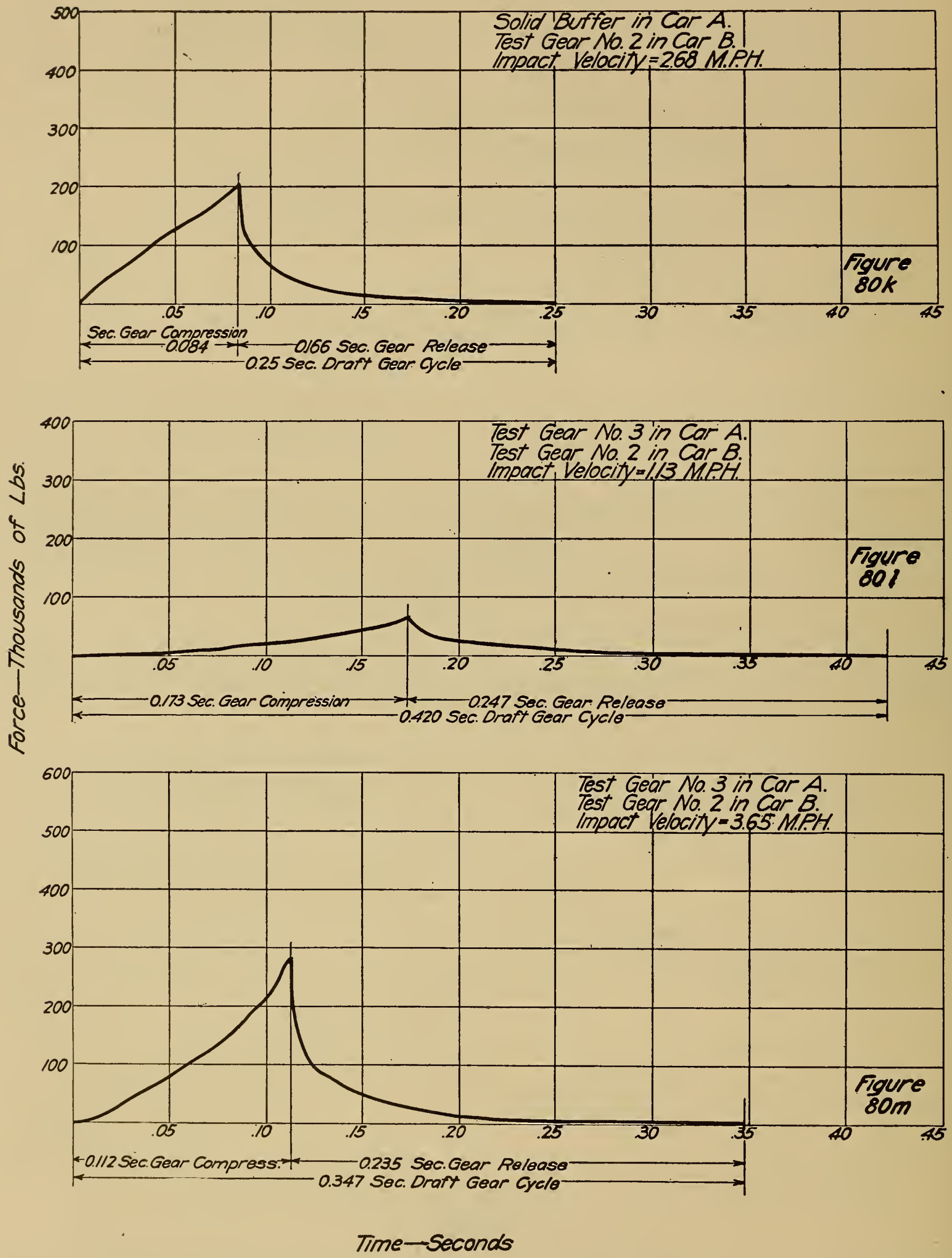
Figs. 80d, 80e and 80f—Velocity Curves, Westinghouse D-3 Gears



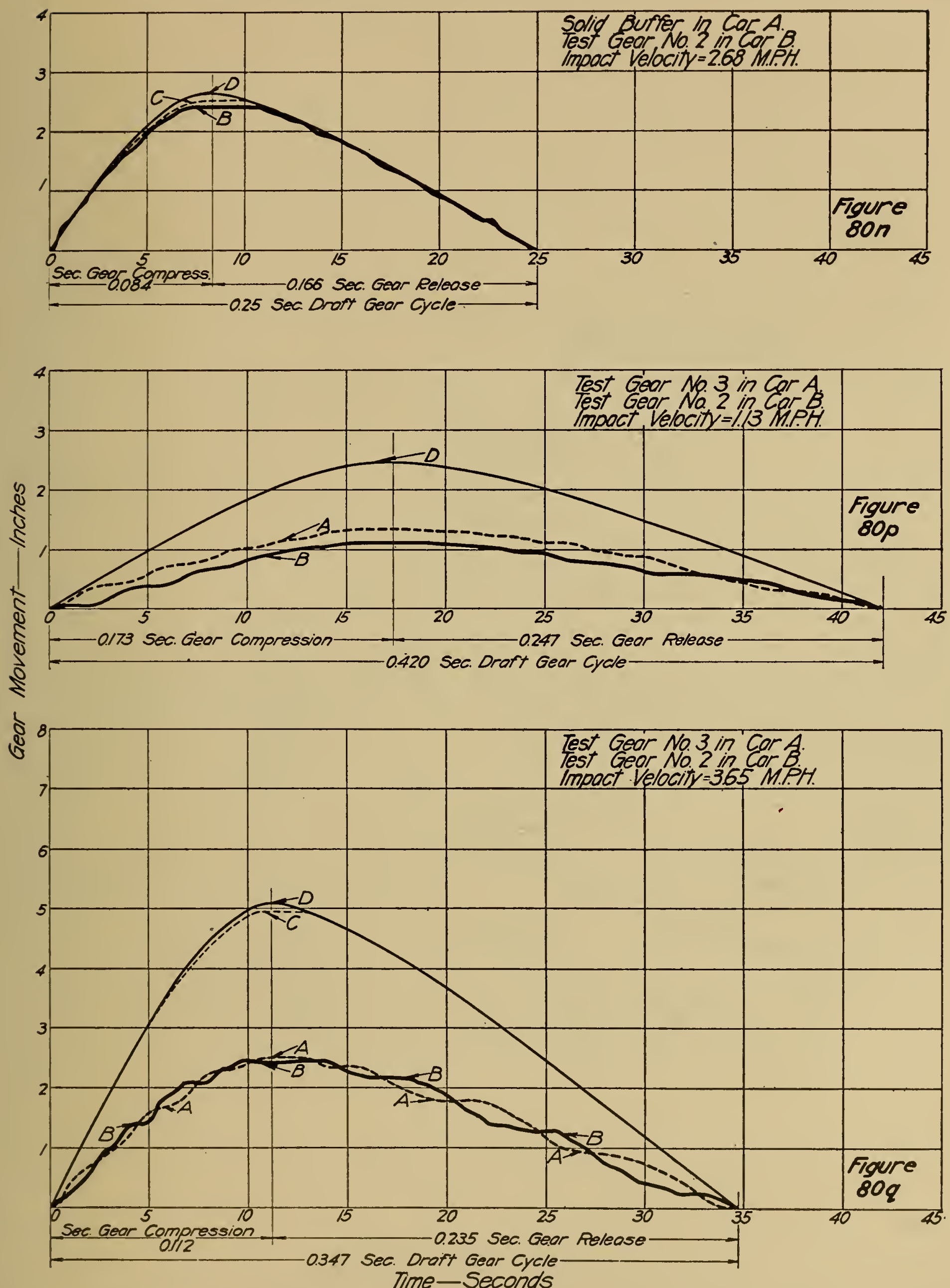
Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft gear cycle.

FIGS. 80G, 80H AND 80J—ENERGY CURVES, WESTINGHOUSE D-3 GEARS



FIGS. 80K, 80L AND 80M—TIME-FORCE CURVES, WESTINGHOUSE D-3 GEARS



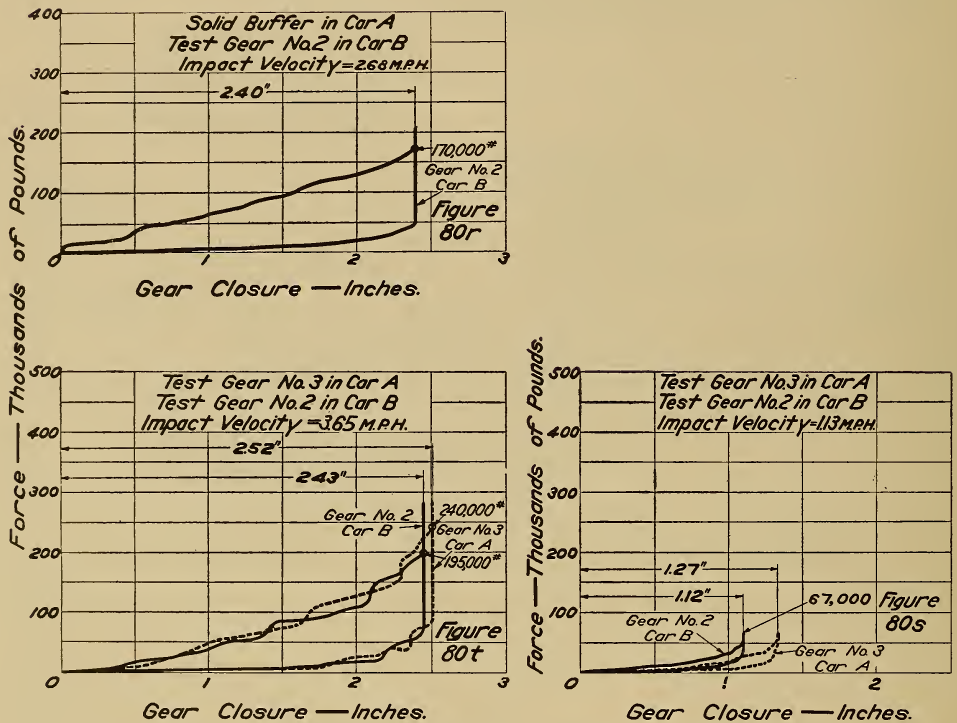
Curve D, determined from superimposed car-movement curves, represents combined draft gear movement and yield of car bodies.

Curve C, obtained by eliminating car body yield from curve D, represents true combined movement of both gears.

Curve B, traced on small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

Figs. 80n, 80p and 80q—Time-Closure Curves, Westinghouse D-3 Gears



Figs. 80r, 80s and 80t—Force-Closure Diagrams, Westinghouse D-3 Gears

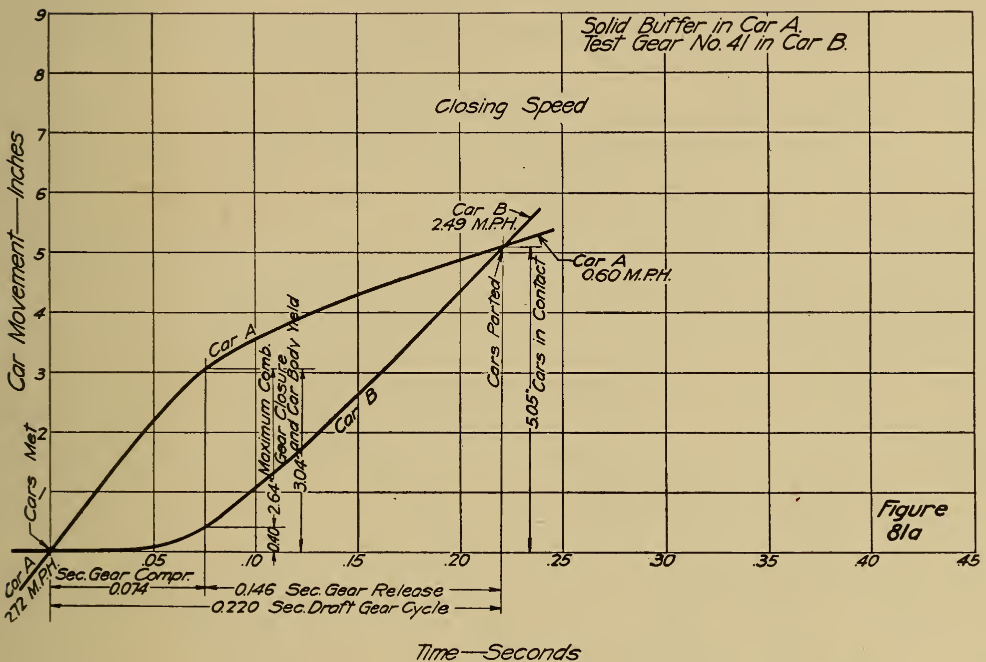
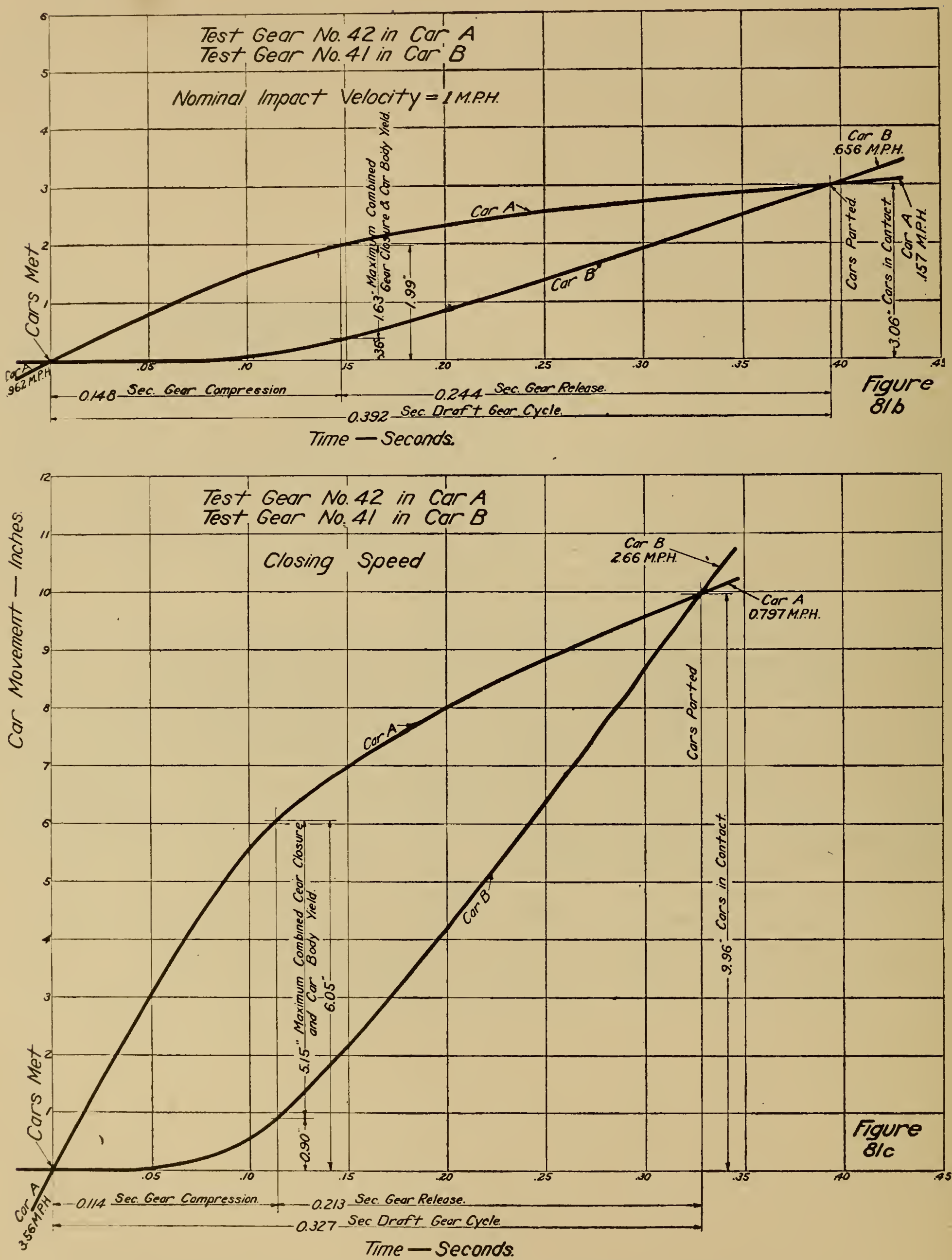
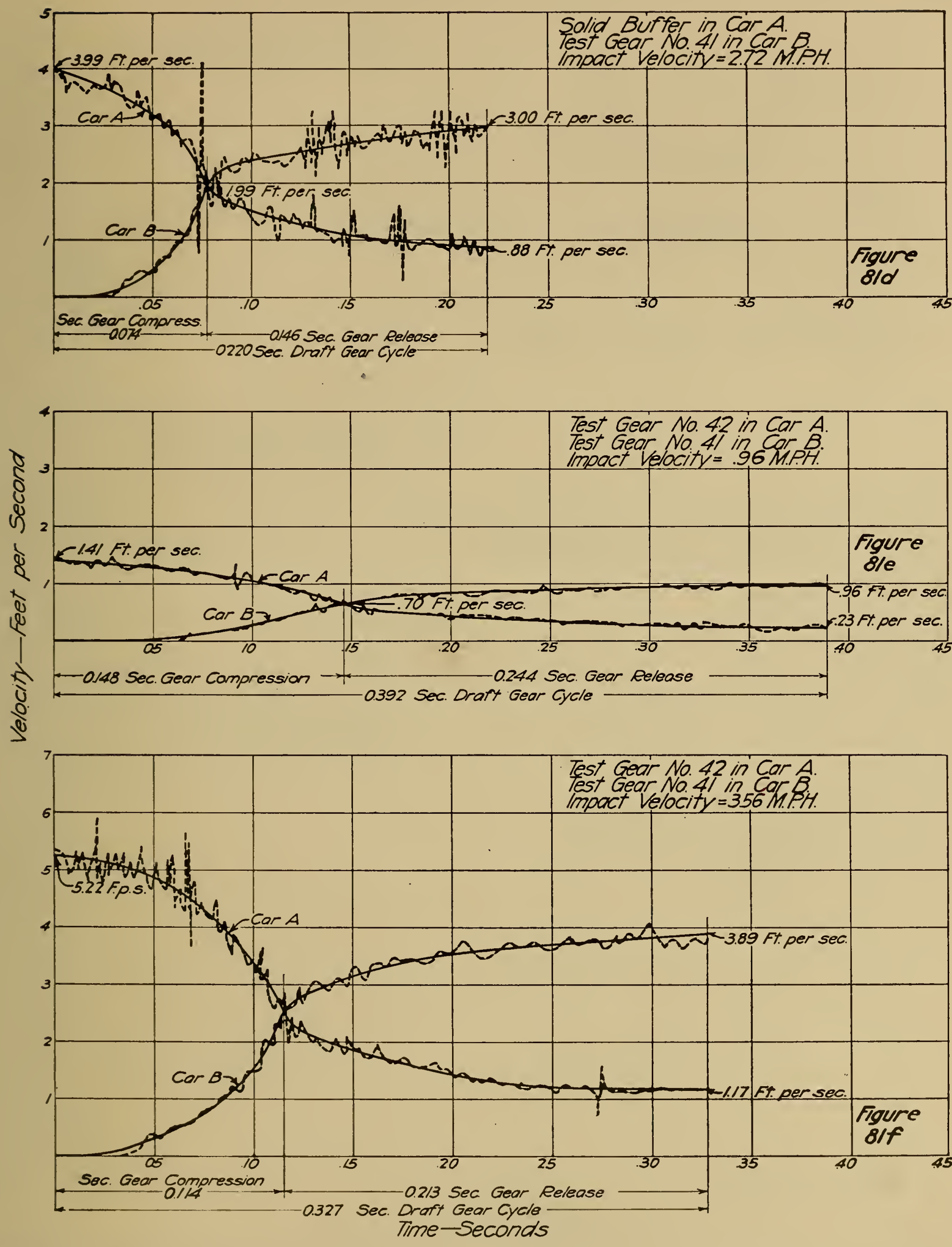


Fig. 81a—Car-Movement Curves, Superimposed, Gould No. 175 Gears
These Curves Drawn by Cars in Test



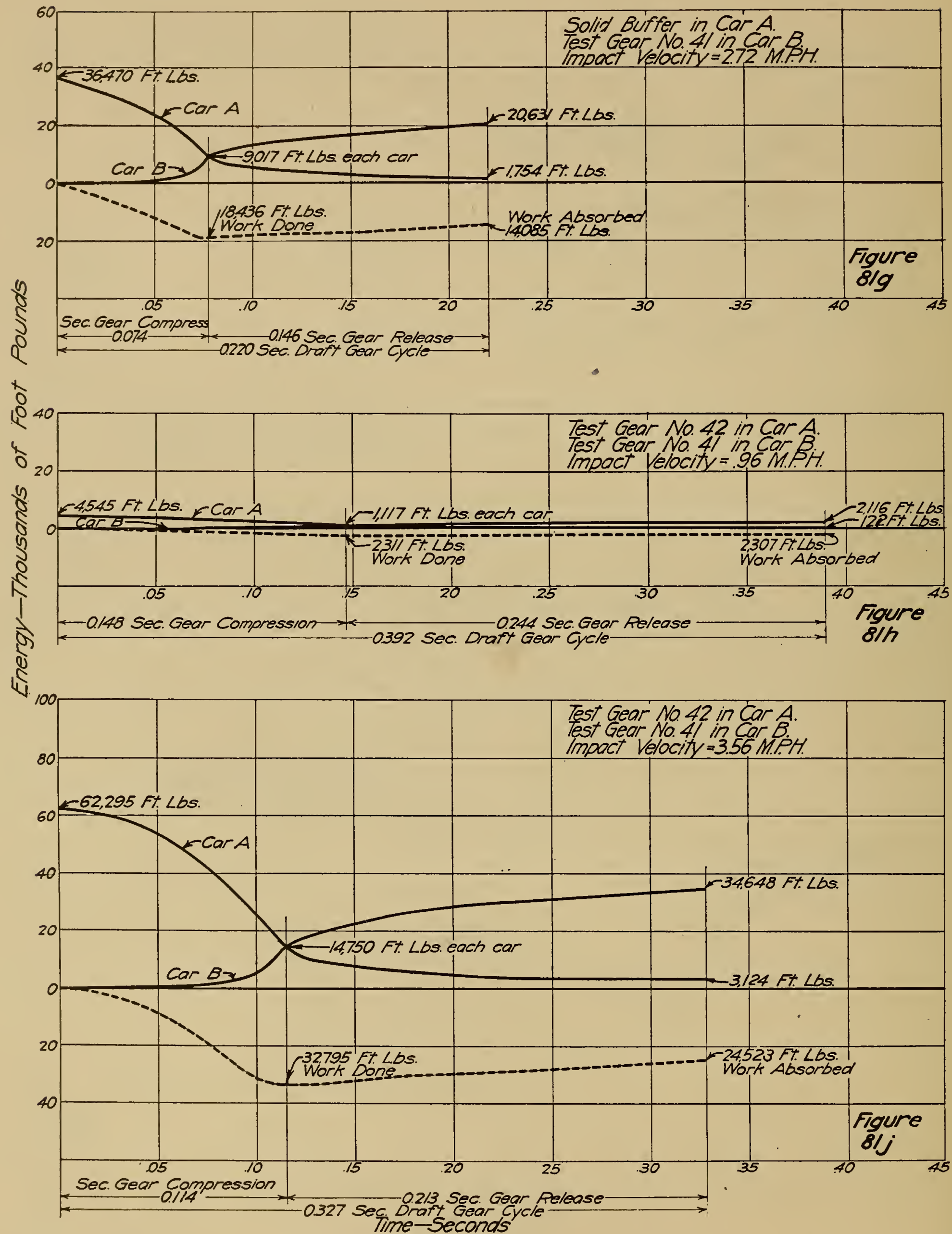
Figs. 81b and 81c—Car-Movement Curves, Superimposed. Gould No. 175 Gears
These Curves Drawn by Cars in Test



Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

The irregularities are due in general to vibrations of the car structure induced by draft gear action.
Full lines represent the mean velocity curves.

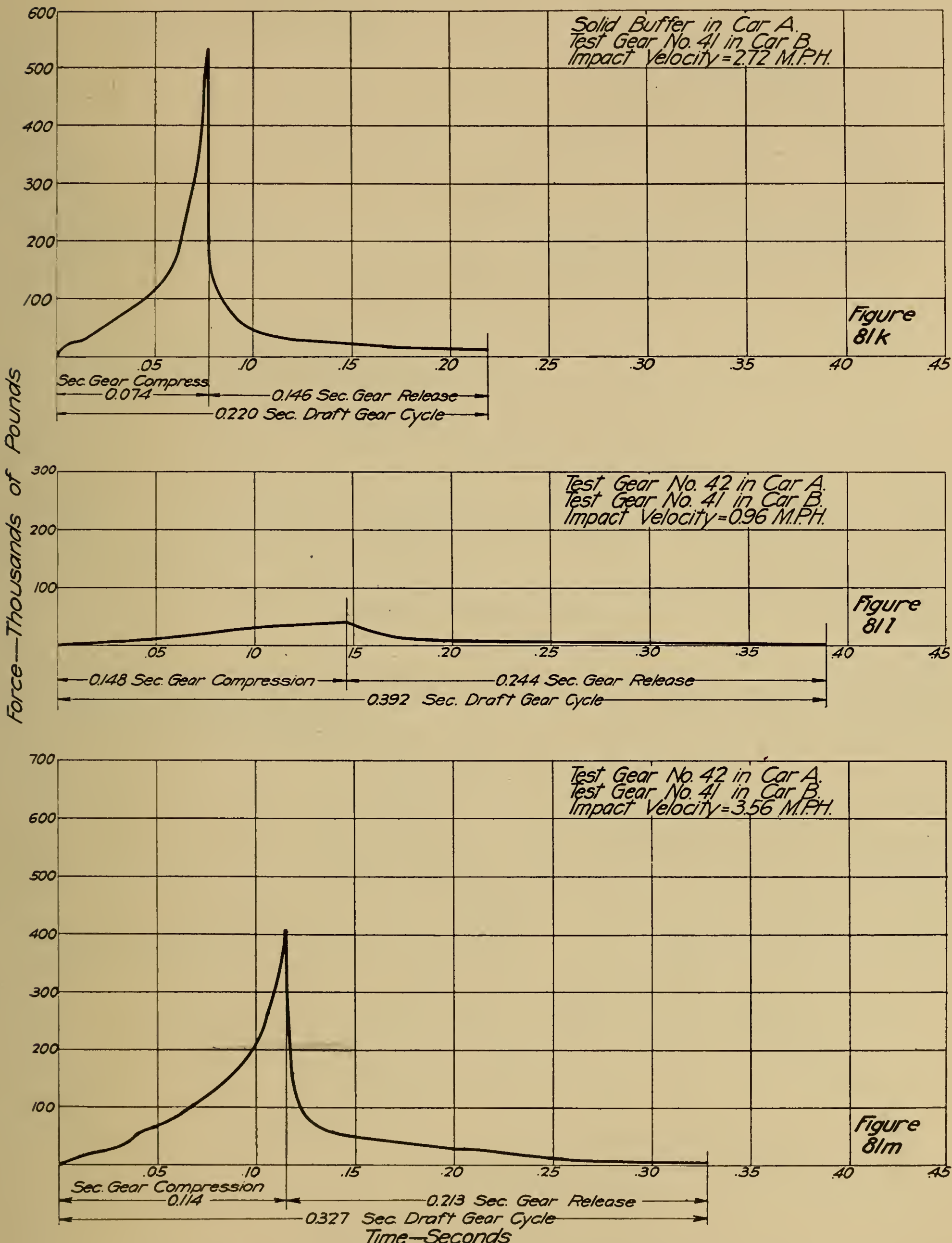
FIGS. 81D, 81E AND 81F—VELOCITY CURVES, GOULD NO. 175 GEARS



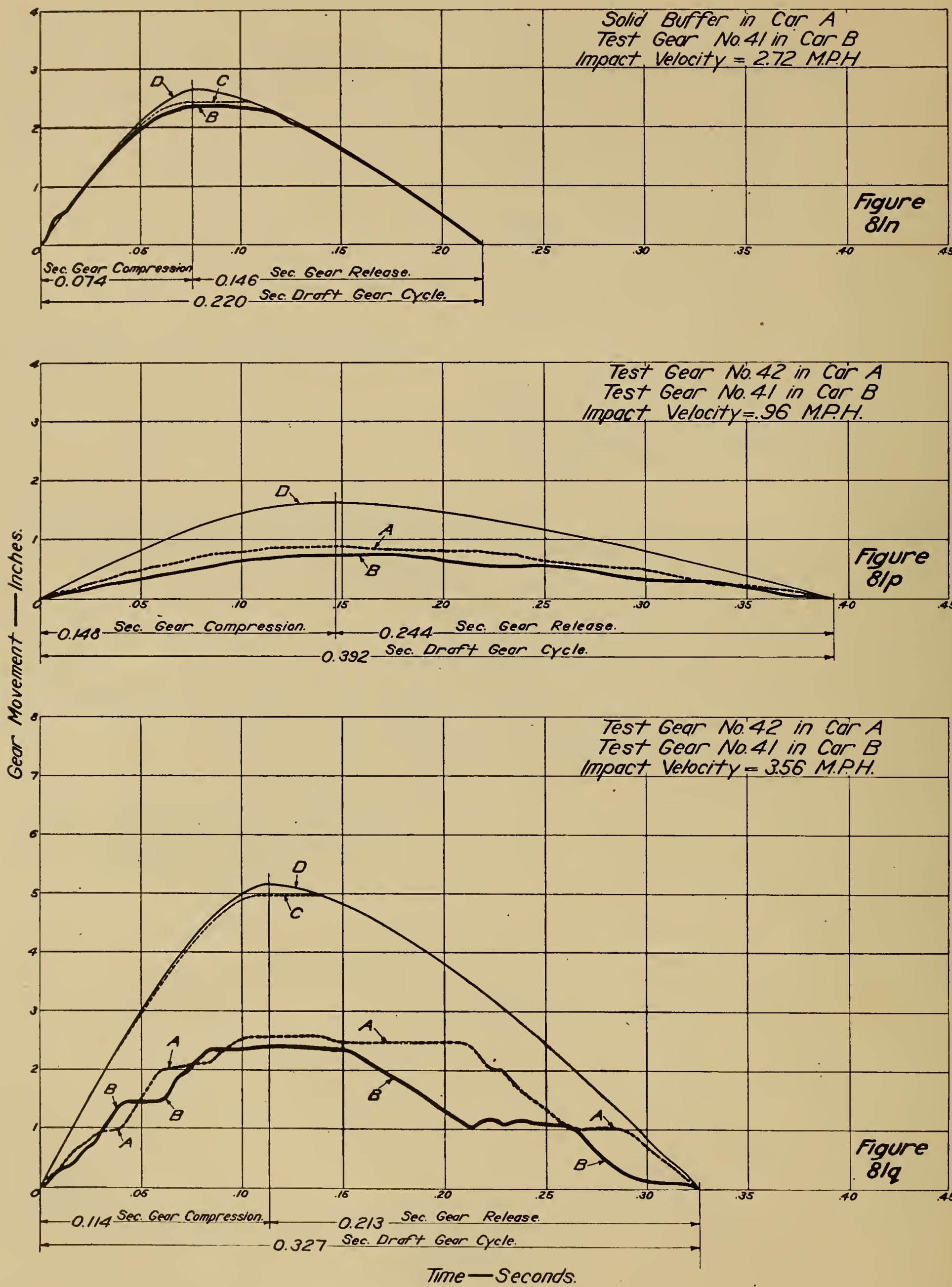
Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft gear cycle.

FIGS. 81G, 81H AND 81J—ENERGY CURVES, GOULD NO. 175 GEARS



Figs. 81k, 81l and 81m—Time-Force Curves, Gould No. 175 Gears



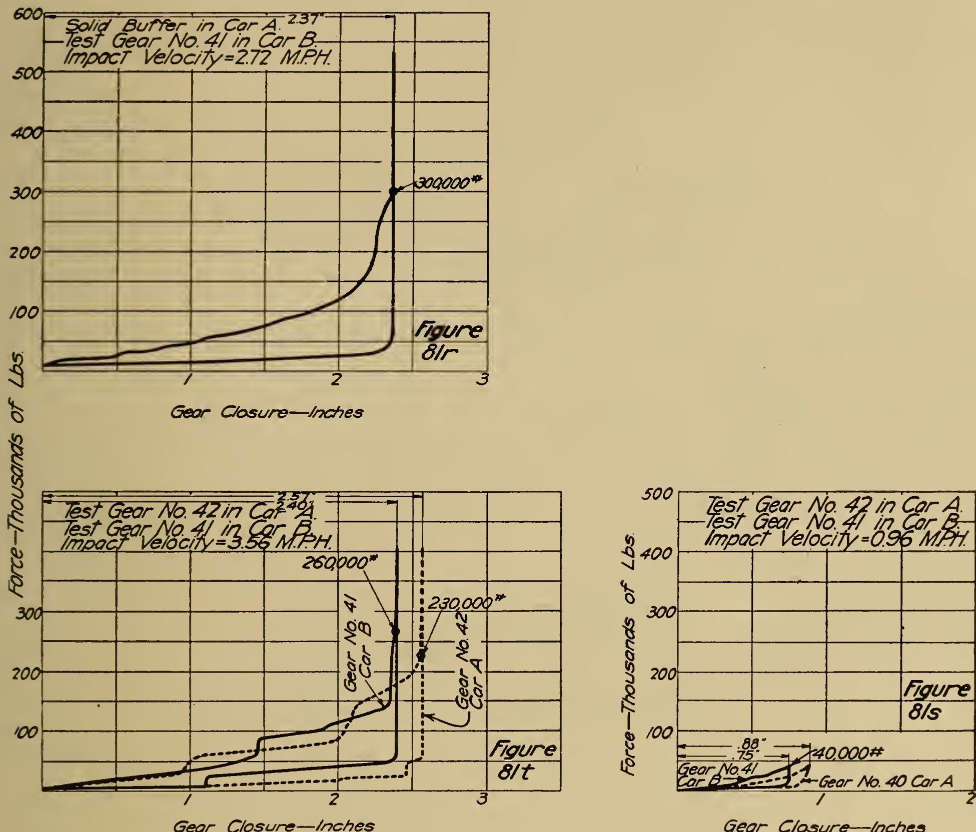
Curve D, determined from superimposed car-movement curves, represents combined draft gear movement and yield of car bodies.

Curve C, obtained by eliminating car body yield from curve D, represents true combined movement of both gears.

Curve B, traced on small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

Figs. 81n, 81p and 81q—Time-Closure Curves, Gould No. 175 Gears



Figs. 81r, 81s and 81t—Force-Closure Diagrams, Gould No. 175 Gears

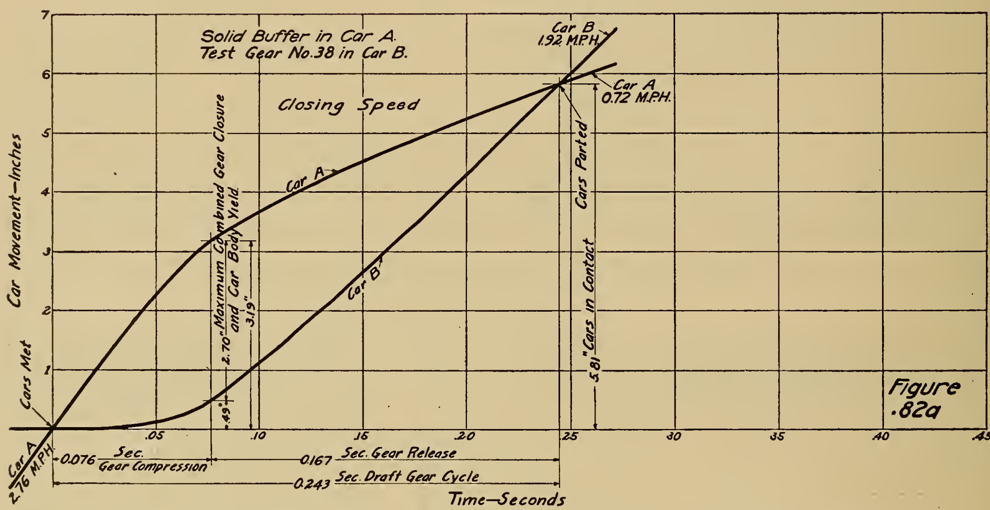
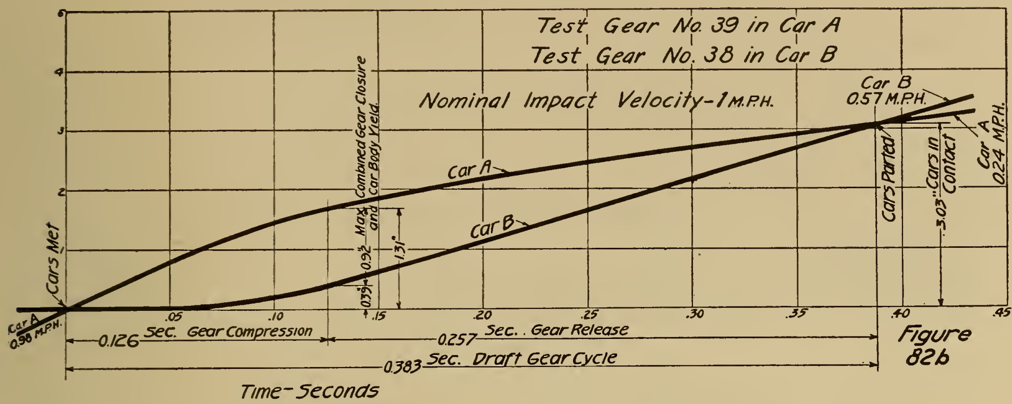
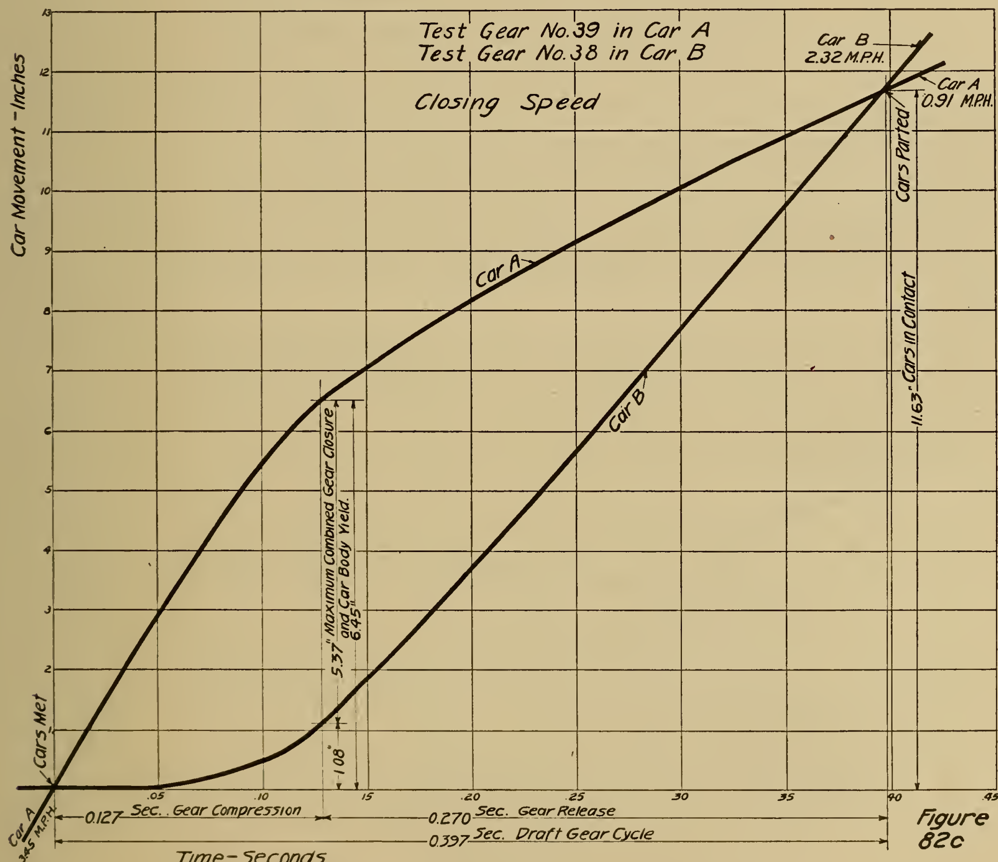
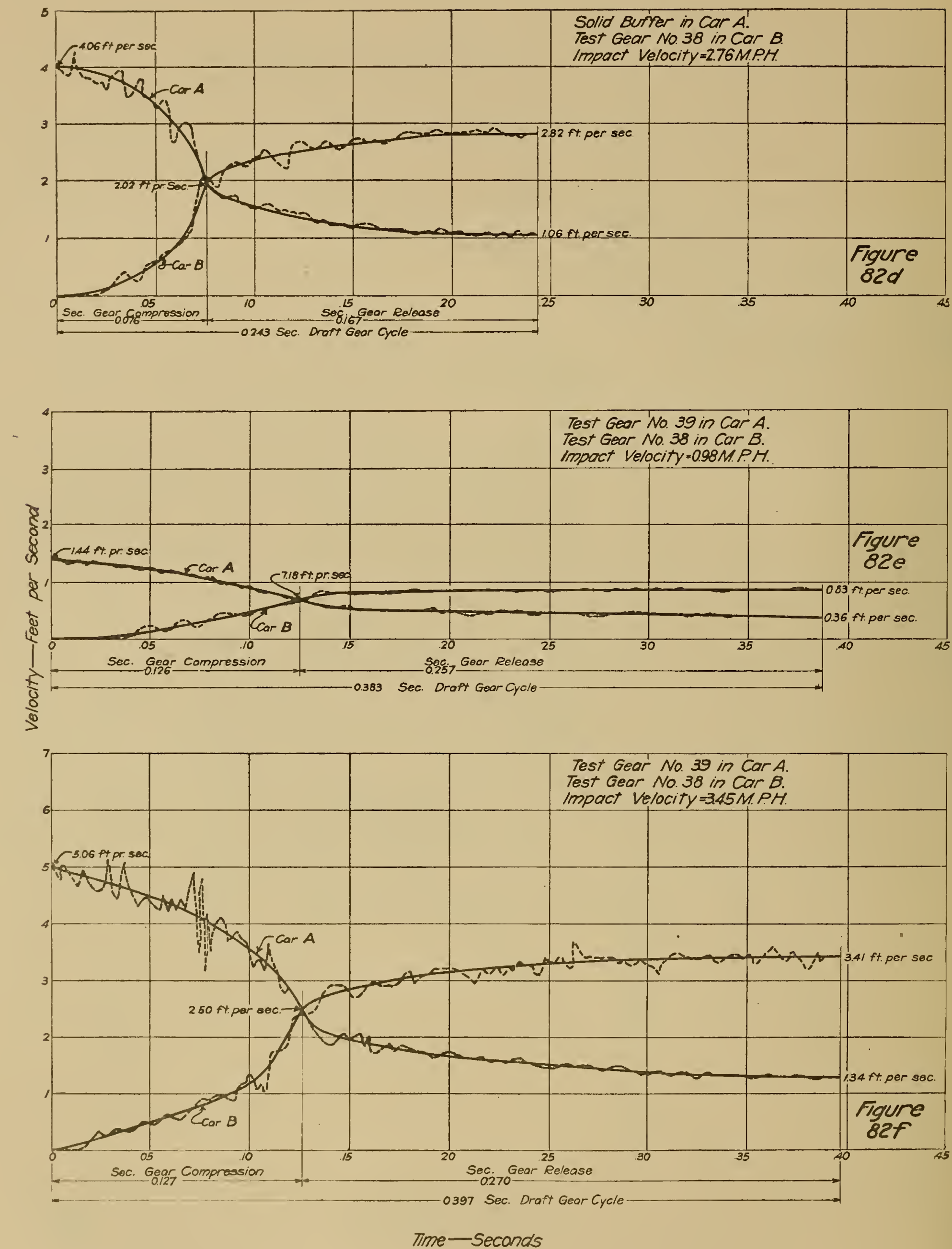


FIG. 82A—CAR-MOVEMENT CURVES, SUPERIMPOSED. MURRAY H-25 GEARS
THESE CURVES DRAWN BY CARS IN TESTS





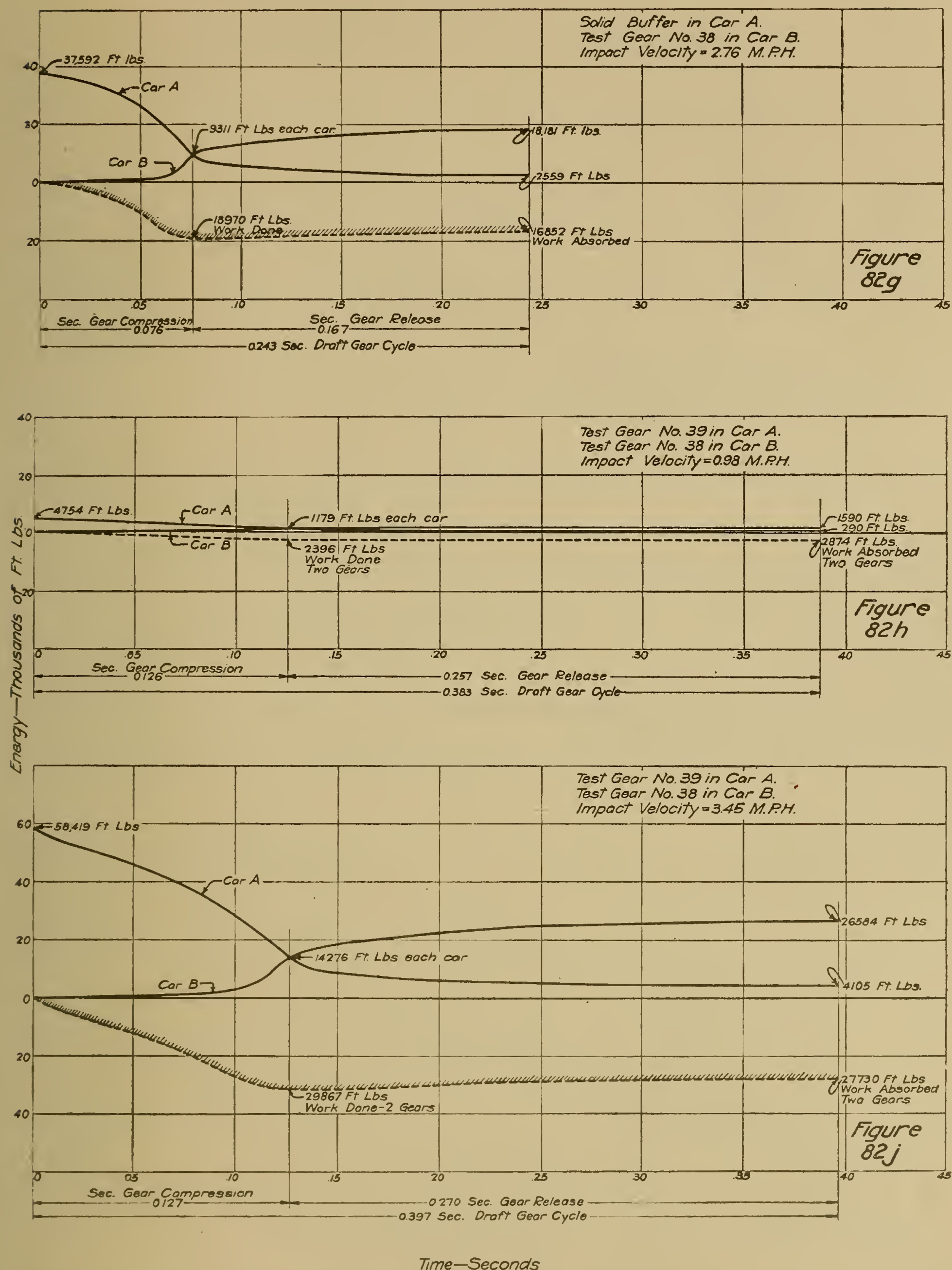
Figs. 82b and 82c—Car-Movement Curves, Superimposed. Murray H-25 Gears
These Curves Drawn by Cars in Tests



Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

The irregularities are due in general to vibrations of the the car structure induced by draft gear action.
Full lines represent the mean velocity curves.

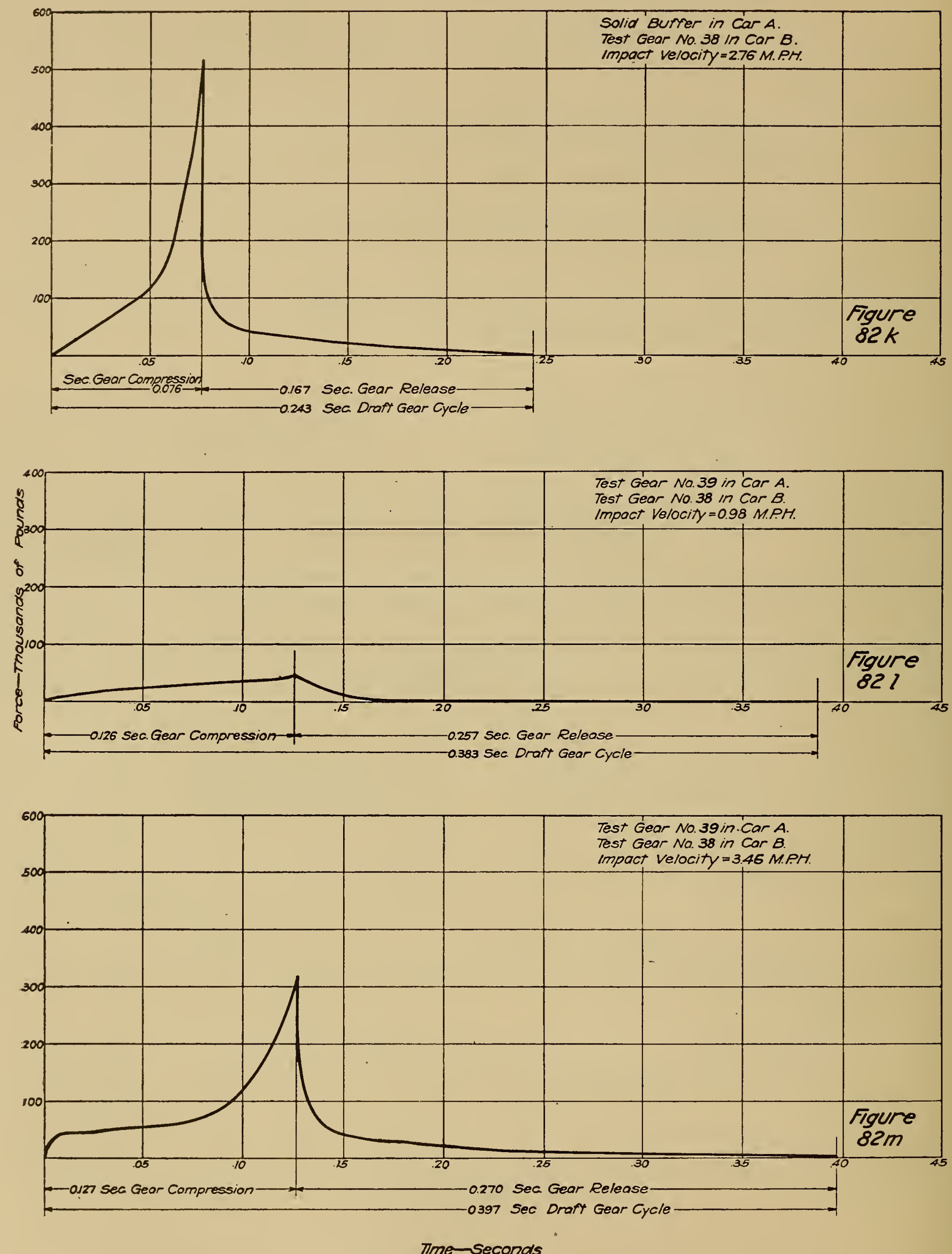
FIGS. 82D, 82E AND 82F—VELOCITY CURVES, MURRAY H-25 GEARS



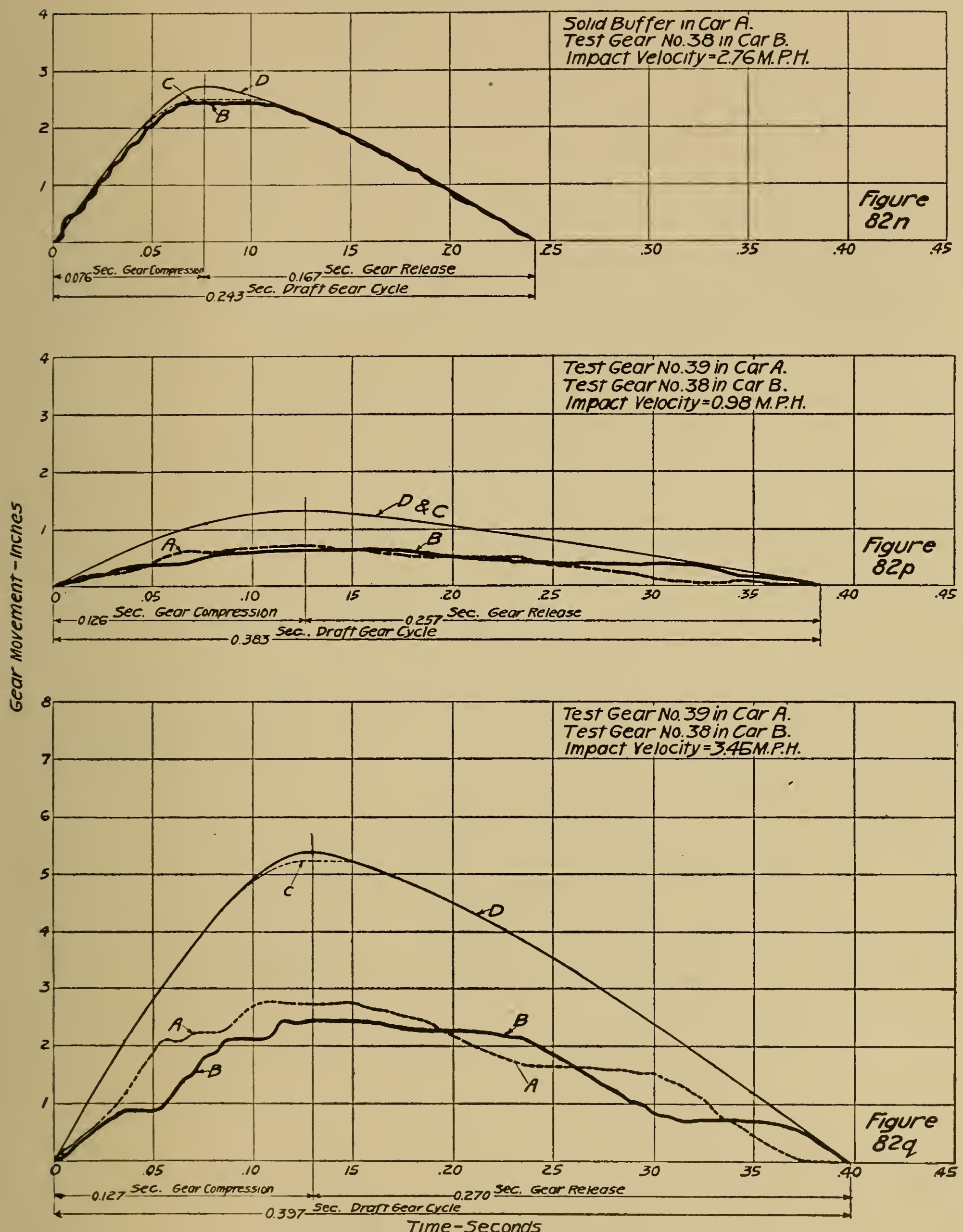
Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft gear cycle.

FIGS. 82G, 82H AND 82J—ENERGY CURVES, MURRAY H-25 GEARS



Figs. 82k, 82l and 82m—Time-Force Curves, Murray H-25 Gears



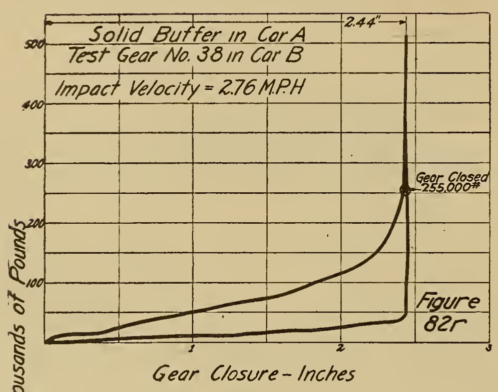
Curve D, determined from superimposed car-movement curves, represents combined draft gear movement and yield of car bodies.

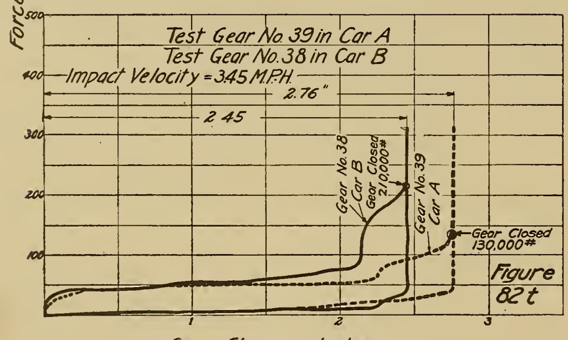
Curve C, obtained by eliminating car body yield from curve D, represents true combined movement of both gears.

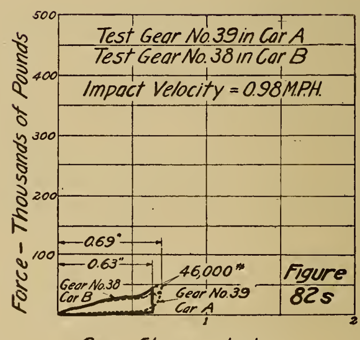
Curve B, traced on small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

Figs. 82n, 82p and 82q—Time-Closure Curves, Murray H-25 Gears







Figs. 82r, 82s and 82t—Force-Closure Diagrams, Murray H-25 Gears

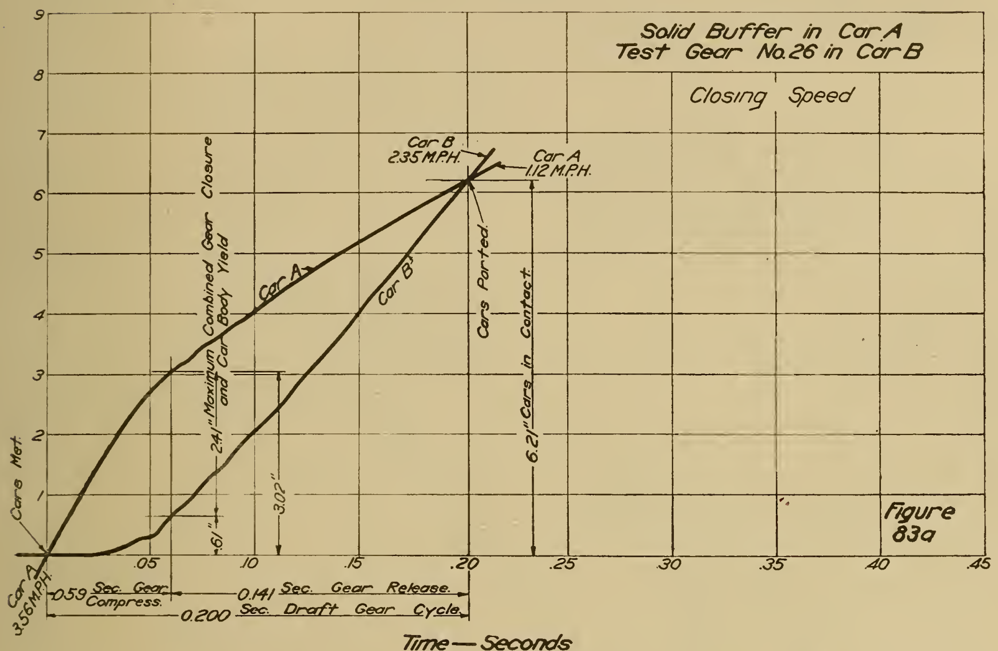
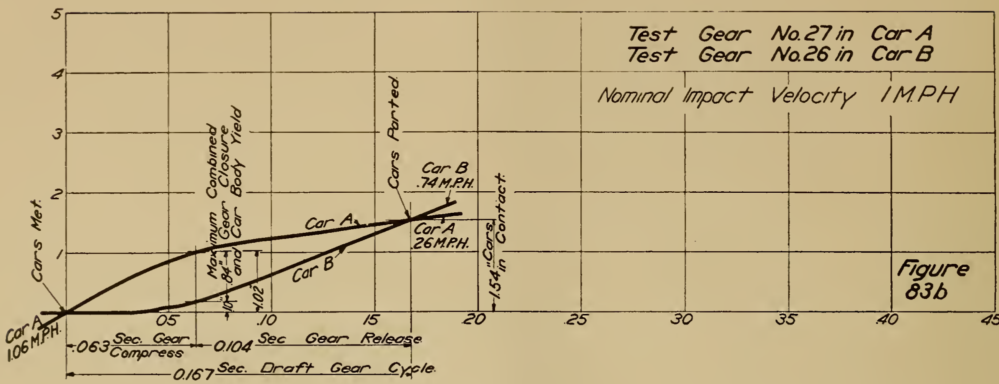
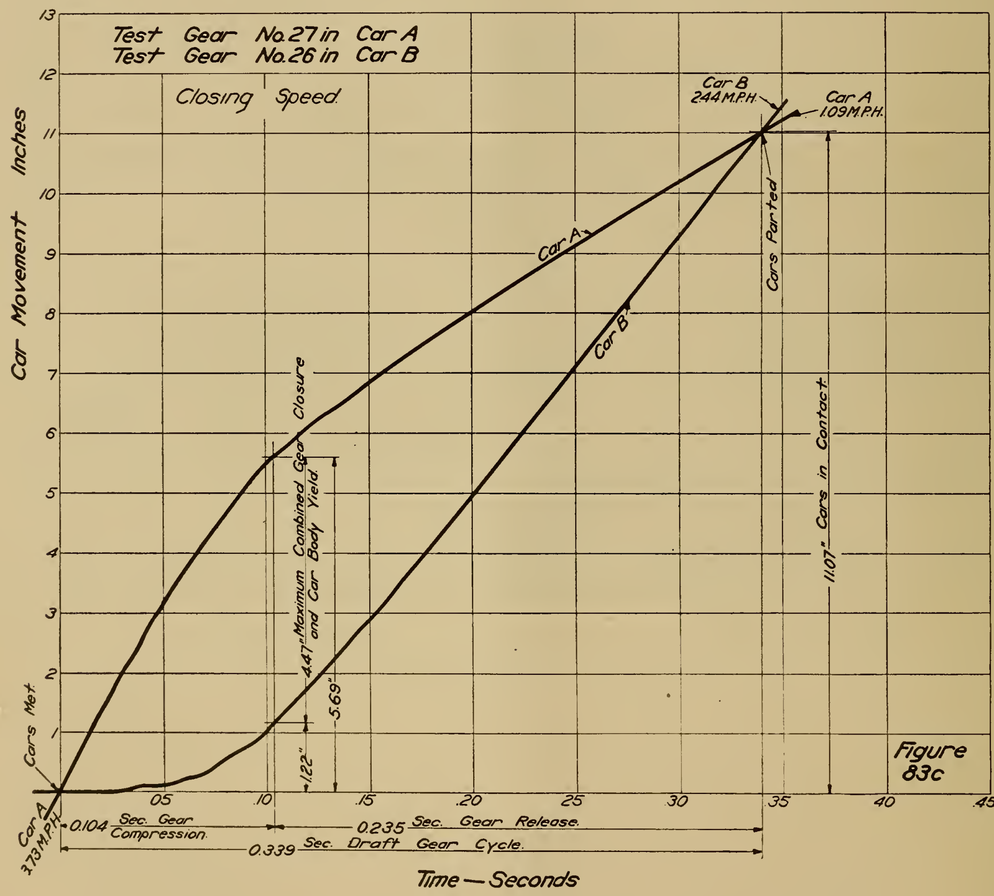
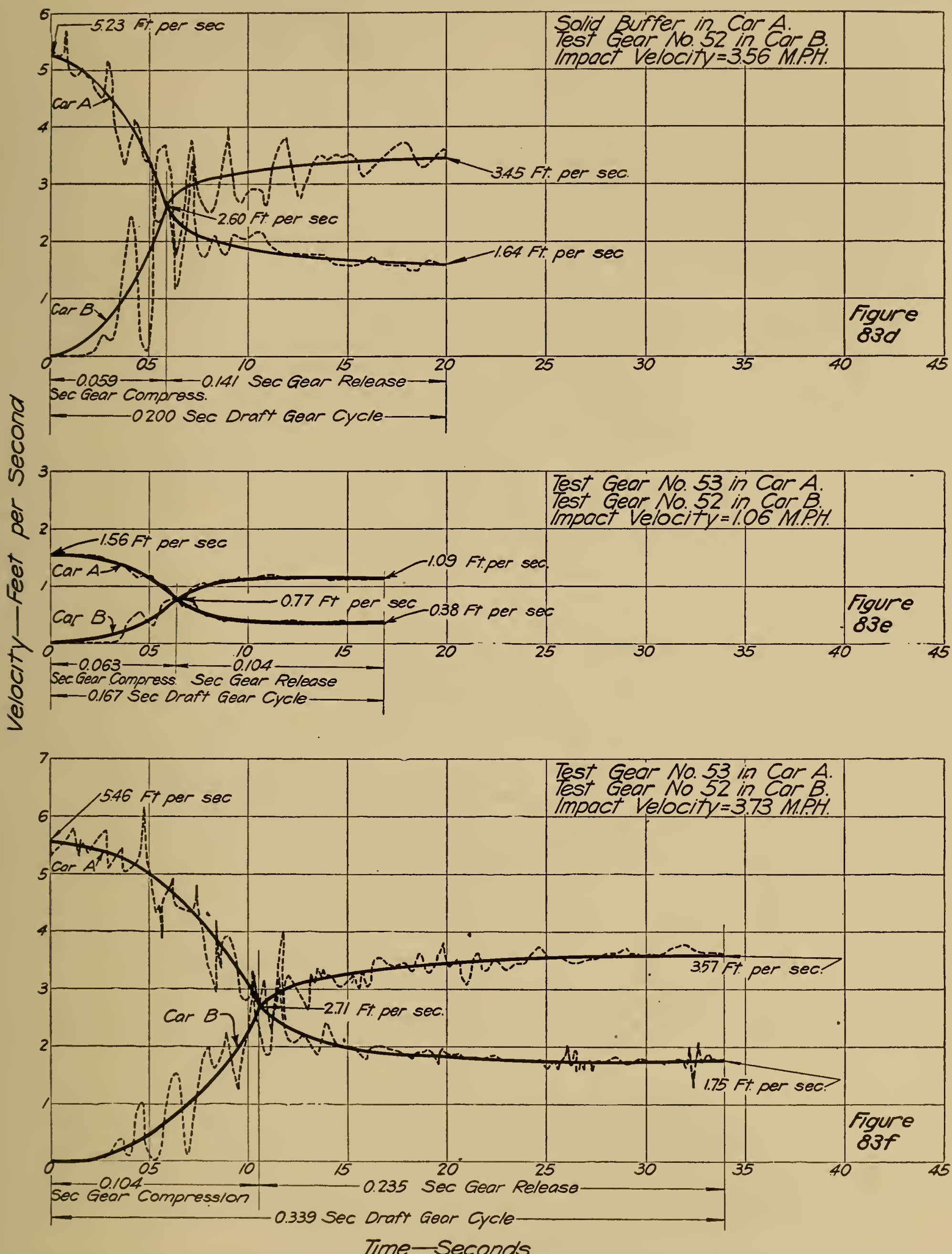


Fig. 83a—Car-Movement Curves, Superimposed. Christy Gears
These Curves Drawn by Cars in Tests





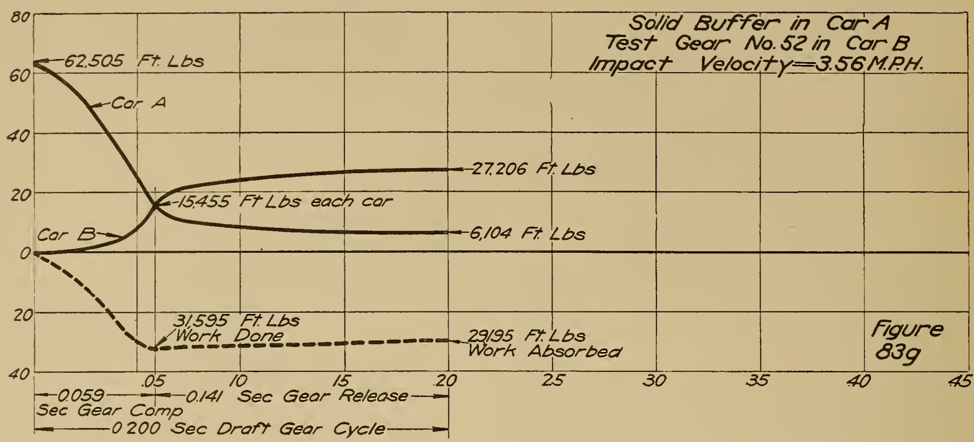
Figs. 83b and 83c—Car-Movement Curves, Superimposed. Christy Gears
These Curves Drawn by Cars in Tests

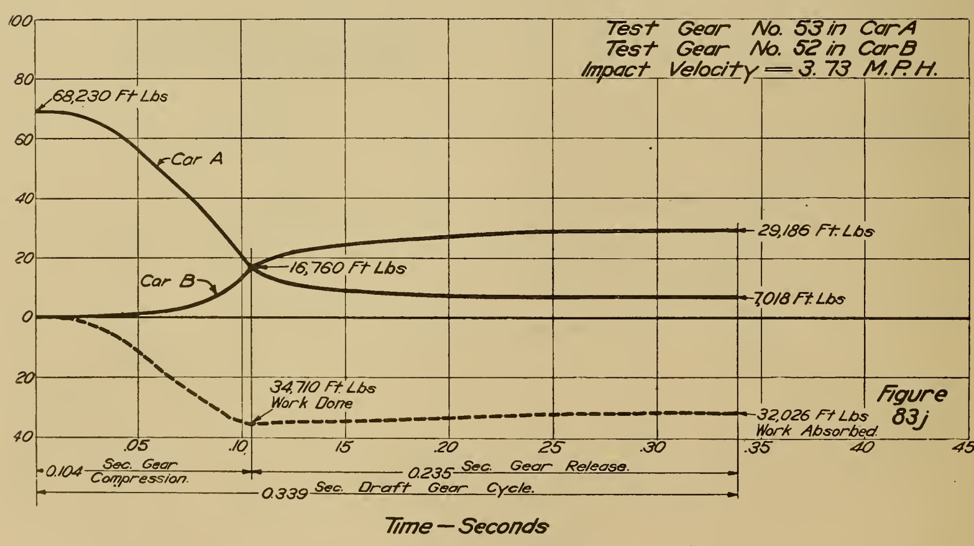


Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

The irregularities are due in general to vibrations of the car structure induced by draft gear action.
Full lines represent the mean velocity curves.

FIGS. 83D, 83E AND 83F—VELOCITY CURVES, CHRISTY GEARS

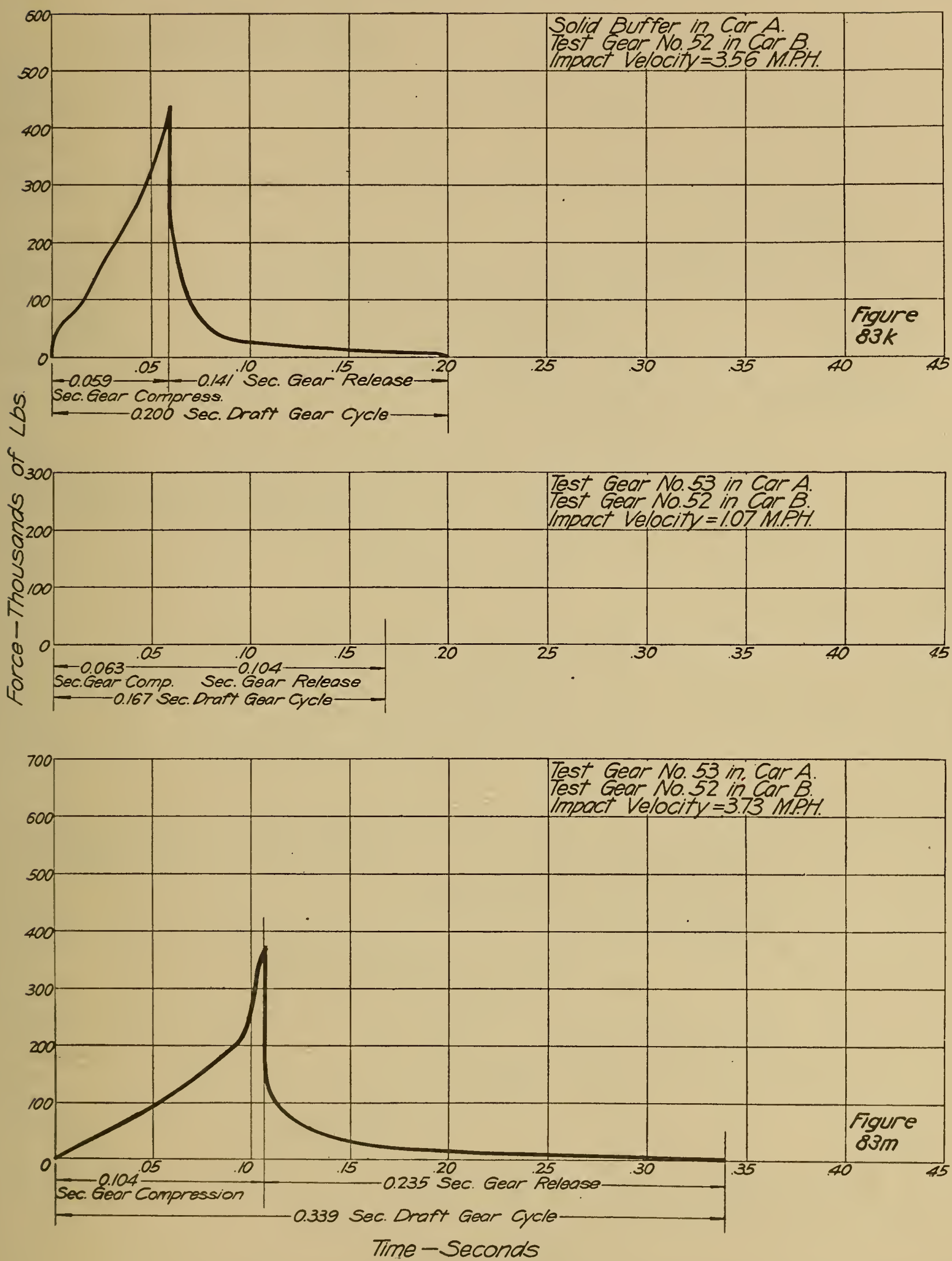




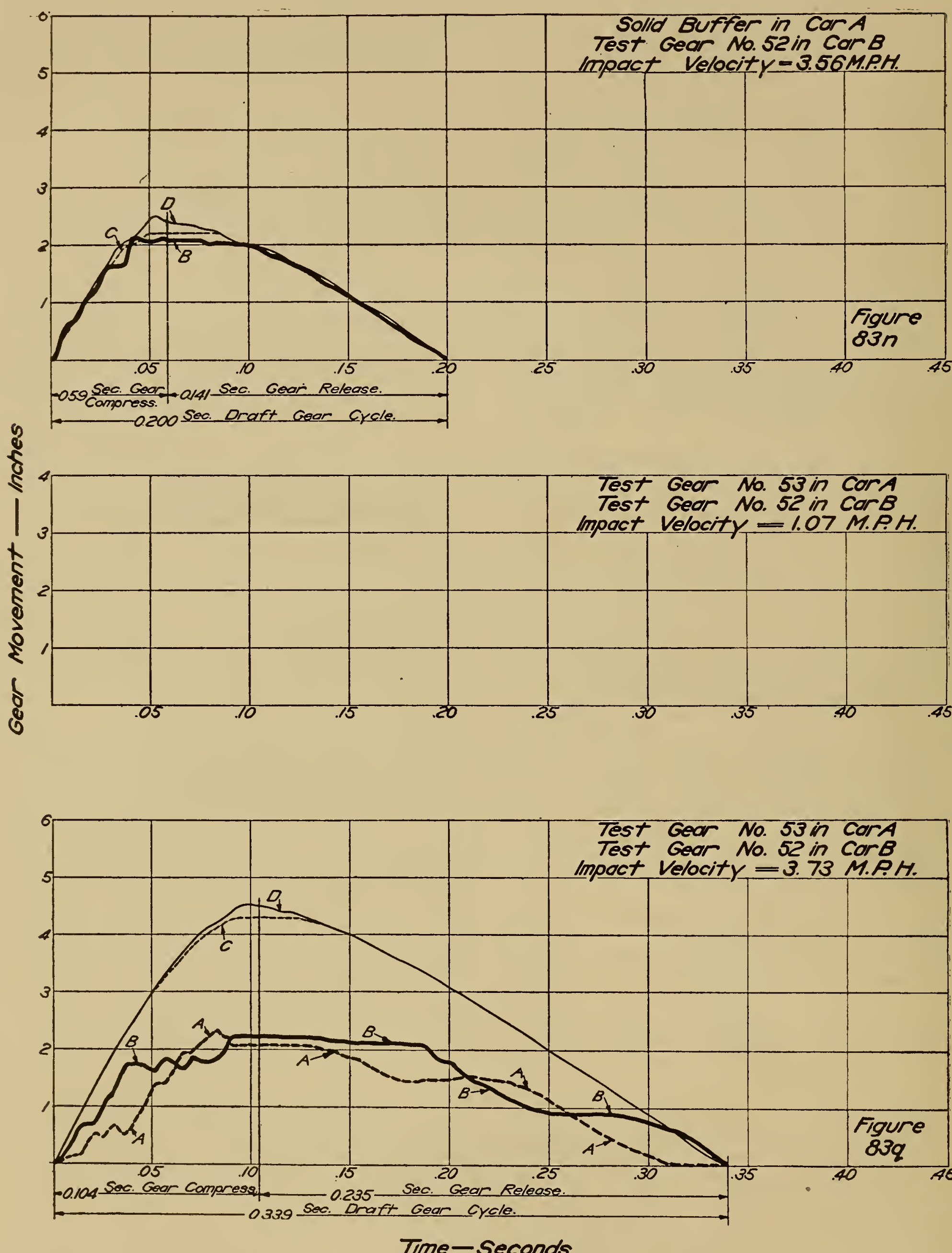
Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft gear cycle.

FIGS. 83G AND 83J—ENERGY CURVES, CHRISTY GEARS



FIGS. 83K AND 83M—TIME-FORCE CURVES, CHRISTY GEARS



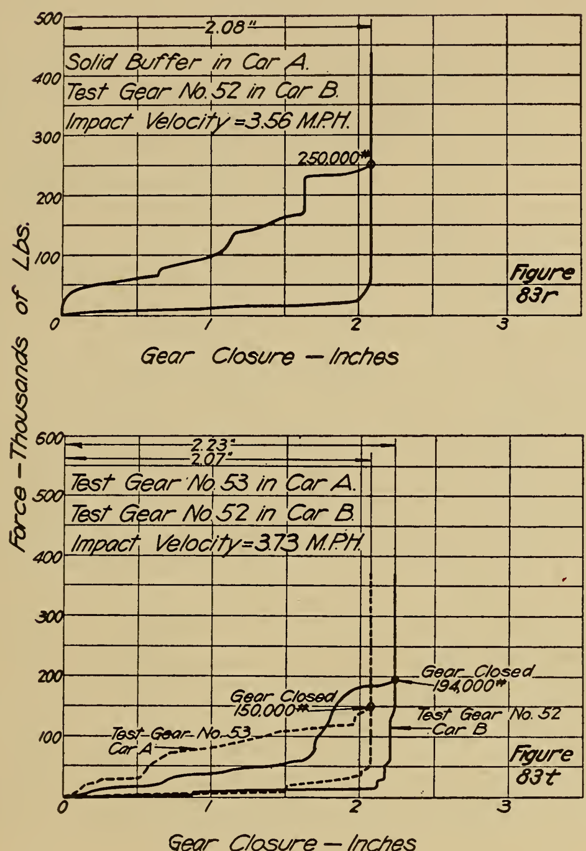
Curve D, determined from superimposed car-movement curves, represents combined draft gear movement and yield of car bodies.

Curve C, obtained by eliminating car body yield from curve D, represents true combined movement of both gears.

Curve B, traced on small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

FIGS. 83N AND 83Q—TIME-CLOSURE CURVES, CHRISTY GEARS



Figs. 83r and 83t—Force-Closure Diagrams, Christy Gears

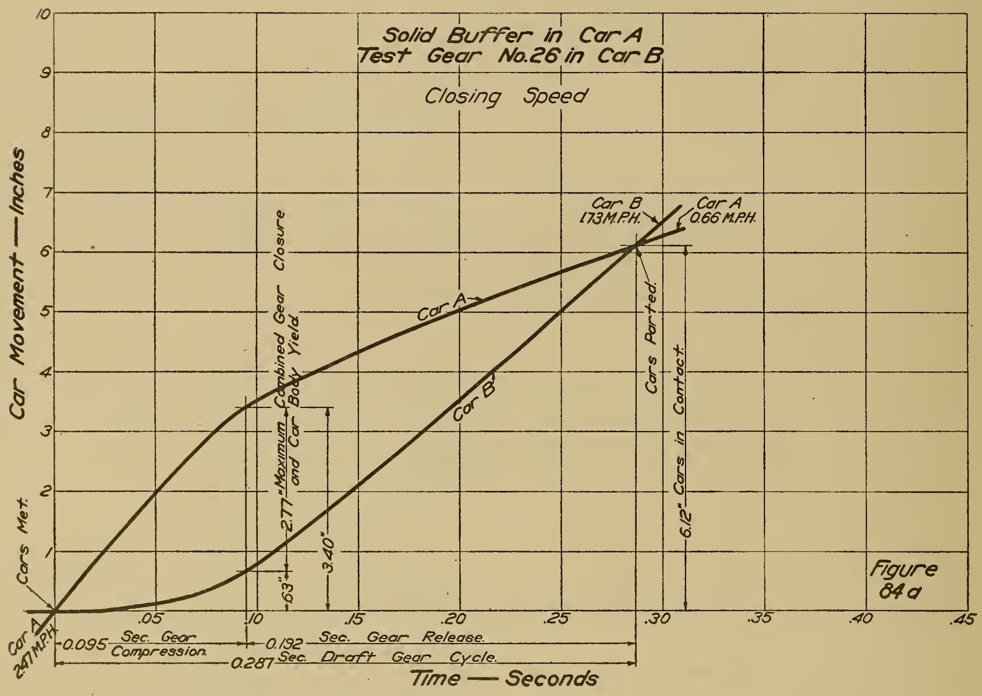
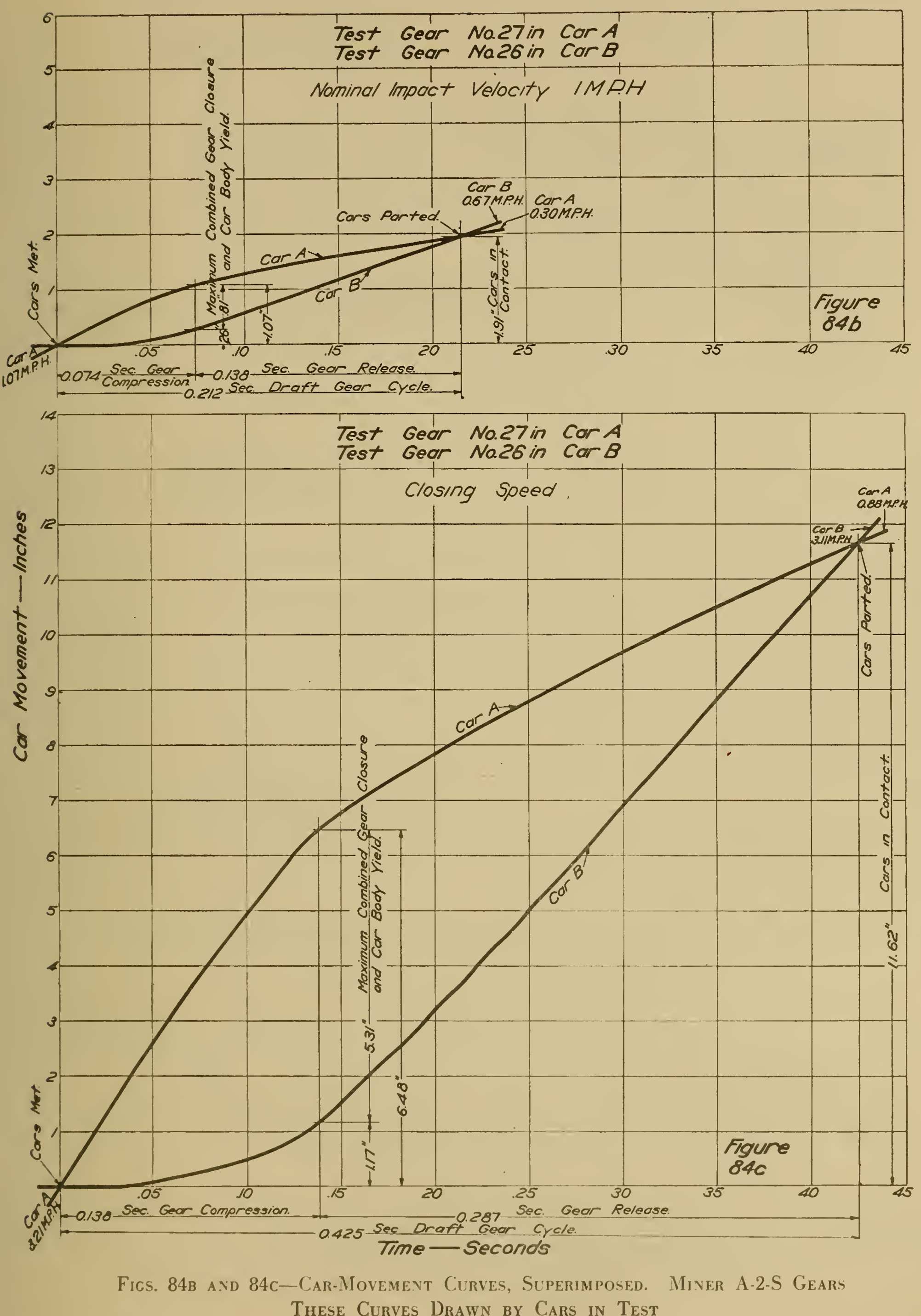
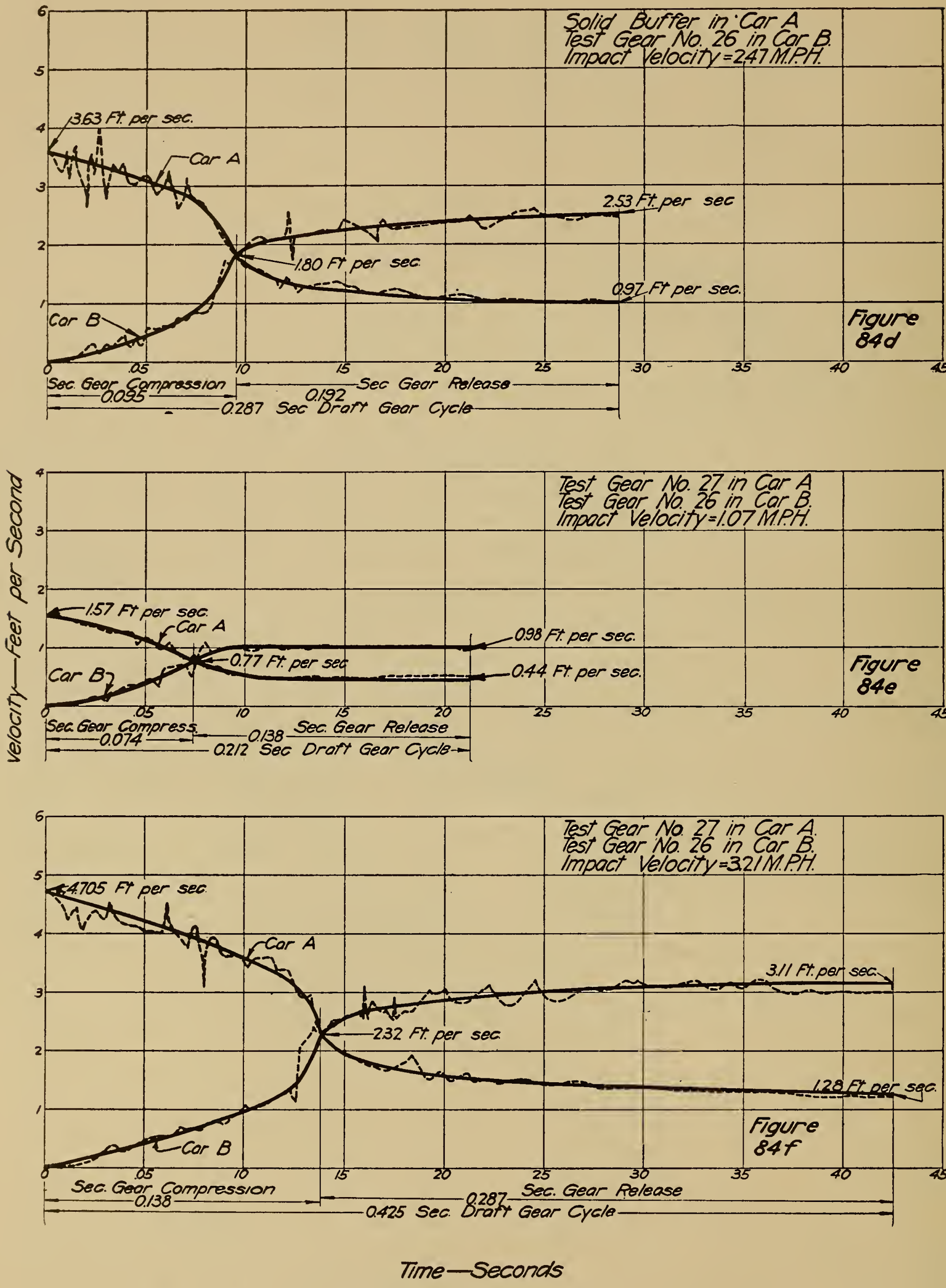


Fig. 84a—Car-Movement Curves, Superimposed. Miner A-2-S Gears
These Curves Drawn by Cars in Test



FIGS. 84B AND 84C—CAR-MOVEMENT CURVES, SUPERIMPOSED. MINER A-2-S GEARS
THESE CURVES DRAWN BY CARS IN TEST

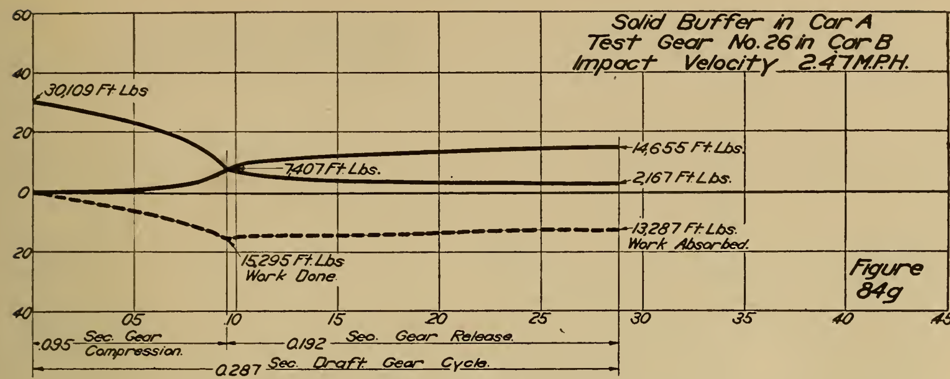


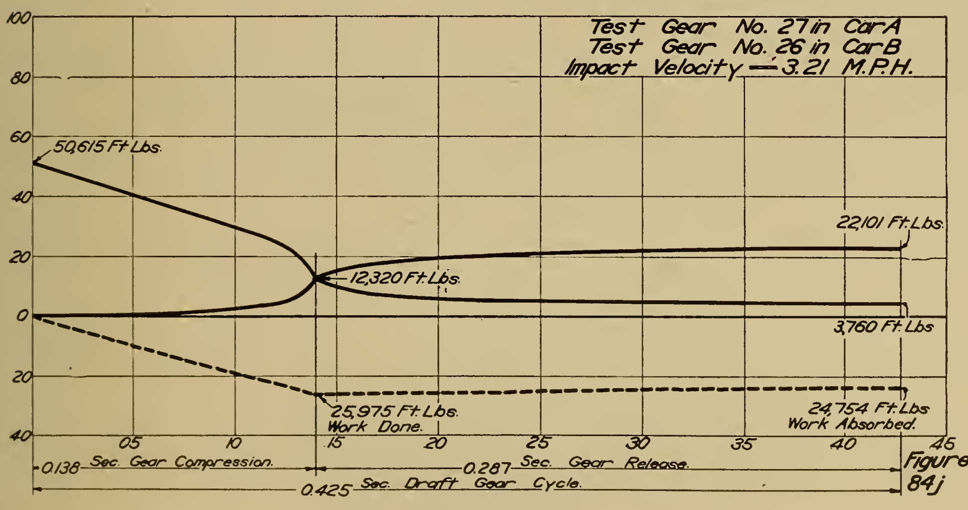
Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

The irregularities are due in general to vibrations of the car structure induced by draft gear action.

Full lines represent the mean velocity curves.

FIGS. 84D, 84E AND 84F—VELOCITY CURVES, MINER A-2-S GEARS

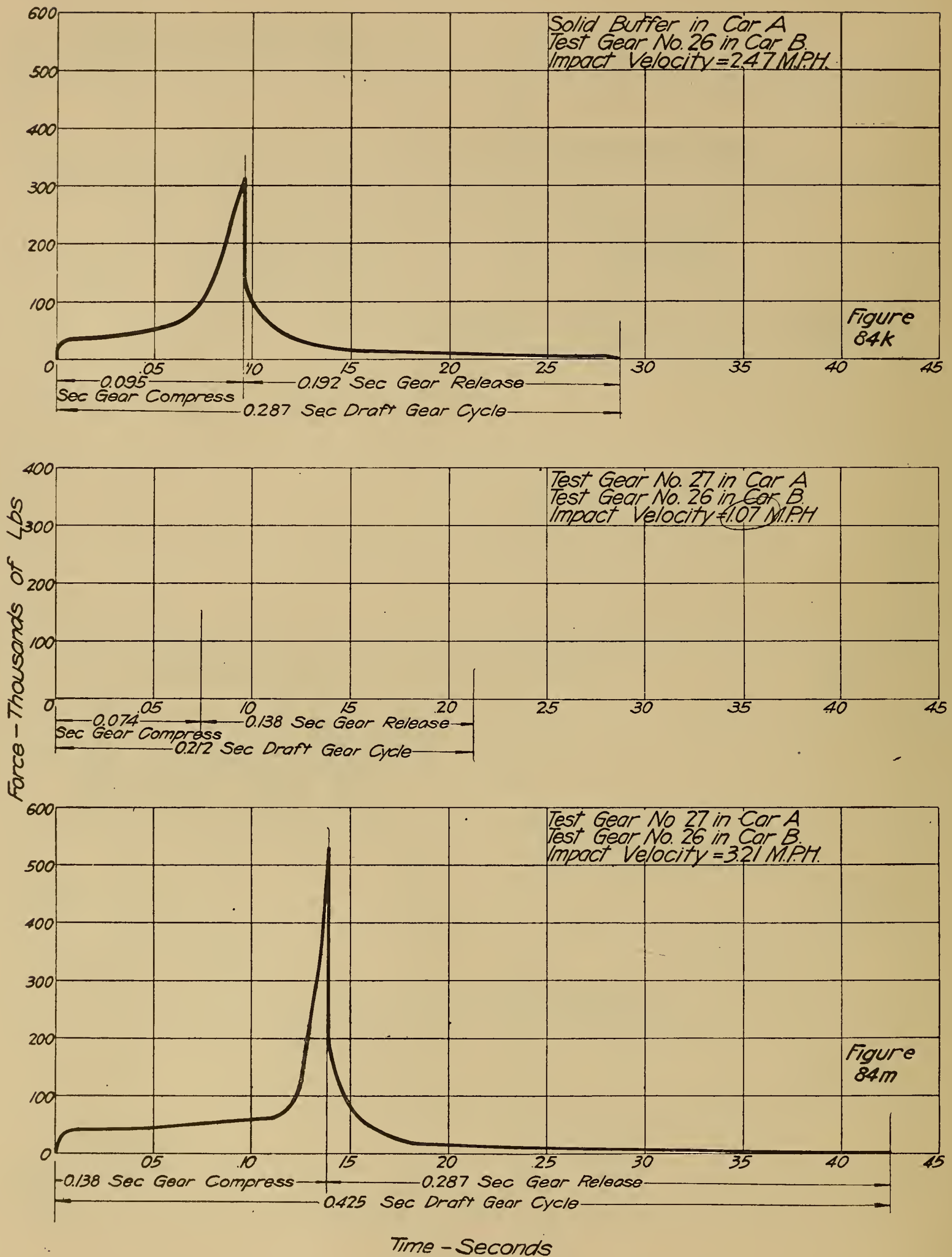




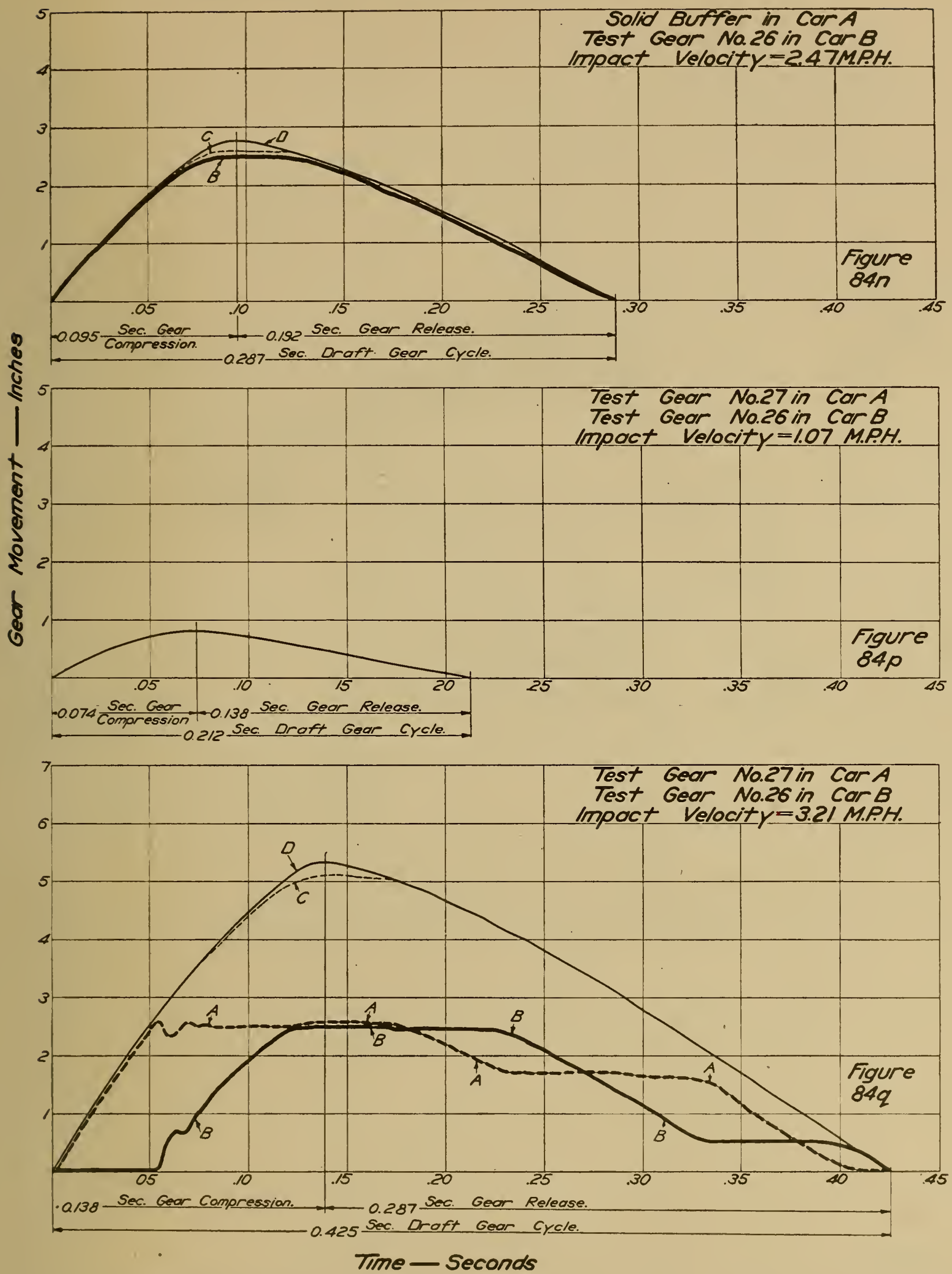
Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft gear cycle.

FIGS. 84G AND 84J—ENERGY CURVES, MINER A-2-S GEARS



FIGS 84K AND 84M—TIME-FORCE CURVES, MINER A-2-S GEARS



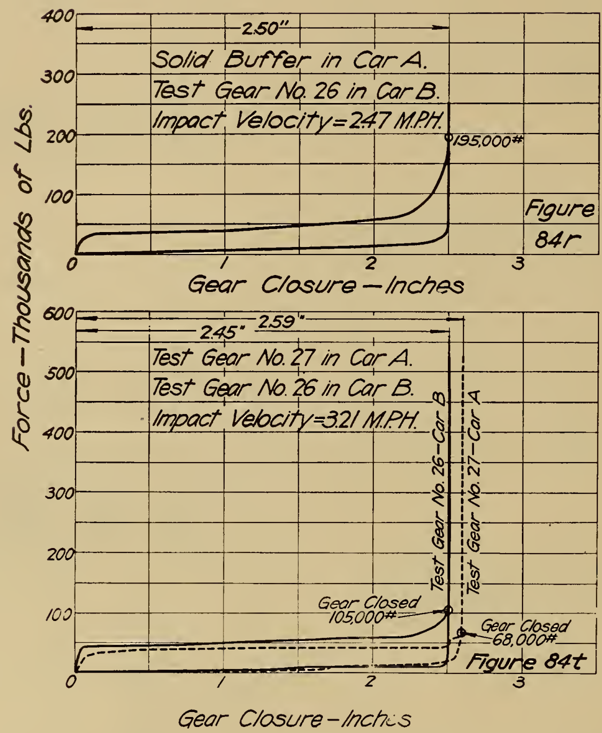
Curve D, determined from superimposed car-movement curves, represents combined draft gear movement and yield of car bodies.

Curve C, obtained by eliminating car body yield from curve D, represents true combined movement of both gears.

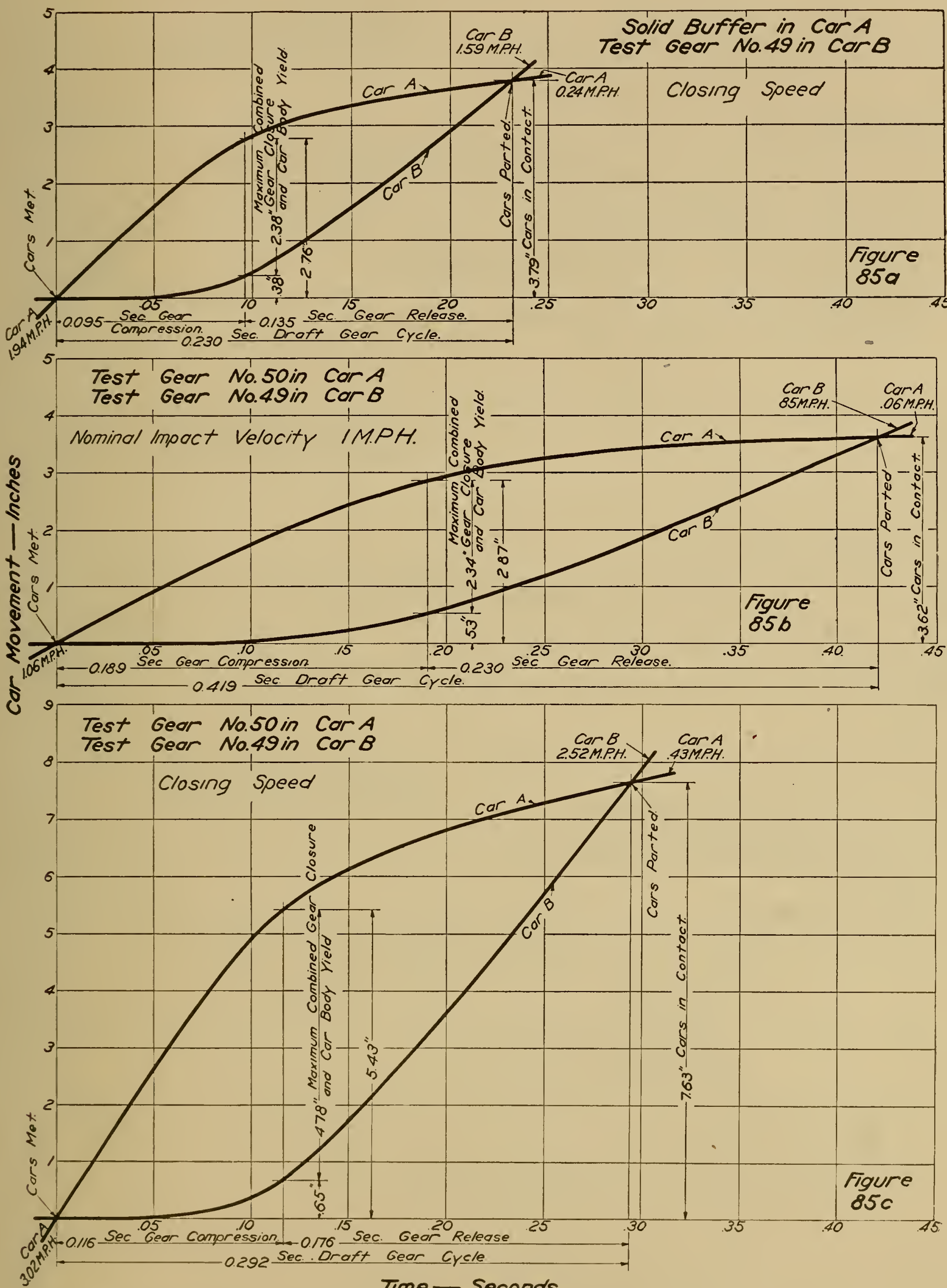
Curve B, traced on small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

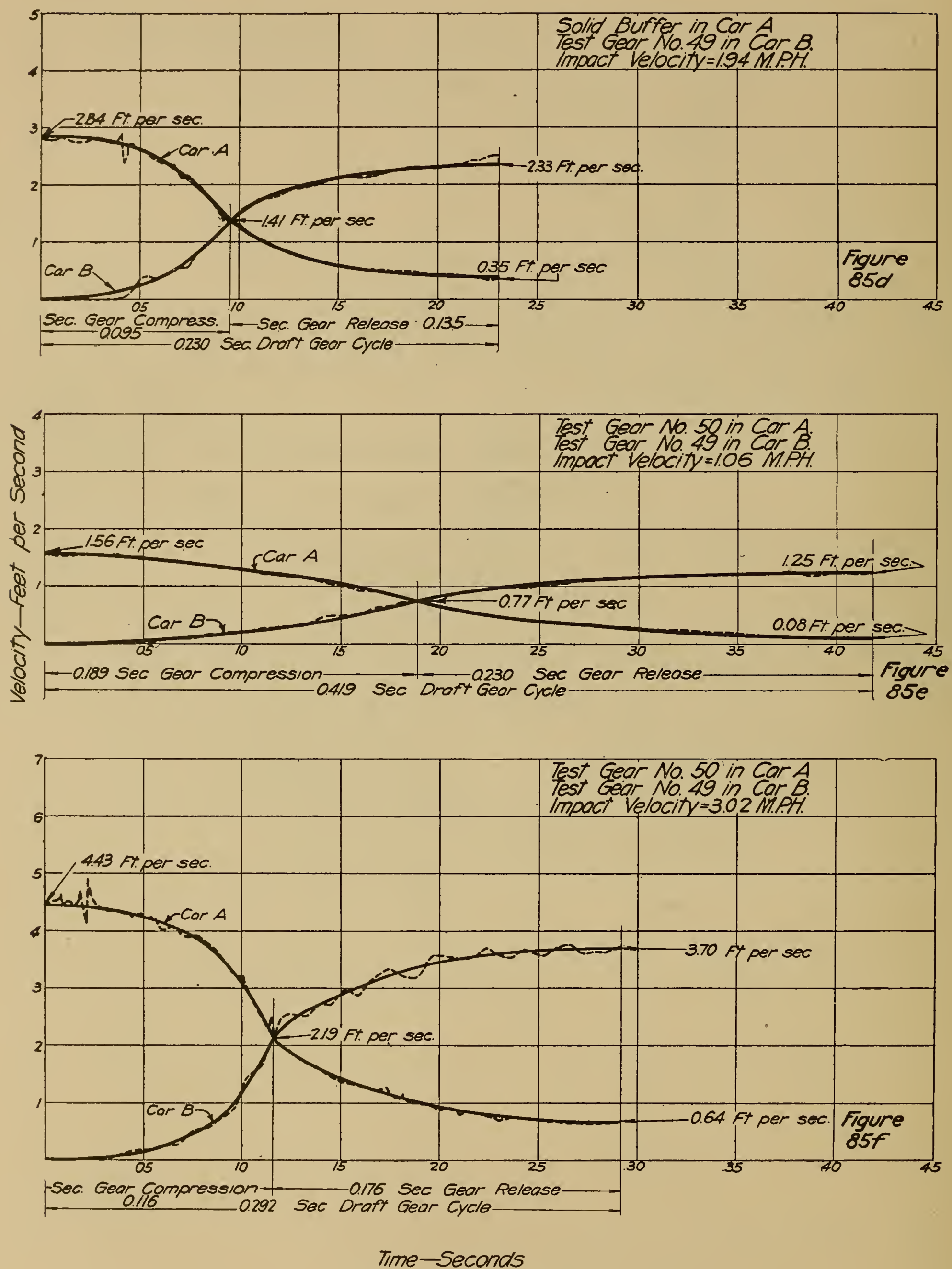
Figs. 84n, 84p and 84q—Time-Closure Curves, Miner A-2-S Gears



FIGS. 84R AND 84T—FORCE-CLOSURE DIAGRAMS, MINER A-2-S GEARS



Figs. 85a, 85b and 85c—Car-Movement Curves, Superimposed. Waugh Plate Gears
These Curves Drawn by Cars in Test

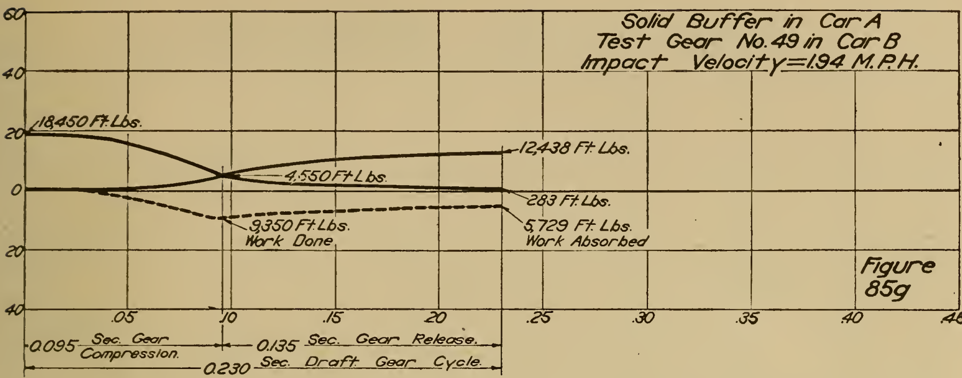


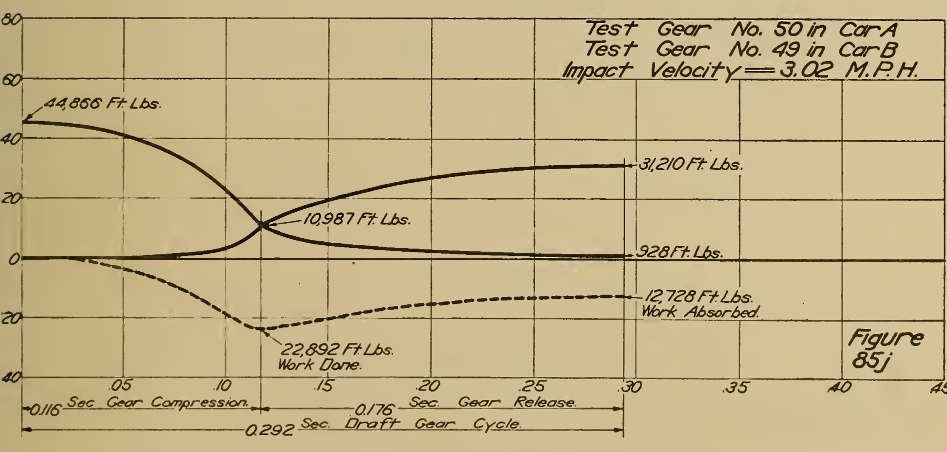
Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

The irregularities are due in general to vibrations of the car structure induced by draft gear action.

Full lines represent the mean velocity curves.

FIGS. 85D, 85E AND 85F—VELOCITY CURVES, WAUGH PLATE GEARS

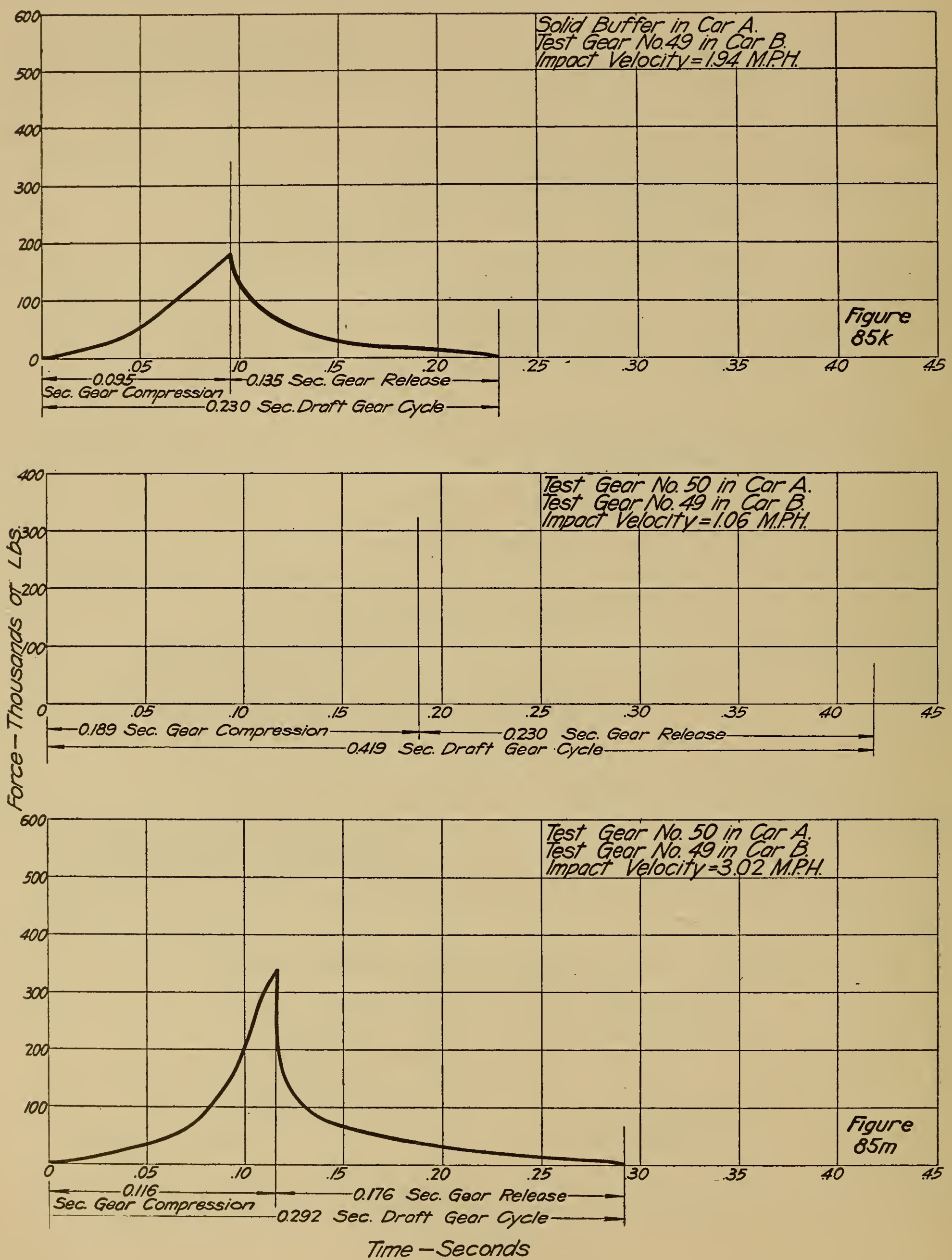




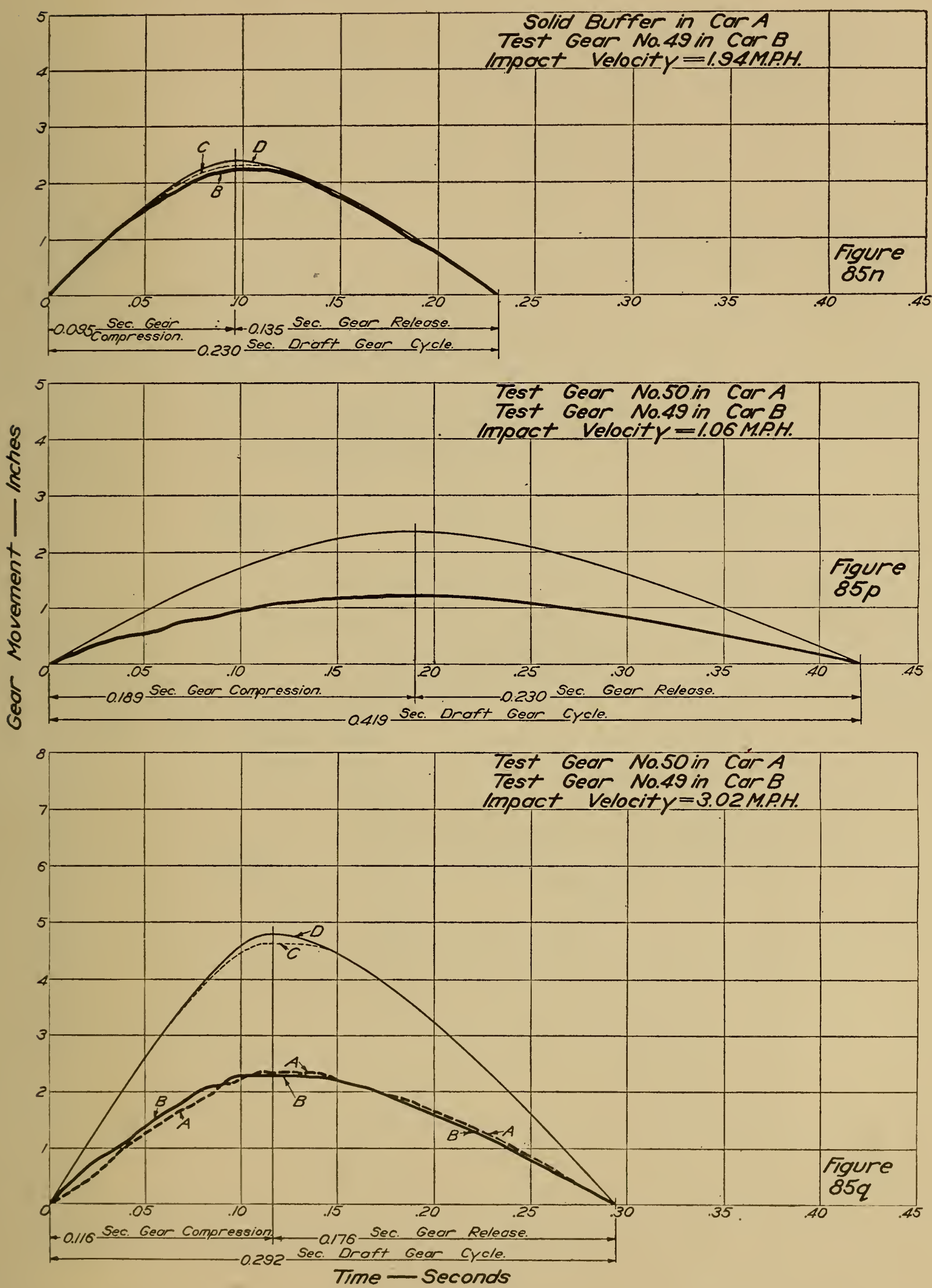
Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft gear cycle.

Figs. 85g and 85j—Energy Curves, Waugh Plate Gears



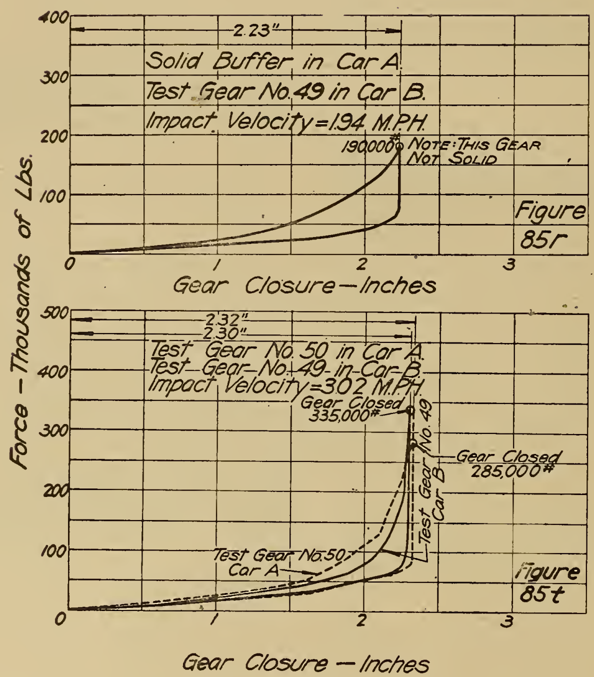
Figs. 85k and 85m—Time-Force Curves, Waugh Plate Gears



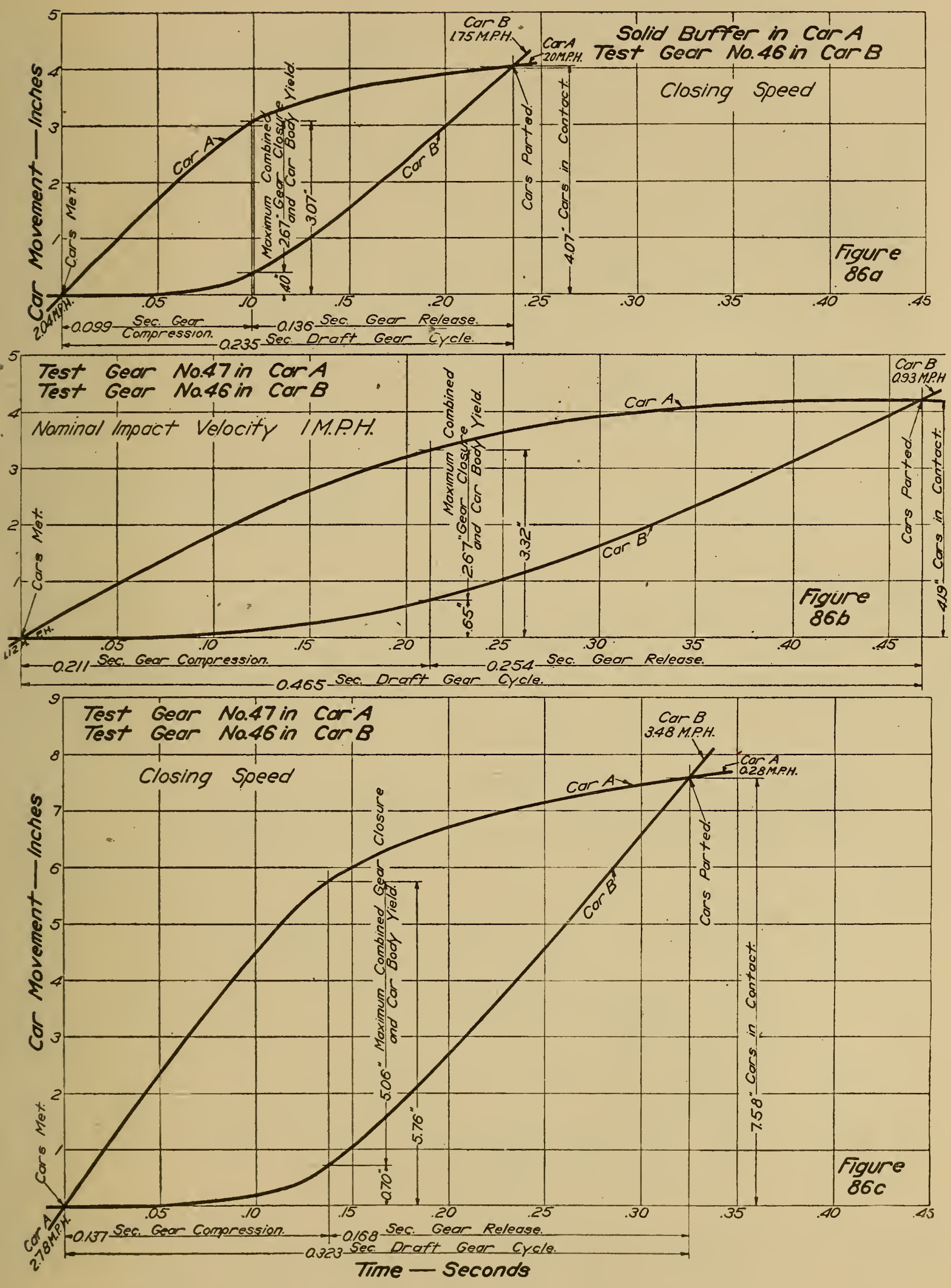
Curve D, determined from superimposed car-movement curves, represents combined draft gear movement and yield of car bodies.
Curve C, obtained by eliminating car body yield from curve D, represents true combined movement of both gears.

Curve B, traced on small drum, represents movement of gear in Car B.
Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

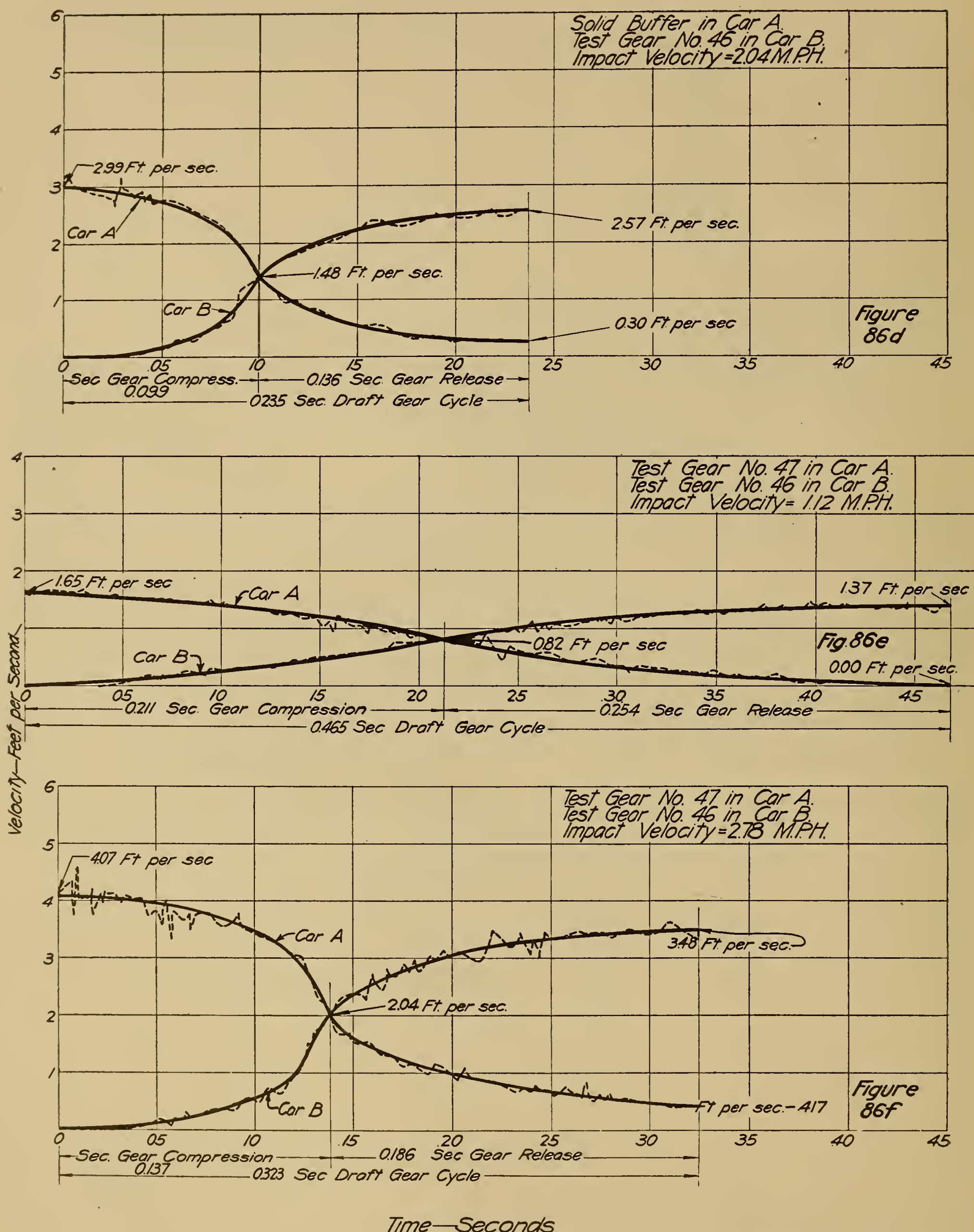
FIGS. 85N, 85P AND 85Q—TIME-CLOSURE CURVES, WAUGH PLATE GEARS



Figs. 85r and 85t—Force-Closure Diagrams, Waugh Plate Gears



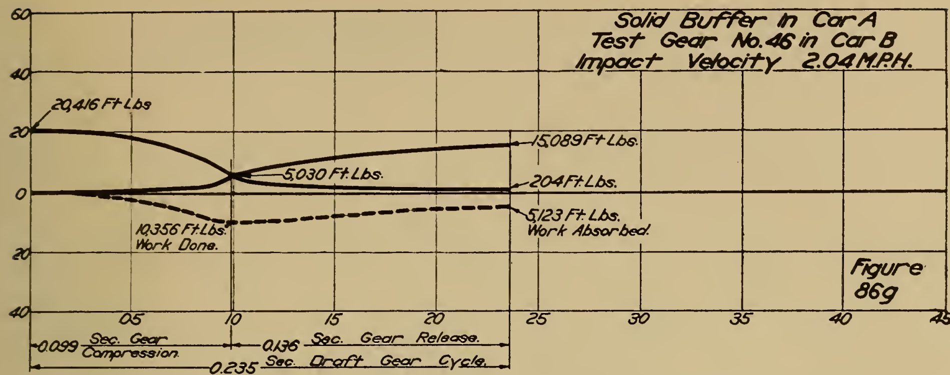
Figs., 86a, 86b and 86c—Car-Movement Curves, Superimposed. Bradford K Gears These Curves Drawn by Cars in Tests

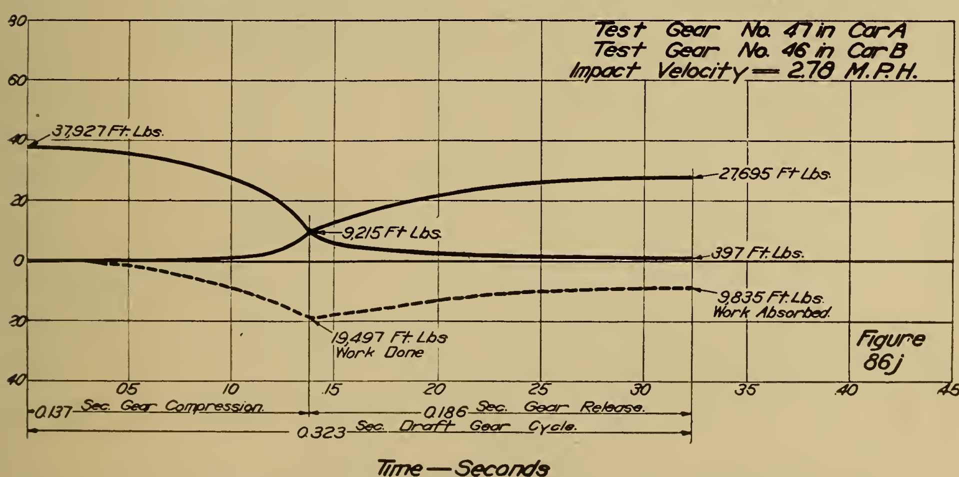


Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

The irregularities are due in general to vibrations of the car structure induced by draft gear action.
Full lines represent the mean velocity curves.

FIGS. 86D, 86E AND 86F—VELOCITY CURVES, BRADFORD K GEARS

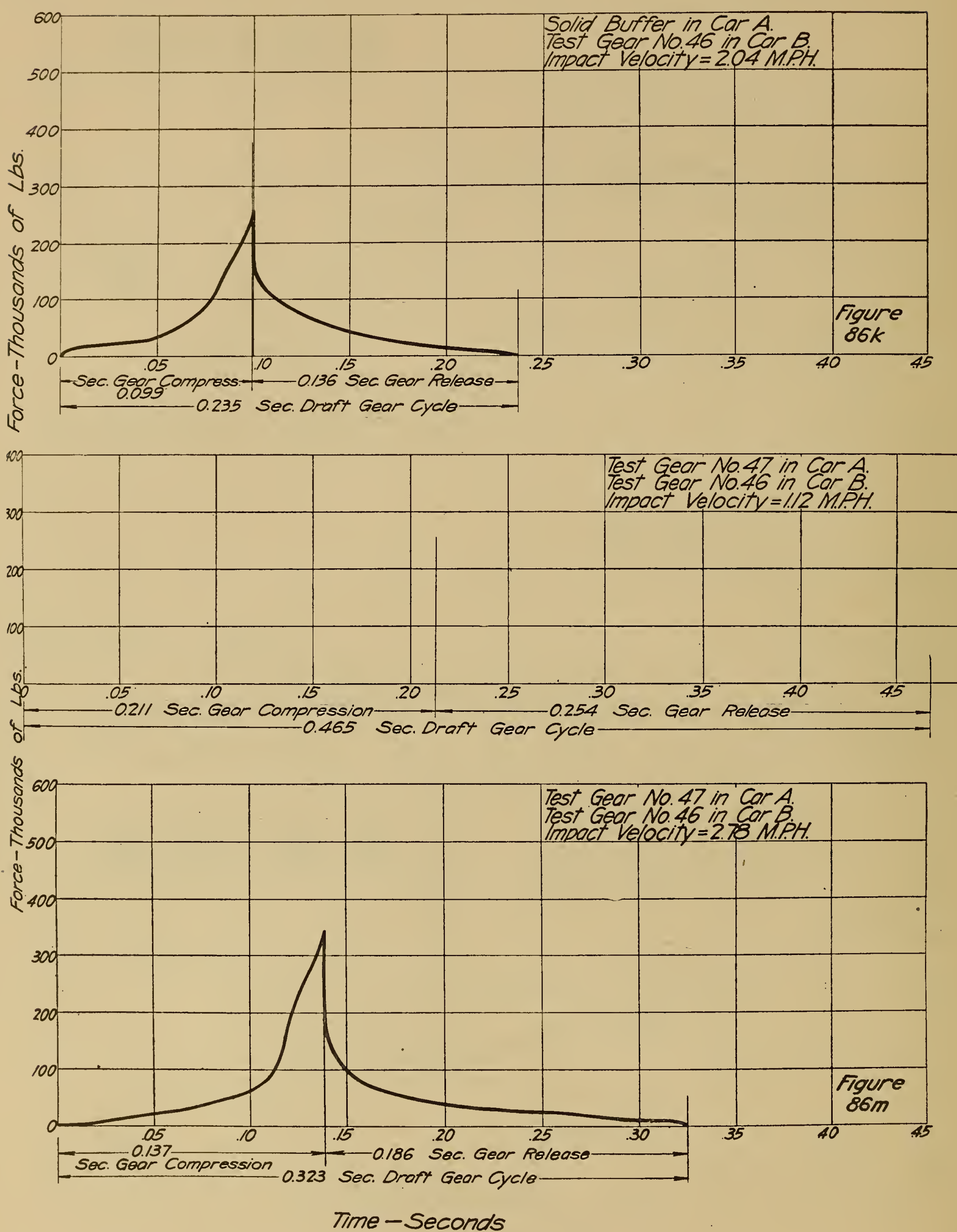




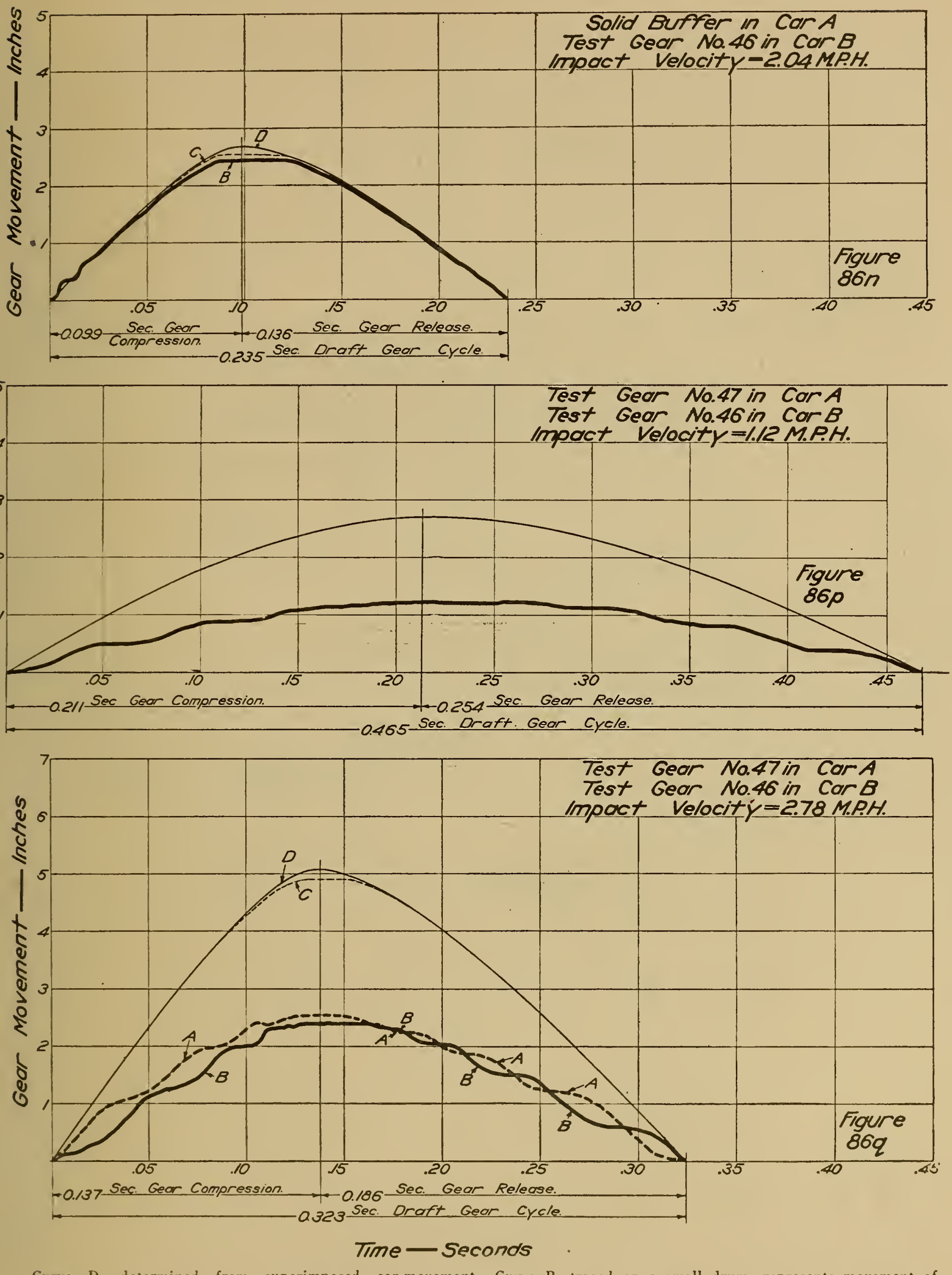
Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft gear cycle.

FIGS. 86G AND 86J—ENERGY CURVES, BRADFORD K GEARS



Figs. 86k and 86m—Time-Force Curves, Bradford K Gears



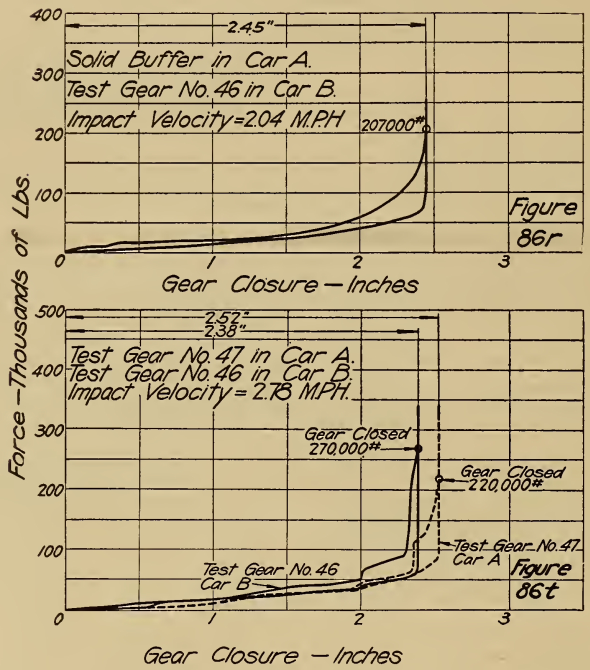
Curve D, determined from superimposed car-movement curves, represents combined draft gear movement and yield of car bodies.

Curve C, obtained by eliminating car body yield from curve D, represents true combined movement of both gears.

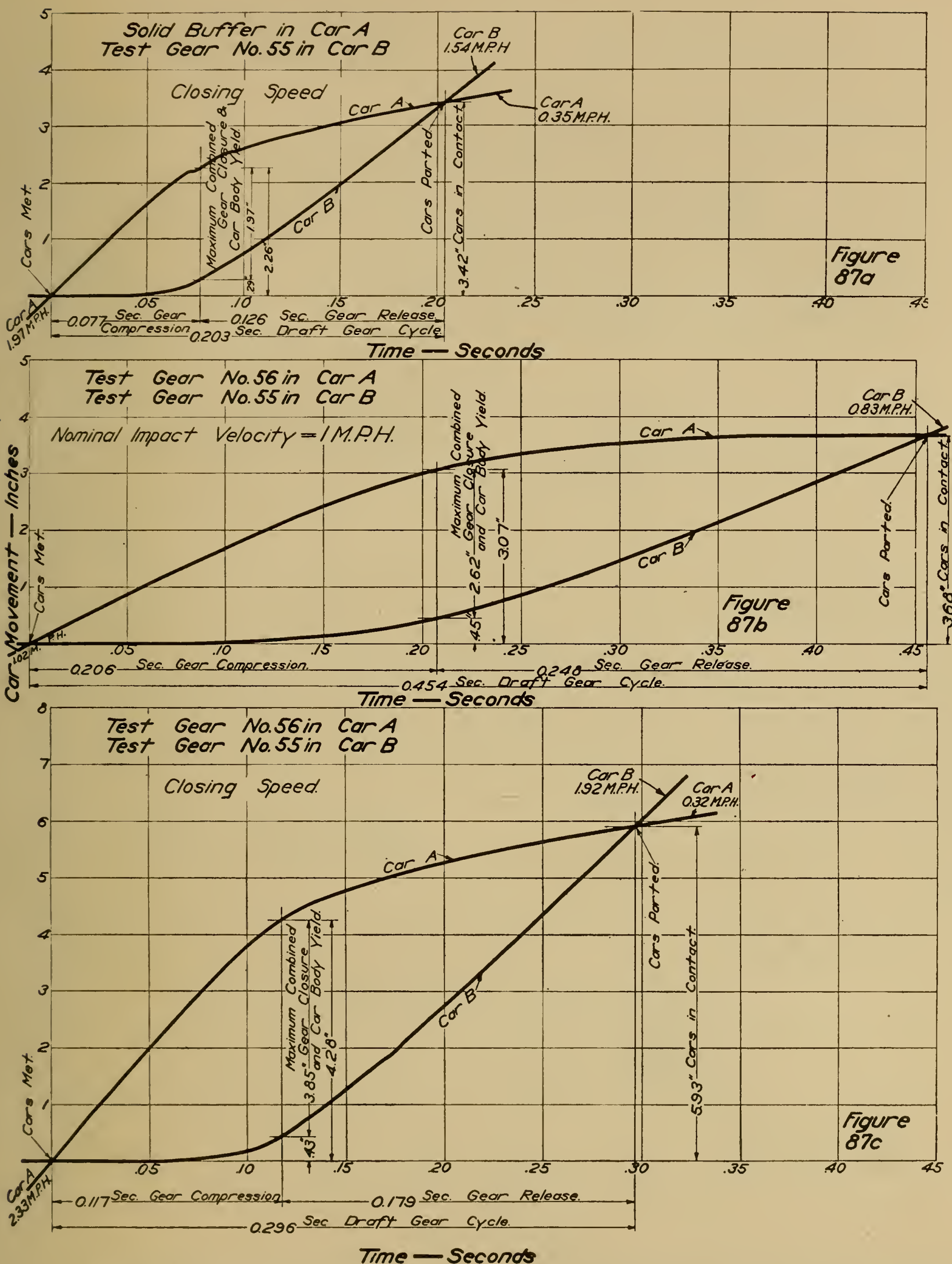
Curve B, traced on a small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

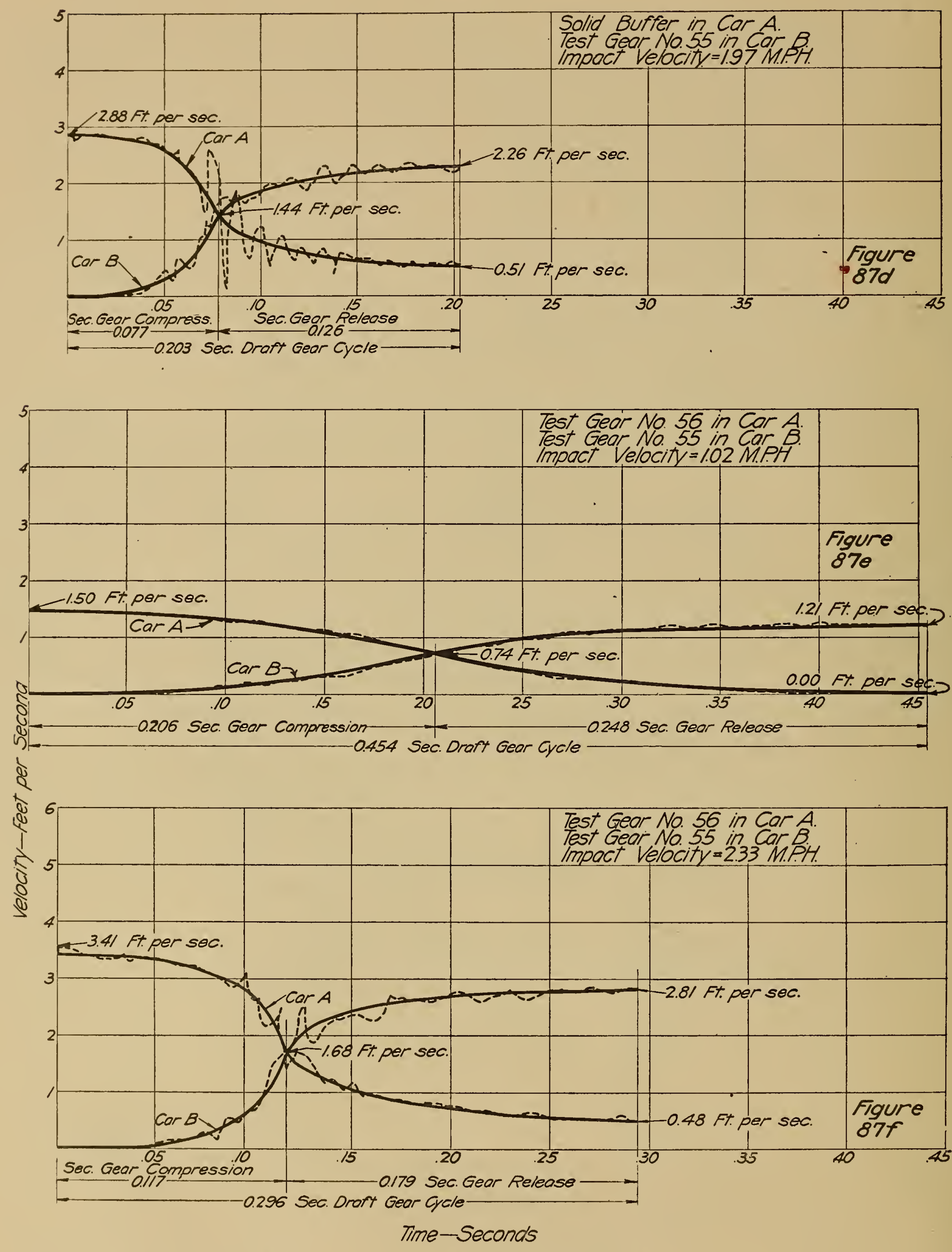
Fig. 86n, 86p and 86q—Time-Closure Curves, Bradford K Gears



Figs. 86r and 86t—Force-Closure Diagrams, Bradford K Gears



FIGS. 87A, 87B AND 87C—CAR-MOVEMENT CURVES, SUPERIMPOSED, HARVEY SPRINGS
THESE CURVES DRAWN BY CARS IN TESTS

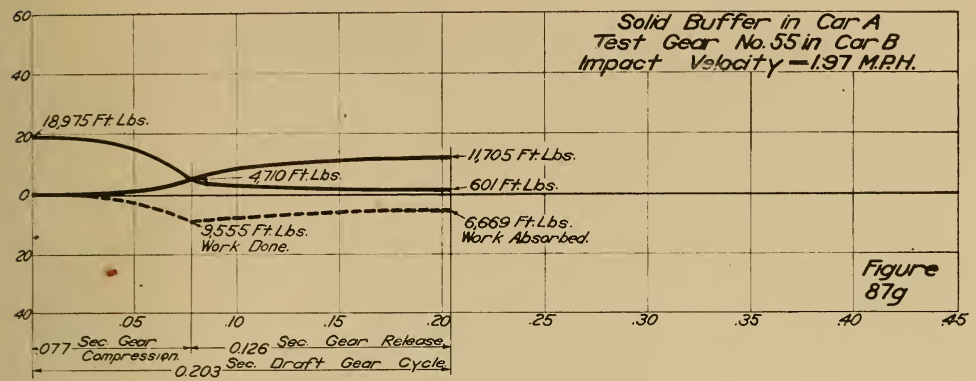


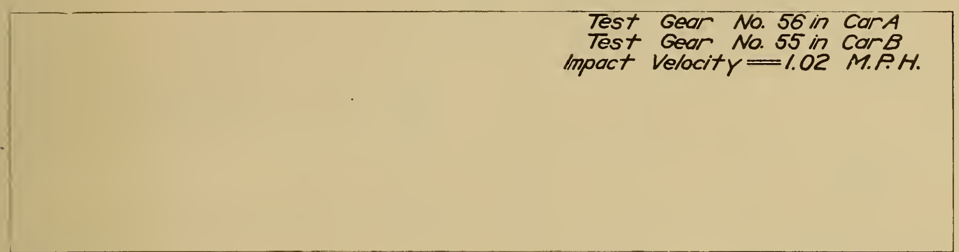
Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

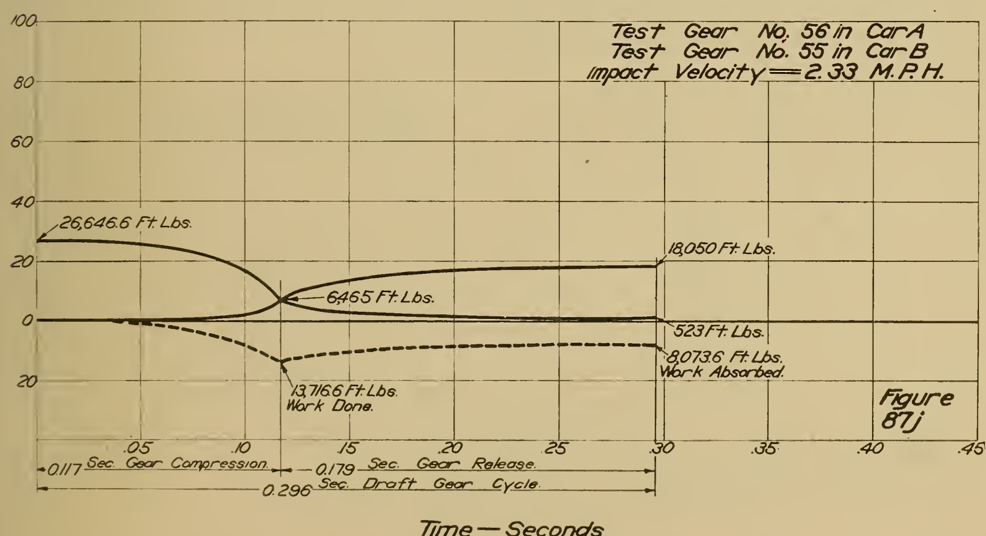
The irregularities are due in general to vibrations of the car structure induced by draft gear action.

Full lines represent the mean velocity curves.

FIGS. 87D, 87E AND 87F—VELOCITY CURVES, HARVEY SPRINGS



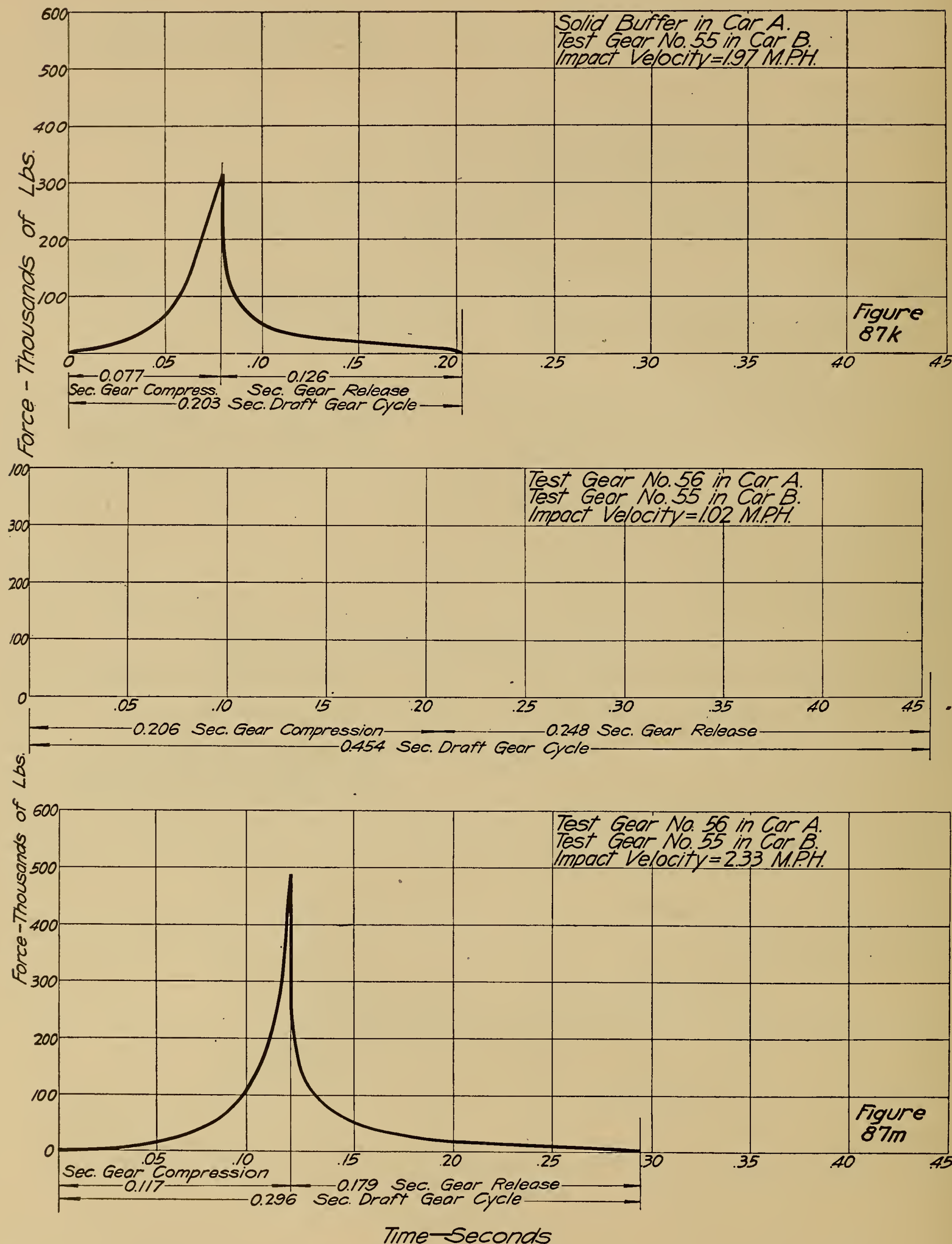




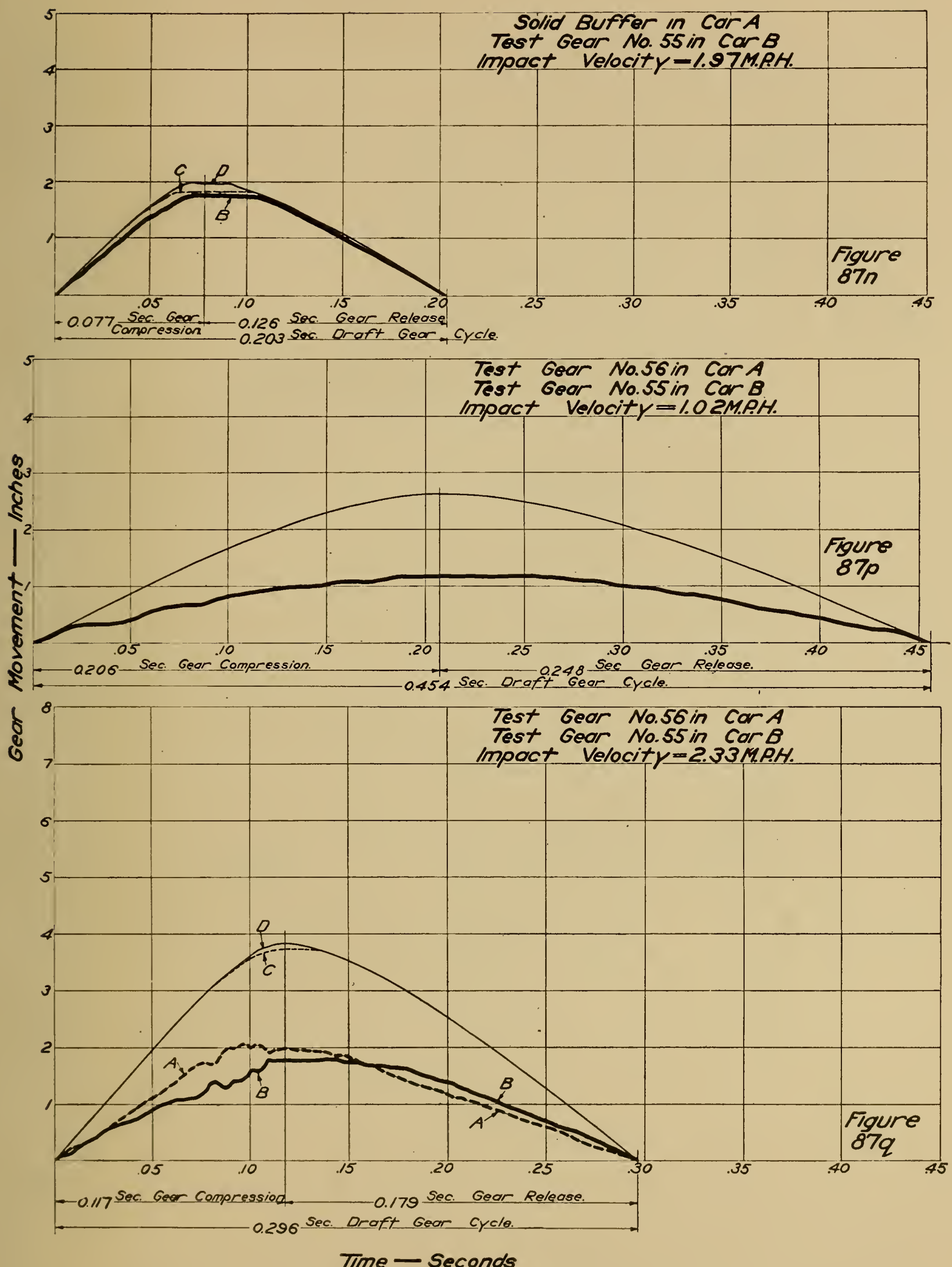
Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft gear cycle.

FIGS. 87G AND 87J—ENERGY CURVES, HARVEY SPRINGS



FIGS. 87K AND 87M—TIME-FORCE CURVES, HARVEY SPRINGS



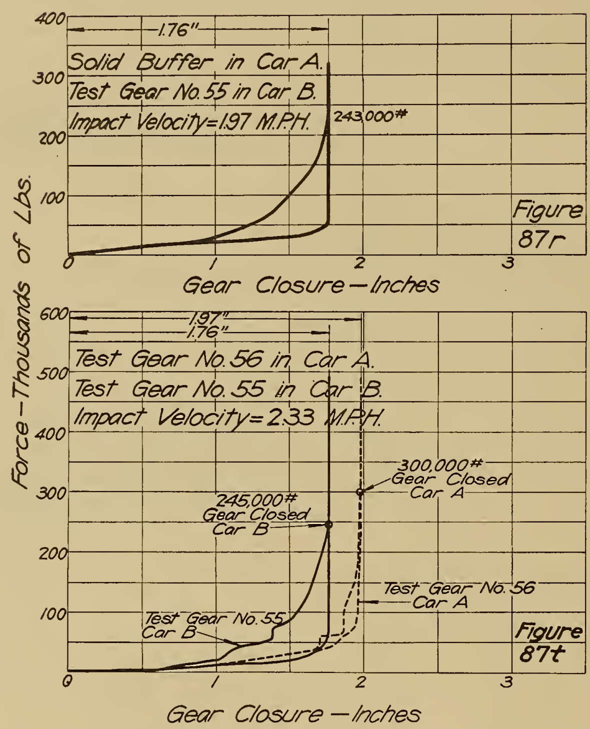
Curve D, determined from superimposed car-movement curves, represents combined draft gear movement and yield of car bodies.

Curve C, obtained by eliminating car body yield from curve D, represents true combined movement of both gears.

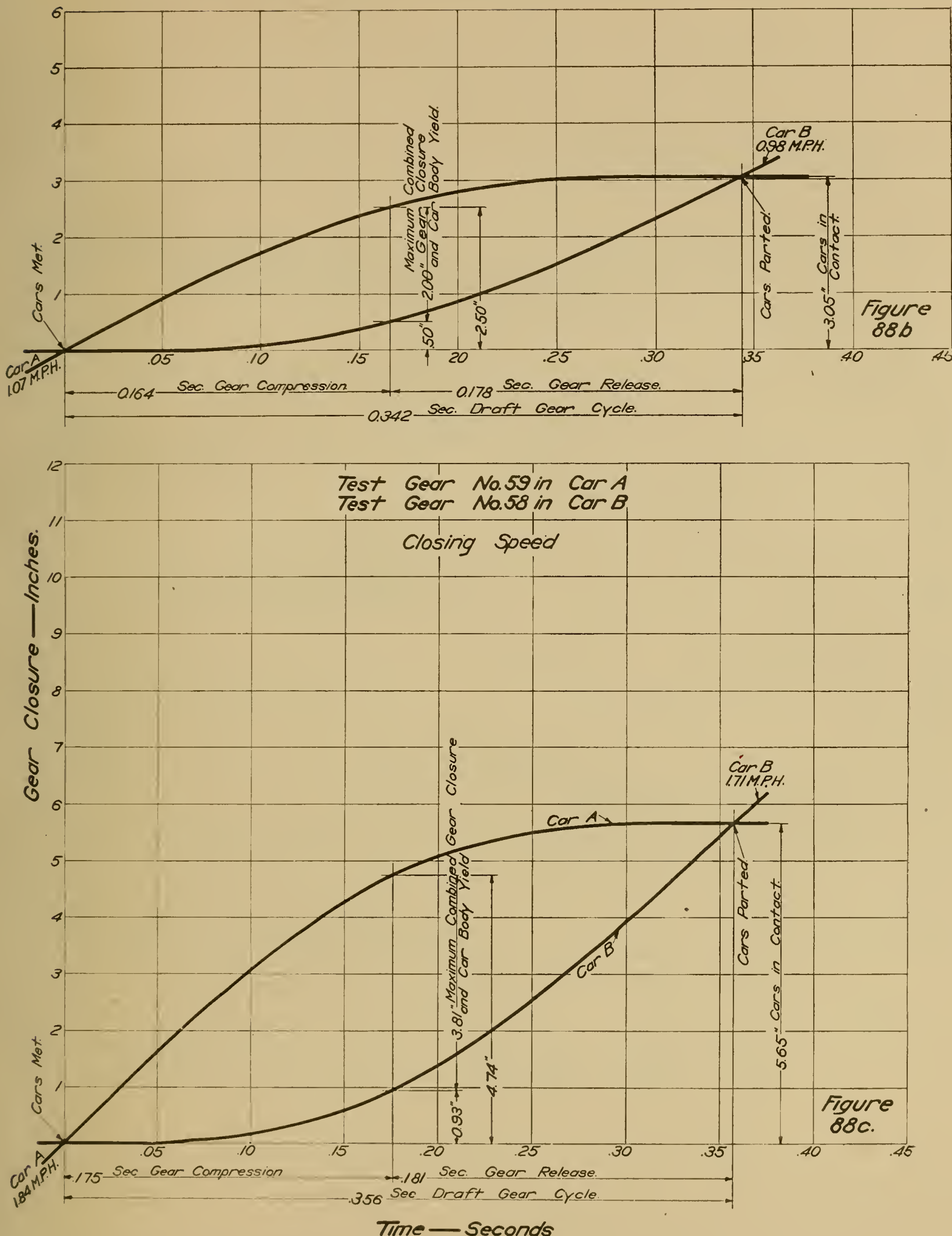
Curve B, traced on small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

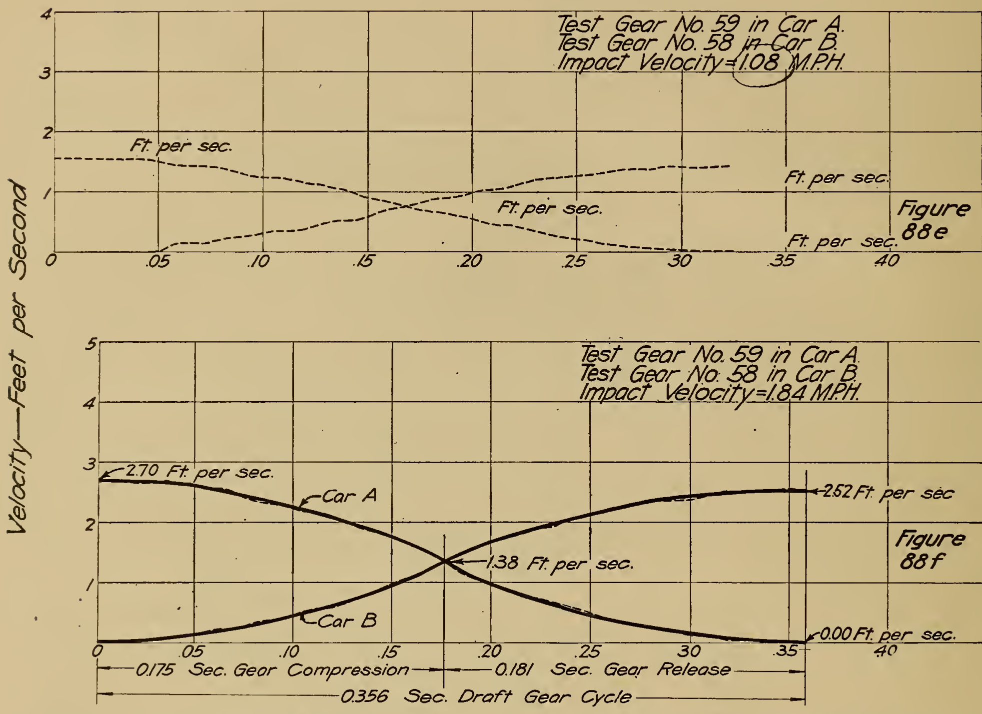
Figs. 87n, 87p and 87q—Time-Closure Curves, Harvey Springs



Figs. 87r and 87t—Force-Closure Diagrams, Harvey Springs



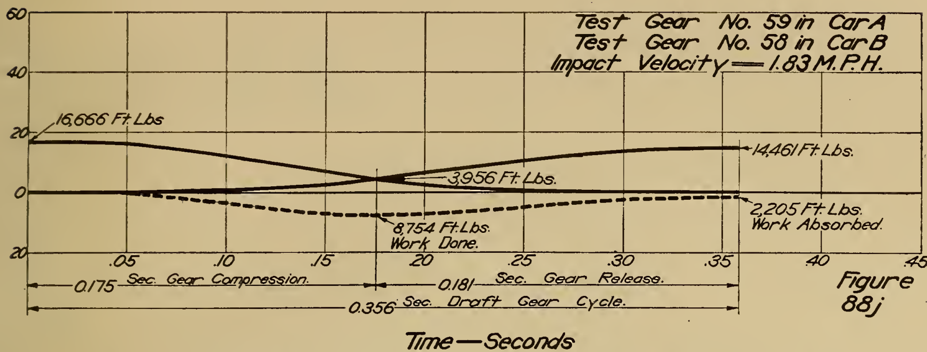
FIGS. 88B AND 88C—CAR-MOVEMENT CURVES, SUPERIMPOSED. A. R. A. CLASS G SPRINGS
THESE CURVES DRAWN BY CARS IN TESTS



Dotted lines represent instantaneous car velocities as determined from the original car-movement curves.

The irregularities are due in general to vibrations of the car structure induced by draft gear action.
Full lines represent the mean velocity curves.

Figs. 88e and 88f—Velocity Curves, A. R. A. Class G Springs



Full lines represent the instantaneous kinetic energy of the moving cars.

Dotted lines represent the energy stored and absorbed during the draft gear cycle.

FIG. 88J—ENERGY CURVE, A. R. A. CLASS G SPRINGS

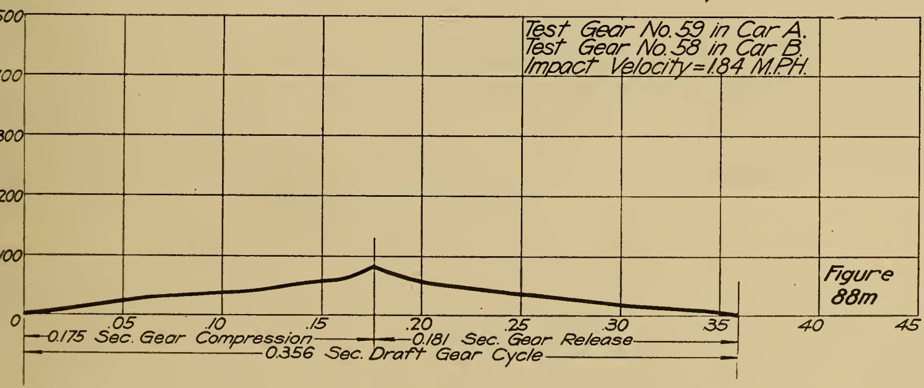
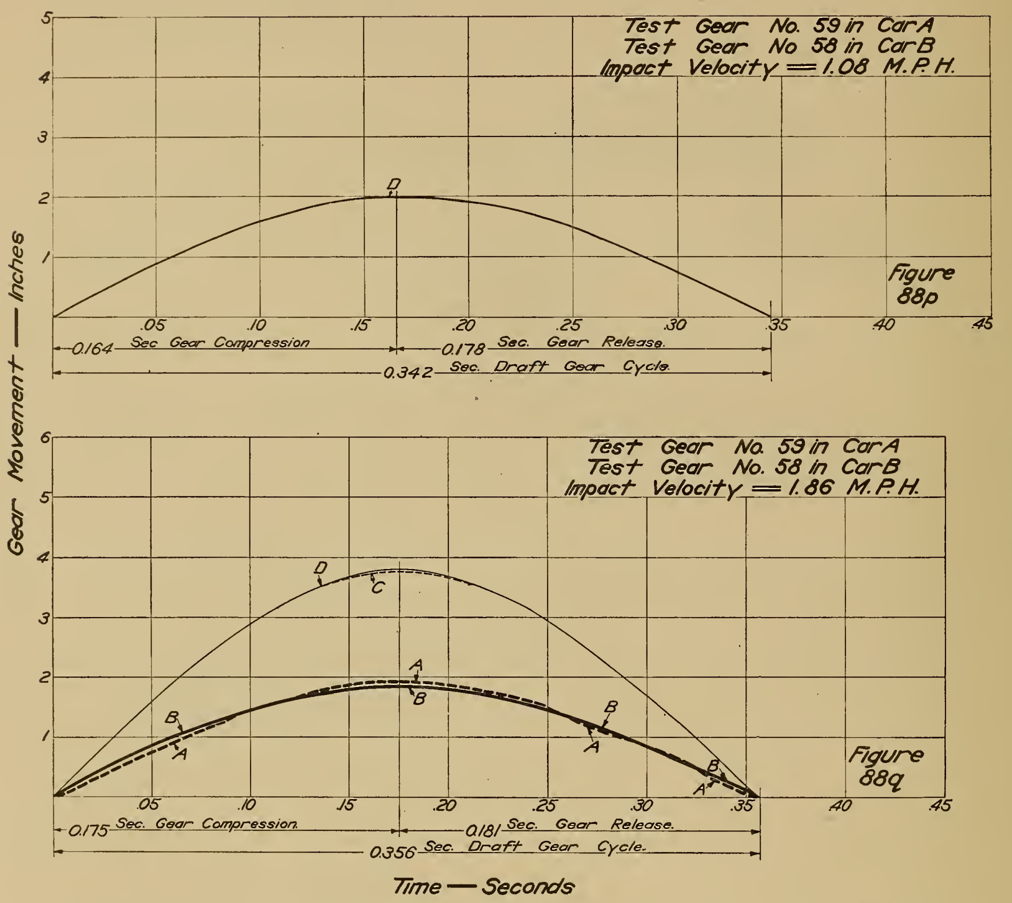


FIG. 88M—TIME-FORCE CURVE, A. R. A. CLASS G SPRINGS



Curve D, determined from superimposed car-movement curves, represents combined draft gear movement and yield of car bodies.

Curve C, obtained by eliminating car body yield from curve D, represents true combined movement of both gears.

Curve B, traced on small drum, represents movement of gear in Car B.

Curve A, derived from curves C and B, represents simultaneous movement of gear in Car A.

FIGS 88P AND 88Q—TIME-CLOSURE CURVES, A. R. A. CLASS G SPRINGS

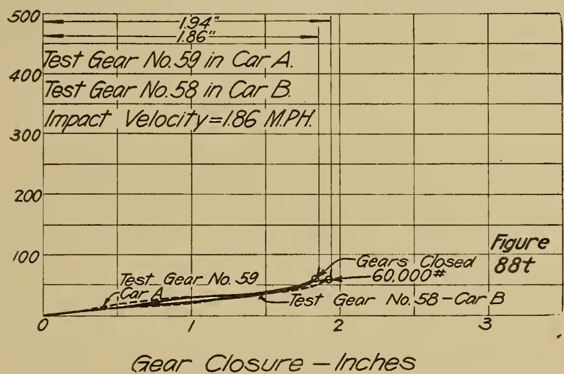


FIG. 88T—FORCE-CLOSURE DIAGRAM, A. R. A. CLASS G SPRINGS

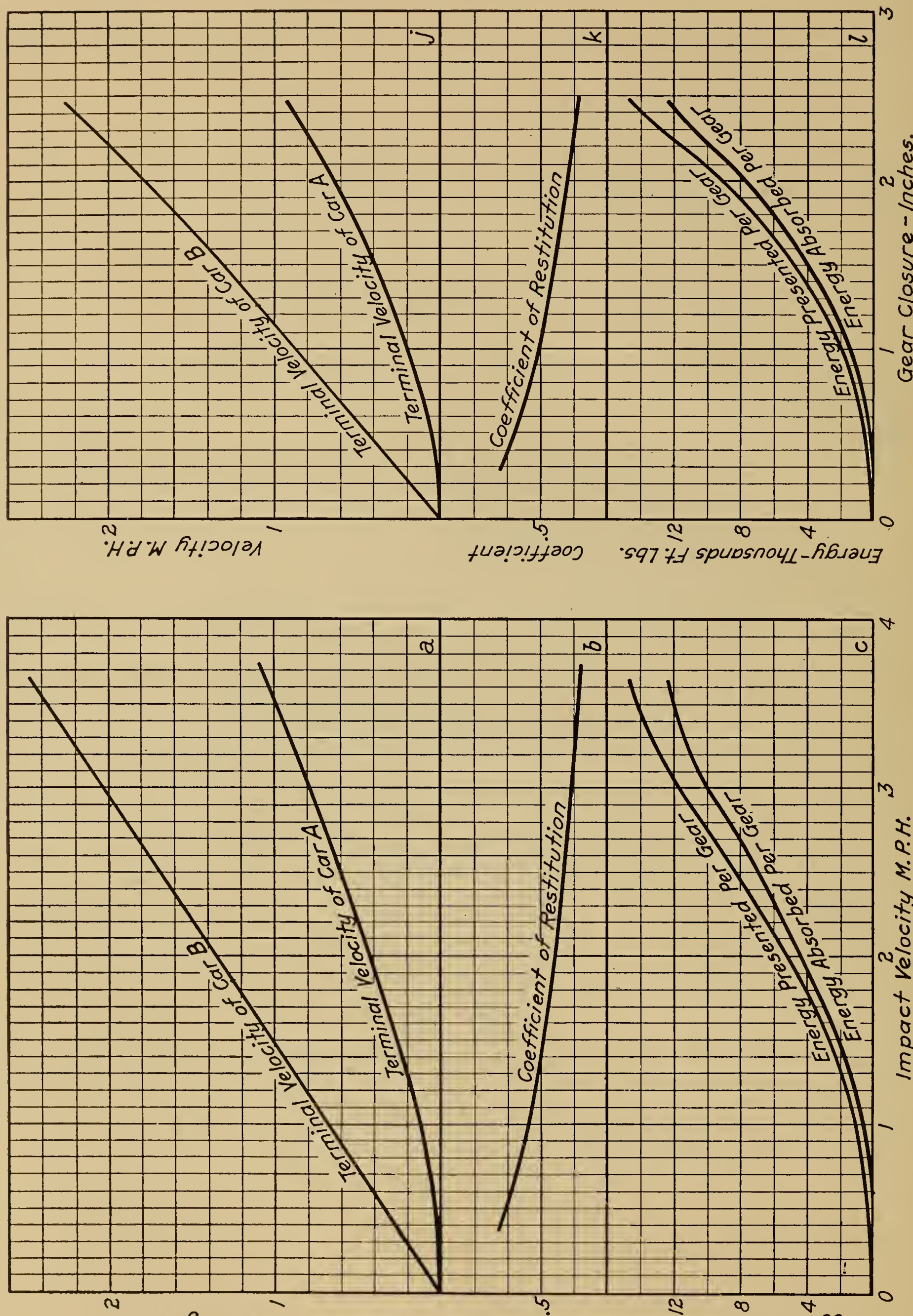


Fig. 89-1—Summary Curves, Westinghouse D-3 Gears in Both Cars

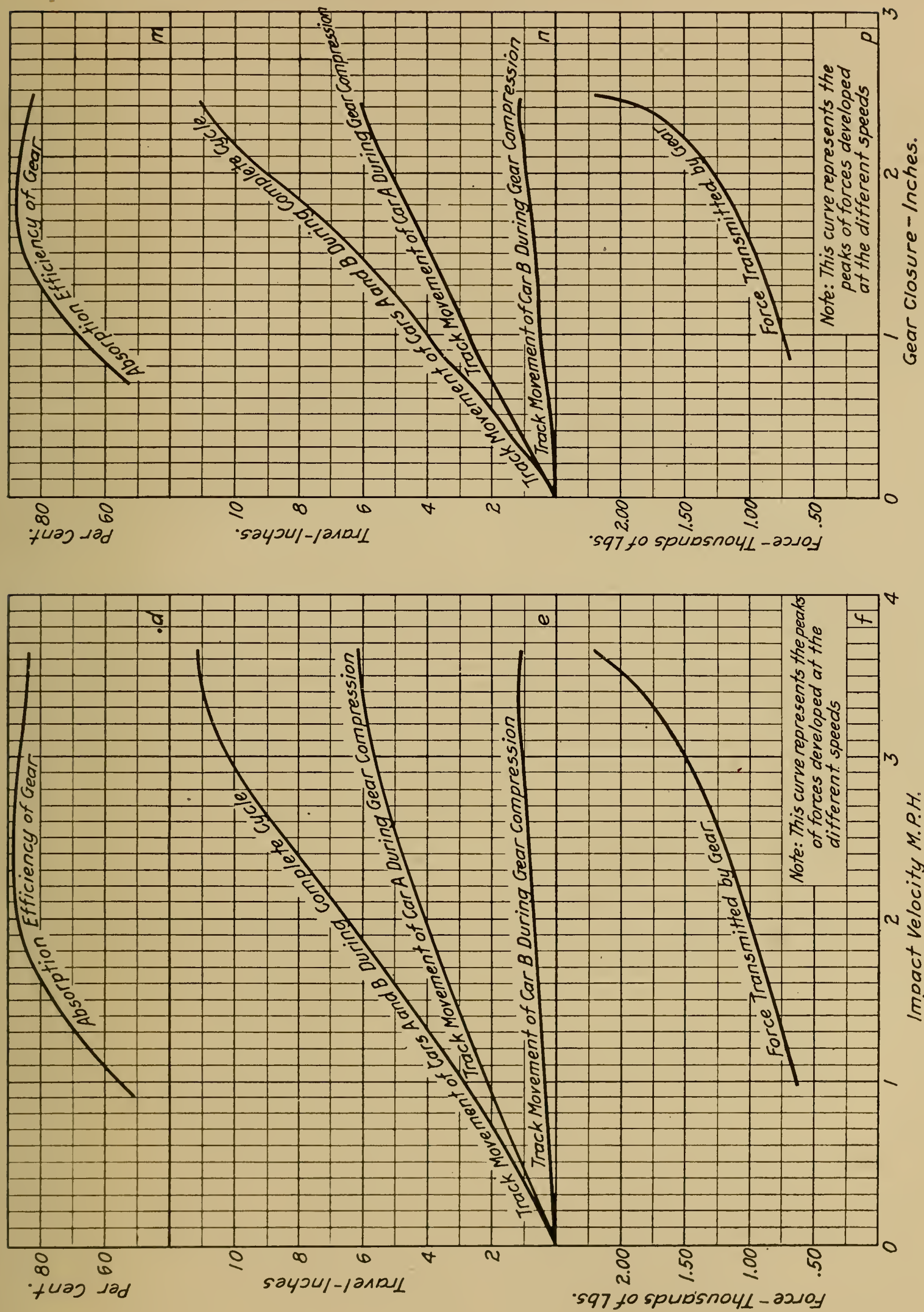


FIG. 89-2—SUMMARY CURVES, WESTINGHOUSE D-3 GEARS IN BOTH CARS

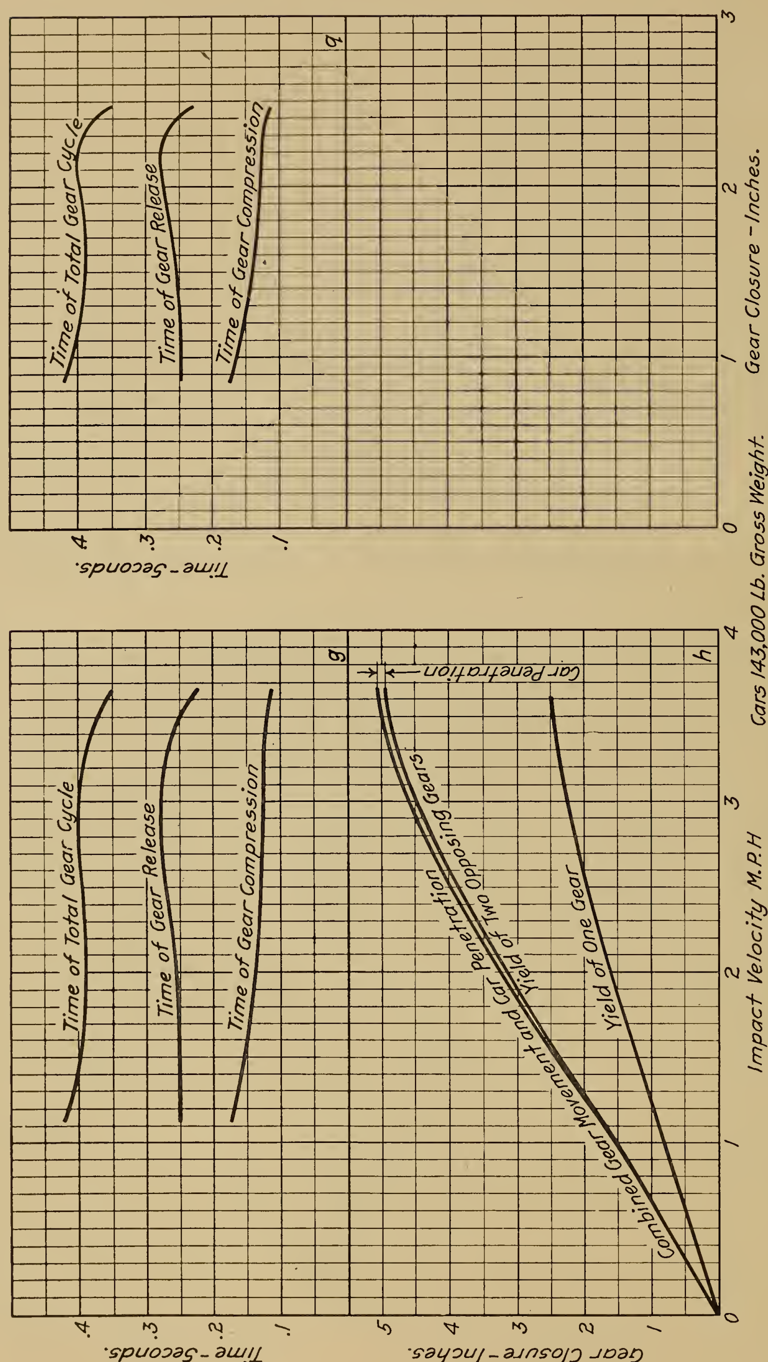


Fig. 89-3.—Summary Curves, Westinghouse D-3 Gears in Both Cars

APPENDIX A

REPORT OF DRAFT GEAR TEST MADE ON NORFOLK & WESTERN RAILROAD, NOVEMBER 4, 1918

Object of Test

This test was conducted for the purpose of determining the relative amount of draft gear movement when a car having a draft gear with an easy compression curve is coupled to a car having a draft gear with a relatively stiff compression curve, and to determine the probable number of foot pounds of work done by each when shocks occur. The test was made on the Norfolk & Western Railroad in a freight run from West Roanoke, Va., to Vicker, Va., a distance of 38 miles. The test was conducted jointly by the Engineer of Tests of the Norfolk & Western Railroad, and the Section of Inspection and Tests of the United States Railroad Administration, a representative of the National Malleable Castings Company being present to assist in the handling of the recording instruments, which were loaned by that company for the purpose of making the test.

Equipment Used

The train consisted of 44 miscellaneous cars, the tonnage being 1,748 tons. From West Roanoke to Elliston, over an undulating grade, it was handled by Norfolk & Western locomotives 422 and 1477 on the head end, the first of these being a Class M of 40,000 lb. tractive effort, and the second being a Mallet, Class Z1a, of 73,000 lb. tractive effort. At Elliston, the foot of an ascending grade of 1.32 per cent, the Class M locomotive was put on the rear end to act as a pusher, and at Christiansburg, the top of the grade, the Class M was cut off entirely. From Christiansburg to Vicker is a descending grade.

The cars from which the records were made were Norfolk & Western 100-ton gondolas, empty, being the first and second cars in the train. The rear end of the first car, Norfolk & Western 100,147, was equipped with a Sessions type K draft gear, and the coupled head end of the second car, Norfolk & Western 101,534, was equipped with a National type H-1 draft gear. Both cars had experimental M.C.B. type C couplers, No. 5 contour, and the draft gears were especially prepared for the test. All slack was eliminated from the draft gear attachments. The Sessions K gear as it was applied to the car had $\frac{5}{16}$ in. initial compression, the National type H-1 gear being applied with but enough initial compression to take out all slack. Holes were cut in the floors of the cars for the application of the recording devices and for observing the action of the draft gears.

Preparation of Draft Gears

The Sessions type K gear used was removed from a Norfolk & Western cabin car and was in practically new condition, the friction surfaces being worn to a smooth bearing, but not enough to remove all of the irregularities of manufacture. The friction faces were wiped off and the gear set up in the 200,000 lb. testing machine of the Norfolk & Western Railroad, and an attempt made to close it. Repeated sticking and bombardment of the gear led to the application of a thin coat of tallow on the center friction block to enable the closing of it in this machine without the necessity of sledging the gear. The closing speed was $\frac{3}{16}$ in. per minute. This treatment of the gear was necessary also to give the easier compression line desired for the purpose of the test, since as previously stated the primary purpose of the test was to observe the action of a stiff gear when coupled with an easy gear. The greased center friction block did not entirely eliminate sticking of the gear, the compression curve shown in Fig. A-2 being plotted directly from the readings taken from the beam of the static machine after a number of preliminary compressions to insure uniform action. The dotted compression line indicated on this curve is worked to in this report as the probable compression line in a quick closing of the gear.

The National type H-1 gear was removed from the same car, No. 101,534, to which it was reapplied for the test, and after wiping off the friction wedges to remove coal dust which had fallen over the gear while cutting the hole in the car floor, the gear was run down as far as the 200,000 lb. machine would compress it for a number of times. The compression curve shown in Fig. A-1 for this gear was plotted directly from the readings taken from the beam of the testing machine. It should be noted that whereas this gear is designed for a total movement of 2½ in., it could only be compressed to .93 in. in this 200,000 lb. machine. The action of this gear in the static machine was smooth and regular.

Recording Apparatus

The records of coupler or draft gear movement were made upon a moving ribbon of paper, one pencil being arranged to draw a datum line on the paper and with provisions for indenting this datum line when desired, as for marking off time increments. The pencil recording the draft gear action was caused to move to one or the other side of the datum line responsive to draft gear action in pulling or buffing, the recording arm being attached to the butt end of the coupler. The original continuous records made in this test are on file in the office of the Engineer of Tests of the Norfolk & Western Railroad, points of interest being abstracted as Figs. A-3 to A-11 inclusive of this report.

The connection between the coupler butt and the recording pencil was through a reducing mechanism, so that the following scale should be used for measuring draft gear movement on the cards.

$\frac{3}{32}$ in.	offset	on record	= ¼ in.	coupler	movement
$\frac{5}{32}$ in.	"	"	= ½ in.	"	"
$\frac{9}{32}$ in.	"	"	= 1 in.	"	"
$\frac{15}{32}$ in.	"	"	= 1½ in.	"	"
$\frac{11}{16}$ in.	"	"	= 2 in.	"	"

The following tabulation gives the relative resistance of the two gears used for various amounts of travel, the loads being those obtained in the static tests and proper allowance being made for the initial compression of the Sessions gear.

Coupler Movement	*Sessions K Gear with Greased Center Friction Block*	*National H1 Gear*
¼ in.	10,400 lbs.	10,400 lbs.
½ in.	20,850 lbs.	37,850 lbs.
¾ in.	44,000 lbs.	116,750 lbs.
.93 in.	54,000 lbs.	200,000 lbs.
1 in.	59,000 lbs.	Capacity of testing machine reached at .93 in. travel of National H1 gear.
1½ in.	87,000 lbs.	
1⅞ in.	108,000 lbs. gear solid	

The compression curves, Figs. A-1 and A-2, and the above tabulation, are not to be considered as a comparison of the normal action of the two gears, as it has already been explained that the capacity of the Sessions K gear was purposely reduced for the purpose of the test.

Discussion of Cards

The portion of the record reproduced as Fig. A-3 shows the action of the two gears, beginning with the train moving on level track and showing the draft gear movements when the train was slowed down for orders and then accelerated. The record, which should be read from right to left, starts with the Sessions K gear compressed 1 in., the National gear at the same time showing 3/4 in. of compression. After building up the speed again the Sessions gear stood at 3/4 in. compression and the National at 5/8 in.

In Fig. A-4, with the train moving on a slight ascending grade, the train was brought to a stop for a red signal, the Sessions gear moving 5/8 in. and the National 3/4 in. On the succeeding start, the Sessions gear went to 1 7/8 in. and the National to 1 in. movement. The Sessions gear stuck and bombarded at two points during this pulling compression. The influence of the bombardment of the Sessions gear is manifested in the diagram of the National gear.

The card, Fig. A-5, was made when the train was slowed down for orders, the Sessions gear moving 7/8 in. and the National gear 1 in. The Sessions gear was sticking during this part of the diagram. On starting, the Sessions gear, after sticking one time, went to 1 1/4 in. and the National to 3/4 in. From the static cards there were required 2,000 ft. lb. of energy to close the Sessions gear this 1 1/4 in., and 2,053 ft. lb. to close the National gear the 3/4 in. at the same time.

The card, Fig. A-6, was produced when a stop was made from a slow speed.

The card, Fig. A-8, was obtained when the train passed through a dip in the track (Balls Hole) and shows several compressions of the gears due to the slack running in and shows also a quick pulling compression of both gears as the locomotive started the train up the grade. As the slack ran in, the Sessions gear was compressed 3/8 in. while the National gear was compressed 15/16 in. It is presumed that the greater movement of the National gear was due to the Sessions gear sticking. On the succeeding pull the Sessions moved 1/4 in., while the National moved 5/8 in. On the succeeding start, after sticking, the Sessions gear moved to 1 3/4 in., while the National gear stood at 1 1/8 in.

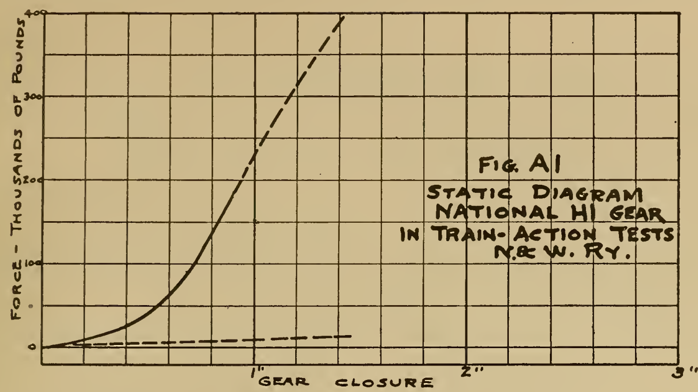
The card, Fig. A-10, was obtained when a sudden stop was made with the pusher on the rear end of the train, the pusher running in the slack against the front engine. The Sessions gear went solid, the movement being 1 7/8 in., while the National gear moved 1 7/16 in. On the succeeding start, which was made on the ascending grade, the National gear responded immediately to the amount of 1 in. movement, while the Sessions gear lagged in action and finally bombarded to 1 1/2 in. movement.

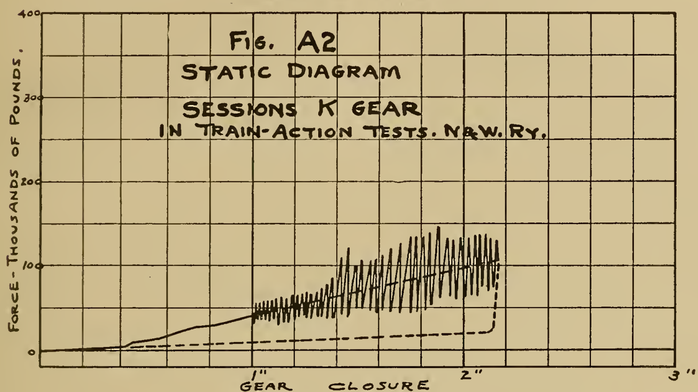
Fig. A-11 shows a typical section of record obtained going up the hill from Elliston to Christiansburg, on a steady pull and at comparatively uniform speed. Both gears stood at 1 1/4 in. movement.

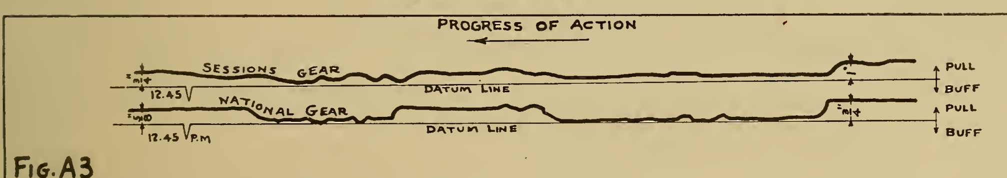
General

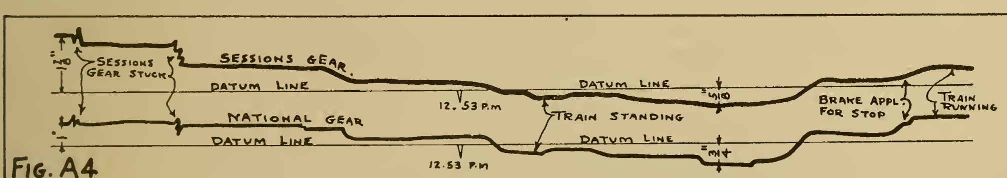
The National gear used appeared in general to be quicker in movement and more responsive to impulses than this particular Sessions gear. In pulling, it is almost invariably the case that the National gear compressed uniformly and gradually, while in most instances the Sessions gear obtained its final position after one or more bombardments. In release both gears responded almost instantly, and in the majority of cases a quick buff produced harmonious action in both gears. It is noticeable, however, that on a quick buff the National gear, even though having a stiffer resistance curve in the static machine, frequently shows more travel than the Sessions gear. With a slow buff as in Fig. A-5 the Sessions acted through a succession of bombardments.

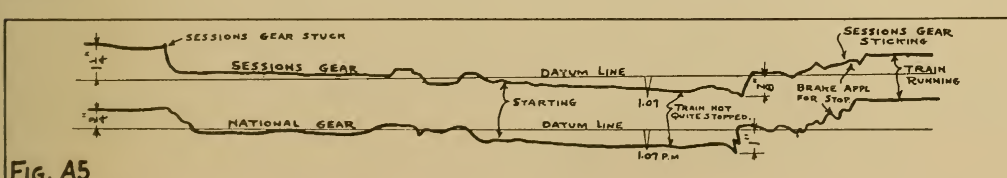
In a continued steady pull, such as represented by the lines in Fig. A-11 the absence of see-saw movements of any extent was noticeable in both gears.











Chronographic Records of Draft Gear Action in Train Service, Norfolk & Western Railway

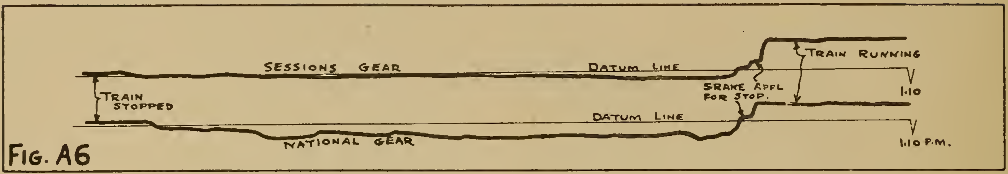


FIG. A6

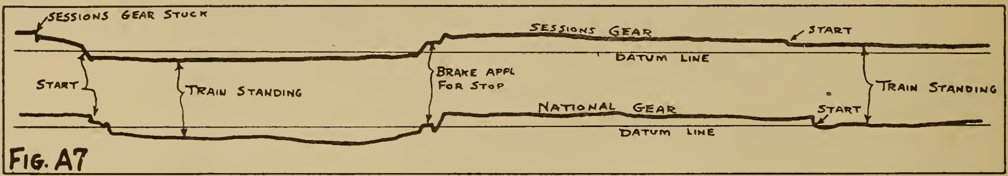


FIG. A7

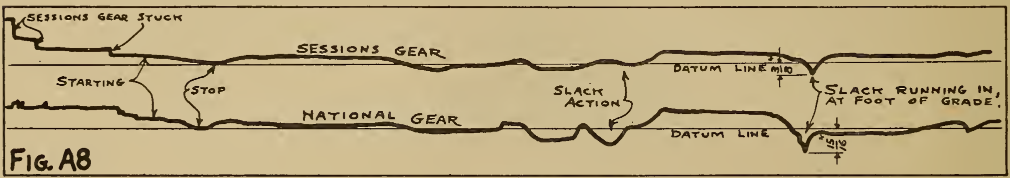


FIG. A8

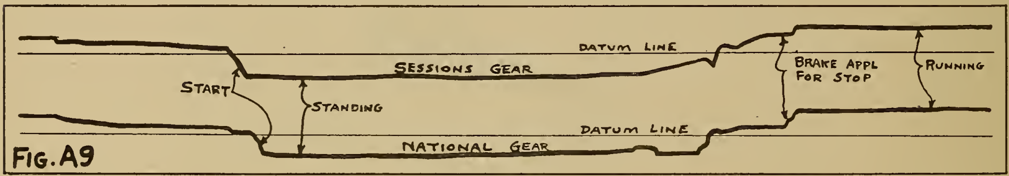


FIG. A9

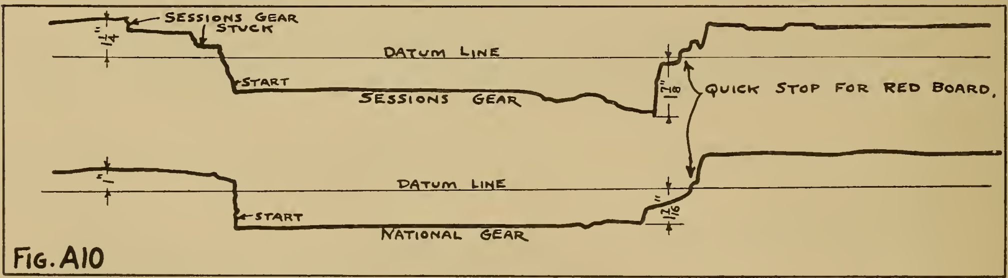


FIG. A10

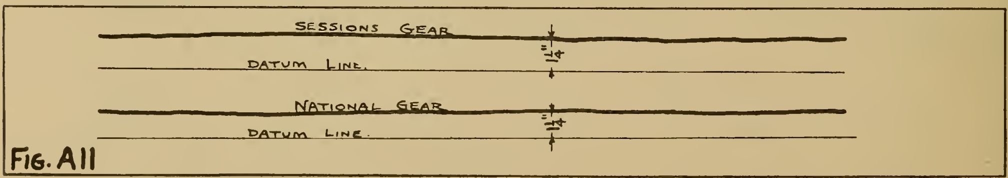


FIG. A11

CHRONOGRAPHIC RECORDS OF DRAFT GEAR ACTION IN TRAIN SERVICE, NORFOLK & WESTERN RAILWAY

APPENDIX B

TESTS OF CAR CONSTRUCTION

In accordance with recommendations of the Committee on Standards, high speed impact tests of car construction were made by the Inspection and Test Section of the United States Railroad Administration at the car impact plant of the T. H. Symington Company at Rochester, New York, February 25 and 26, 1920.

The following were present during all or a portion of these tests:

B. W. Kadel, assistant engineer, Inspection and Test Section, U.S.R.A.

E. M. Richards, special engineer, Inspection and Test Section, U.S.R.A.

L. H. Schlatter, representing Draft Gear Committee, A.R.A.

J. A. Pilcher, W. J. Robider and John McMullen, sub-committee of Car Construction Committee, A.R.A.

J. R. Onderdonk, B. & O. Railroad.

L. H. West, Merchants Dispatch Transportation Company.

B. B. Milner, New York Central Railroad.

D. S. Barrows and I. O. Wright, representing the T. H. Symington Company.

A total of four tests were made: Tests 1 and 2 to determine the value of the application of metal draft arms for the reinforcement of wood center sills. Tests 3 and 4 to show the performance of U.S.R.A. cars at high impact velocities and to determine the relative value, in buffing, of U.S.R.A. draft gear attachments having the separate rear draft lugs, and of draft gear attachments having the central back stop casting for distributing the impact force to the car sills.

Test No. 1—Wood Draft Sills

The first test was of 40-ton box cars with wood center sills, using N.Y.C. Car No. 214,423 as car A (striking car) and N.Y.C. car No. 226,768 as car B (struck car). The opposing ends of these cars were fitted with wood draft sills.

These cars have two 5 in. x 8 in. center sills, two 4½ in. x 8 in. side sills and four 4½ in. x 8 in. intermediate sills, with one-piece cast steel body bolsters beneath the sills. The draft sills extend from beneath an 8 in. x 8 in. oak end sill back to the body bolster, where they abut suitable pads cast to the body bolster. The draft sills are doweled and bolted vertically to the center sills and end sill. Malleable iron tandem cheek plates are bolted to the center sills and draft sills, and have lugs gained into both the draft sills and the center sills. Sub-sills extend from bolster to bolster beneath the main center sills and these abut suitable pads cast on the body bolster. The cars were equipped with tandem spring draft gears with 5 in. x 7 in. old-standard couplers and wrought steel riveted yokes. The coupler horn was allowed to strike a heavy cast steel striking plate, which was bolted to the face of the 8 in. x 8 in. oak end sill, the buffing force on the draft sills thus being limited to the resistance of the two Class G springs, viz., 60,000 lb. The cars had been fitted with new sills throughout for the tests, and the steel ends had just been applied. The cars were loaded with sand to give a total gross weight of 123,000 lb. per car, the sand being partly frozen in the cars.

These cars were given tests at successive impact speeds of 4 5, 7, 8, 10, 12 and 14 miles per hour.

At 7 M.P.H. the coupler heads began to scale and continued scaling throughout the tests. At 8 M.P.H. the end at the struck end of car B began to bulge out and the one at the opposite end of car B began to bulge in. At 10 M.P.H. the ends of car A began to bulge. This bulging increased throughout the test for both cars. A slippage of $\frac{1}{32}$ in. could be detected between the draft sills and center sills at 5 M.P.H., but this did not increase during the remainder of the test. A slippage of $\frac{1}{16}$ in. occurred between the cheek plates and the draft sills at 7 M.P.H., but this also did not increase as the test proceeded.

At the conclusion of the test, the striking end of car A had bulged 1⅞ in. and the struck end of car B 3¾ in. The draft sills were shattered where they abut the bolster, but no breakage of either draft sills or center sills occurred. The bolsters slipped back ¼ in. during the tests and the striking castings moved ⅛ in. each. The coupler carrier irons bent down ⅛ in. The coupler horns were not noticeably injured except for some scaling and the striking castings were in good condition. The ends of the center sills, after the test, were dropped approximately 1 in. each, but as this measurement was not checked in advance, it is not definitely known that this occurred during the test. The ends, however, scaled along the bottom edge, which indicates that these ends were straightening out and allowing the center sills to droop. Except for the bulged ends, no particular damage to these cars was apparent and they were fit for service.

Test No. 2—Metal Draft Arms

The same box cars were then shifted so as to bring the opposite ends together and test No. 2 made, N.Y.C. car No. 214,423 now being car B and N.Y.C. car No. 226,-768 car A. The opposing ends of the cars were equipped with metal draft arms, which were built up of angles and channels proportioned to just meet A.R.A. requirements. The design was made by the Inspection and Test Section and does not represent the particular details of any proprietary device. The metal arm did not abut the bolsters, but a gusset plate was riveted to the top flange of the bolster and to the bottom flange of the draft sill angle, these angles extending back 5 ft. over the bolster towards the center of the car. The tandem cheek plates were riveted to a channel below the main draft arm angles, there being no stop lugs on these cheek plates. The coupler horn was allowed to strike as in the previous test, the striking casting, however, being of malleable iron instead of cast steel. The load on the draft arms at the center line of the coupler was thus limited, as before, to the resistance of the two Class G springs.

These cars were given tests at successive speeds of 5, 6, 10, 12, 14 and 16 miles per hour.

At 10 M.P.H. the coupler heads were scaling and this scaling continued throughout the test. At 10 M.P.H. also the ends of the center sills began to droop slightly and at the end of the tests had drooped ⅞ in. on car A and ¾ in. on car B. No bulging of the ends occurred during this test, although the drooping of the sills appeared to result from a straightening of the transverse corrugations of the ends.

At the 16 M.P.H. run one of the cast steel body bolsters was broken transversely, one center sill was broken on car B and both center sills broken on car A. The center sill breakage in each instance occurred over the bolster, the crack developing from the top of the sill. No slippage of cheek plates occurred, but the draft arms as a whole moved an average of ⅛ in. with respect to the center sills. The coupler carrier irons were bent down $\frac{1}{16}$ in. and the striking castings moved $\frac{3}{16}$ in. on the draft arms.

The performance of the cars in both the foregoing tests was unexpectedly good. In each instance after the 14 M.P.H. test both cars were fit for service, the breakage of sills and bolster occurring at the 16 M.P.H. run. The fitting up of the wood draft sills was an especially good job and it is quite probable that extended service would produce looseness, which would not be the case with metal draft arms. In the limited number of tests made it was observed that neither type of construction had an especial advantage over the other. No pulling tests were made, nor was it practical to make a considerable number of lower speed impacts, which unquestionably would have produced failure. The comparative merits of the two types of construction, however, are believed to be indicated by these tests at regularly increasing speeds.

The results of these tests show the following:

1. That metal draft arms do not offer any noticeable advantage, in buffing, over properly applied wood draft arms if the latter are kept tight.

2. It should be observed that the sill breakage occurred in each instance over the body bolster, although the application of the present A.R.A. rule would indicate that the unreinforced wood center sill between the bolsters is of less value than the same sills reinforced over the bolster.

3. That it is permissible to allow the coupler horn to strike in wood car construction and probably so in steel cars with wood end sills.

4. That there is a pronounced downward force at the coupler carry iron and an upward force at the bolster which may result in deformation or breakage at both points. As both these forces must be added to the static load, cars should be constructed with bolsters rigid enough to resist the upward tendency, and the end sill and carry iron should be securely tied to the end of the car.

Test No. 3—Draft Attachments with Central Stop Casting

For this test two 70-ton U.S.R.A. low side gondola cars were used, P. & R. car 7378 being car A and P. & R. car 7379 being car B. These cars have fish belly center sills with steel sides, steel plates, drop ends and wooden floor. Each car was loaded with sand to give a total gross load of 184,000 lb. per car, the sand being partly frozen in the cars.

The cars were new and had been equipped with Farlow 2-key draft gear attachments, T. H. Symington Company's Print F-2437. Flat face dummy couplers were used instead of the regular couplers. There being no coupler horns, the entire blow was taken through the draft gear attachments. Steel blocks of 54 sq. in. cross section were used instead of draft gears, the full load being taken through this block and being delivered upon the back stop casting through the intervening parts of the attachments. The second key had ¼ in. clearance in the cheek plate key slot. The coupler shanks were made of an extra heavy design so as to reduce as far as practicable the deformation and failure of this part. The net areas of the several parts in buffing are as follows:

Dummy coupler shank, back of head, 24 sq. in. cast steel.

Dummy coupler shank at key slot, 17½ sq. in. cast steel. (Note—For reference, the type D coupler has an area of 16.9 sq. in. back of the head, and 13.4 sq. in. at key slot.)

Front follower block, 17¼ sq. in. malleable iron.

Rear follower block, 17⅛ sq. in. malleable iron.

Yoke, 1¼ in. x 5½ in. (section), 33½ sq. in. bearing area against back stop.

Back stop casting, 19½ sq. in., cast steel.

Back stop casting, 38 rivets through center sills and 4 rivets through bottom bolster tie plate, all rivets ⅞ in., total of 25.2 sq. in. in shear.

Keys, 1½ in. x 6 in.

Malleable iron cheek plates, fourteen ⅞ in. rivets each.

The cars were given tests at successive speeds of 4, 5, 6, 8, 10, 12 and 14 miles per hour.

At 6 M.P.H. the couplers started to scale and deform at the key slots, this deformation continuing throughout the test. At 8 M.P.H. the front portions of the back stop castings showed slight scaling. At 10 M.P.H. this scaling became pronounced and continued throughout the remainder of the test. At 10 M.P.H., also, three rivets at one diagonal brace sheared off and others of these rivets had loosened.

At the conclusion of these tests the following conditions were found:

Condition of Cars

The opposing drop ends of the cars had bulged out, both at the top and bottom. In car A the bulging amounted to 3¼ in. at the top and 2¼ in. at the bottom. In car B it amounted to 1¾ in. at the top and ¾ in. at the bottom. On both cars the corner posts, which are formed of heavy bent plates and serve as stops for the ends, were bent from the impact of the load. The upstanding legs of the end sill angles were also bent out from this same force.

On both cars the body bolsters at the opposing ends of the cars were bent down at the ends, equivalent to the centers of the bolsters being forced upward. In car A the center sills were also bent slightly from this same condition. The entire ends of the cars were down $1\frac{3}{16}$ in. for car A and $\frac{5}{16}$ in. for car B. The end sill of car A was bowed inward ⅞ in. and that of car B, 1 in. Neither of the end sills were bowed down.

On car B one of the diagonal braces was sheared and torn loose and all diagonal braces were either scaling or had loose rivets. The floor boards of both cars had shifted 1½ in. and the floor clips were displaced. These floor clips began to drop off early in the test and do not appear to be a satisfactory type of construction. The floor boards of both cars were crushed at the bolsters and at the end sills from shifting. One intermediate wood sill of car A was shattered from the same cause.

At two points on the bolsters of car A cracks developed at rivet holes through the flanged bolster webs. These cracks resulted from the horizontal bending of the bolster when the sides of the car attempted to run ahead of the center sill. No spreading of the center sills occurred.

Condition of Coupler and Draft Attachments

Dummy Couplers, cast steel — Shank bent both vertically and laterally, and upset and deformed at key slots. Shortened an average of ¾ in. each.

Cheek Plates, malleable iron—No failure or injury of any kind. Second key had been bearing slightly, indicating momentary elastic compression of parts.

Back Stop Castings, cast steel—Slipped on rivets 3/64 in. Front end upset 7/64 in. Not injured perceptibly and removal or repairs unnecessary, except that two rivet heads jumped off at the final run.

Yokes, wrought steel—No failure or injury of any kind.

Coupler Keys, wrought steel—Not bent or injured.

Second Keys, wrought steel—Bent an average of $\frac{3}{32}$ in. each. Serviceable without repairs.

Front Follower Blocks, malleable iron—Shortened $\frac{1}{16}$ in. Not injured perceptibly. No repairs necessary.

Rear Follower Blocks, malleable iron—Shortened $\frac{1}{32}$ in. Not injured perceptibly. No repairs necessary.

In this test the cars suffered more than the draft gear attachments. It is noticeable that not a single part of the attachments was damaged to an extent requiring removal or repairs during this test, and that the draft gear pockets had elongated but $\frac{5}{32}$ in. each.

The car damage was greater to car A (striking car) than to car B (standing car). The back stop castings of the attachments first beginning to scale at 8 M.P.H. this point is taken as the comparative critical speed for these attachments, and a value of 64, or the square of 8, is accordingly set for these attachments.

Test No. 4 — Attachments with Separate and Independent Draft Lugs

Two of the U.S.R.A. 70-ton low side gondolas were used for this test, the cars being new and having the regular U.S.R.A. cast steel yoke and draft gear attachments. P. & R. car 7381 was used as car A and P. & R. car 7380 as car B. Each car was loaded with sand to give a total gross load of 184,000 lb. per car, the sand being partly frozen in the cars.

These cars have the regular front and rear cast steel draft lugs riveted to the center sills, the rear lugs each having twelve $\frac{7}{8}$ in. rivets and the front lugs ten $\frac{7}{8}$ in. rivets, and three $\frac{3}{4}$ in. rivets each. The rear draft lugs, from the drawings, extend to within $\frac{1}{4}$ in. of the bolster center casting, which has twelve $\frac{3}{4}$ in. rivets through the center sills and four $\frac{3}{4}$ in. rivets through the bottom bolster tie plate. The same steel blocks of 54 sq. in. cross section were used instead of draft gears, as in the previous test, there being the regular $2\frac{1}{4}$ in. followers in front of and behind these blocks, bearing upon the stop faces of the draft lugs. The net bearing area of the followers upon the two lugs, in buffing, is 50 sq. in. The lugs are ribbed to support this bearing surface. A tie plate extends across the bottom flanges of the center sills beneath the draft lugs to reduce the spreading tendency of the sills from the eccentric loading upon the lugs. Dummy couplers with flat buffing faces were used in these tests, these being duplicates in every respect of those used in test No. 3. The full buffing force was delivered as before, through the steel block to the rear stops.

The cars were given tests at successive speeds of 4, 5, 6, 8, 10, 12 and 14 miles per hour.

At 6 M.P.H. the dummy couplers began to scale at the key slots, and scaling and deformation at this point continued throughout the tests. At 8 M.P.H. the opposing ends of the cars were bulged. At 10 M.P.H. the body bolsters were bent slightly.

At 5 M.P.H. the rear lugs had slipped $\frac{1}{8}$ in. on the sills and the stop faces had begun to deform. At 6 M.P.H. the lugs had bent and pulled away from the center sills $\frac{1}{8}$ in. and the draft gear pockets had elongated $\frac{7}{32}$ in. This bending and deformation of the draft lugs increased as the test proceeded, and at the 14 M.P.H. run both of the lugs of car A, and also the bolster center casting, were sheared off and driven back between the sills; the truck center pin also sheared off. From this failure the dummy coupler of car A was also driven back, bending the carrier iron and carrier iron bolt, and breaking the striking casting. The coupler key was bent and the front draft lugs broken away, the key being driven back through the webs of the center sills for 3½ in. On car B one of the rear draft lugs broke at the 12 M.P.H. run, but these lugs were not sheared off, although they slipped on the rivets ¼ in. each. At 8 M.P.H. the rear followers had bent ¼ in. each, bending the draft lug faces also and slightly deforming the webs of the center sills.

At the conclusion of the tests the following conditions were found:

Condition of Cars

The drop ends at the opposing ends of the cars were bulged, that of car A being bulged 3 in. at the top and 2⅜ in. at the bottom. In car B this bulging amounted to 4 in. at the top and 1¾ in. at the bottom. The corner posts were bent as in test No. 3, as well as the upstanding legs of the end sill angles. The ends of the bolsters were bent downward ½ in. in car A and $\frac{5}{16}$ in. in car B. The sills were slightly bent in front of the bolster, the effect being as though the center of the bolster was forced upward. On car A the bolster center casting was driven back and on car B it had slipped ⅛ in. on the rivets. The end sills were not bowed downward, but were bowed inward an average of ⅝ in. The center sills were pushed through the cars an average of $\frac{5}{16}$ in., the diagonal braces being in better condition than in test No. 3, although they showed evidence of failure and loose rivets. The floor boards shifted as in the previous test and the floor clips loosened.

The center sills of car B were buckled ⅜ in. at the bottom flange near the rear draft lugs, those of car A being buckled ⅛ in. The sills were spread an average of $\frac{7}{32}$ in. at the rear draft lugs. The bending of the body bolsters reduced the total side bearing clearance of each of the trucks at the opposing ends of the cars by $\frac{5}{16}$ in. During the test the draft gear pocket of car B was elongated ⅞ in. and that of car A was entirely destroyed.

Condition of Coupler and Attachments

Dummy Couplers, cast steel — Shanks bent both vertically and laterally, and upset and deformed at key slots. Shortened an average of ⅝ in. each.

Front Draft Lugs—Destroyed in car A. Not injured in car B.

Rear Draft Lugs — Destroyed in both cars.

Cast Steel Yokes—Not injured. (Note —These yokes do not come into action in buffing.)

Coupler Keys—Badly bent in car A; required to be burnt out. Not injured in car B.

Front Followers—Not injured.

Rear Followers—Bent ¾ in. in car B. Badly bent in car A. Can be repaired.

Bolster Center Casting—On car B slipped ⅛ in. on rivets. On car A sheared off and bent. Can be straightened and reapplied.

Truck Center Pin—Sheared off. Cannot be used.

Striking Plate—Broken. Can be used.

Carrier Iron—Bent. Can be used.

Carrier Iron Bolt — Bent. Cannot be used.

In test No. 4 the greatest injury was to the draft gear attachments, the majority of parts requiring removal and renewal. Both cars were in bad order after the tests. The damage to the attachments was greater for car A than for car B, while the car damage was probably greater for car B.

The rear lugs of this form of attachment having begun to deform at 5 M.P.H. and to actually bend away from the sills at 6 M.P.H., the greatest critical speed that can be set for them is 6 M.P.H., or a relative value of 36, as compared with 64 for the Farlow attachments used in test No. 3. In basing relative values upon the square of the speeds, it should be remembered that the energy is proportional to the square of the speed, or, in other words, that a car moving at ten miles per hour will roll four times as far as one moving at five miles per hour. An experienced car rider has an instinctive knowledge of this fact in its relation to the kinetic energy of the car, as exhibited by the force with which he applied the brakes under varying speeds.

In these tests, as in tests Nos. 1 and 2, it is unquestionable that a repetition of impacts at lower speeds would have produced failure, but, as before, it is believed the results obtained in these tests represent the comparative value of the two forms of attachments, namely, that the Farlow attachments as tested showed approximately twice the buffing value of the cast steel yoke and lug attachments:

From the results of the test it is apparent:

1. That the buffing force should be distributed to the car sills through a back stop casting bridging between the sills, rather than upon independent draft lugs riveted to each sill.

2. That if the draft gear is to be protected by allowing a front key to strike, there should be substantial members on the sills for stopping the key.

3. That in car construction it is necessary to give consideration to the results of impact when designing the body bolster for vertical loads.

4. That it is important properly to anchor the car floor and superstructure to the center sills in order properly to impart motion to the lading from the center sills.

5. That in cars with wood floors, or open type floors such as hopper cars, particular attention should be given to the diagonal braces in order that the car sides and center sills may be held from independent movement.