THE USE OF A 1/4 SCALE IMPACT TEST PLANT FOR SIMULATING RAILROAD SWITCHING OPERATIONS

Thesis for the Degree of M. S.

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ABSTRACT

THE USE OF A 1/4 SCALE IMPACT TEST PLANT FOR SIMULATING RAILROAD SWITCHING OPERATIONS

by Palle Jacobsen

With the objective of predicting damage potential caused by humping and switching operations performed in practice on railway freight cars, a laboratory scale railway test track with car and lading was investigated as the model system.

Being interested in an output (readings concerned with damage) from the model similar to, or with a simple calculatory relationship to the real system during operation, we tried to compare the components of the input to the two systems.

Impact conditions, input-forces to the lading and self-excited movements in the lading all constitute factors, that must be assumed to be important, with a lack of comparable information on the detailed behavior of the lading as a function of the switching operations alone, our attempt was to discuss and make suggestions in order to adjust the model from an environmental analysis-standpoint, involving parameters such as

- impact velocity
- magnitude, shape and duration of impact forces
- impact quantum (equivalent mass of striked cars)
- rebound velocity
- movement during and after impact
- magnitude and duration of input forces.

Having reached the input to the lading quite successfully, we proposed direct and indirect simulation procedures using items or model items from which it should be possible to describe damage. Correlation to practice is practically unknown, and parameters such as

- superimposed vibrations on impact
- self-excited random vibrations in the load caused by internal sliding, etc.

seemed impossible to simulate in the laboratory. But we still suggest that proposed procedures - after proper adaptation of the test track - such as:

- securing strictly horizontal impact and movement of all outer components
- rebuilding backstop to perform as 3 or 4 striked cars in actual operations, preferably with no rebound travel
- reinforce construction and hauling devices to operate at least masses of 1200 lbs. securely

will make a valid prediction of damage possible for a wide range of commodities being transported by the railroad.

Air-cushions included to divide lading into sections were investigated and showed up to have a decreasing effect on the magnitude of input forces to the lading. Further investigations are suggested including other types of loads than wooden blocks, preferably with a controlled type of internal friction, using unitized loads and smoother sliding surfaces. Also the effect of unitizing, bracing and fastening means such as steel accessories, non-skid plates, anchor plates and retarder plates in connection with steel strapping and wooden type dividers can be investigated and compared to air-cushions.

Palle Jacobsen

An attempt to reproduce the damage of canned goods in fiberboard boxes were not successful, but damage was obtained by exposing one or two cans to the maximum obtainable input-forces.

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By

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1. Objectives

The purpose of the present study is to analyze and to some extent develop an approach in the field of package testing, namely simulation of railroad humping and switching impacts by means of a laboratory scale test plant. The test plant was originally built by "The Chesapeake & Ohio Railway Company" (1)[1], which has donated it to the "Packaging School" at Michigan State University with the expectation that further development would take place. Figure 1 shows the built-up of the test incline, backstop and car.

The present study in the first part is devoted to development of a reasonable input to the test car with its load. Comparisons are made to field conditions[2]. Next the transmitted pulses are measured to the load of wooden blocks with and without air cushions (pneumatic dunnage) in different numbers, locations and exerting different pressures. Finally a few remarks are presented explaining which procedures may be used in the simulation of common types of damage observed during railroad transportation. One example on reproduction of damage is given.

Some suggestions are made for improvement of the test device itself, as well as proposals for further work with the railroad.

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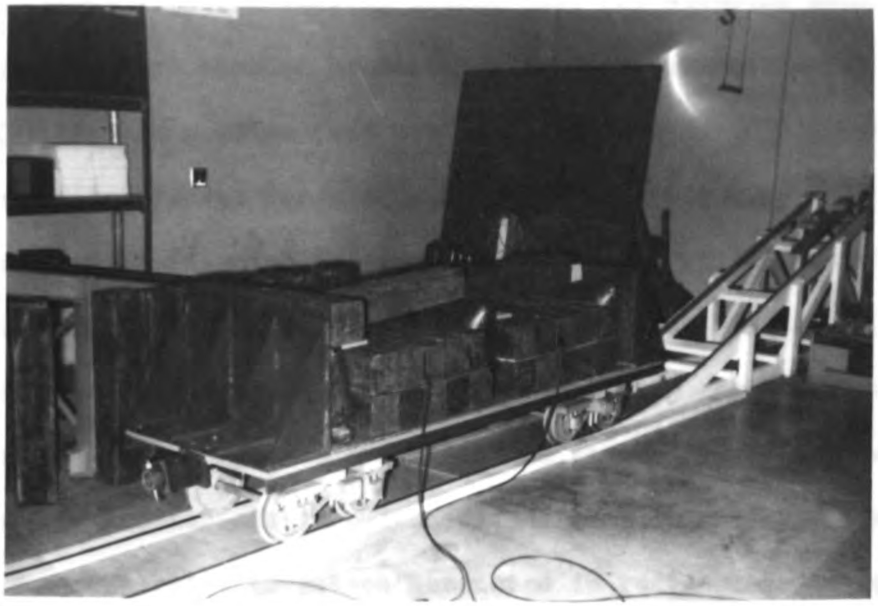
[1] Numbers in parenthesis here and in the following refer to the references or bibliography back in the thesis.

[2] Information given by courtesy of Association of American Railroads (AAR), 59 East Van Buren Street, Chicago 60605, Ill.

Figure 1



Test track with car and load. Including TRELLEBORG air-cushions and two block load cells.

Figure 2

Instrumentation. Counter and relay for switches. Amplifiers, oscilloscope and POLAROID camera. Backstop of test track and light beam trigger.

2. General Background

2.1 Since 1963 the incline impact test for shipping containers (the Conbur-tester) adopted by American Society for Testing and Materials (ASTM) (2) has had a widespread use in the U.S.A. as well as overseas. The standard being, set first of all to secure uniformity of the apparatus used, defines an impact angle of 10°, a rigid wooden bumper perpendicular to the movement of a dolly, and a calibration in terms of impact velocity when the empty dolly is hitting the rigid bumper. Although the device may have some value in comparison of one container or package to another, it is quite evident that it will not necessarily give pulses comparable to pulses generated in real switching operations. As a matter of fact, measurements by use of an Endevco piezoelectric accelerometer (3) have shown that the load-factors (G-factors) are much higher and the durations of impact shorter at a given impact velocity with the rigid wooden backstop than it is when cars are hammering each other with buffers between them. On the other hand, the forces due to the enormous masses behind an actual rail impact are much higher than at the Conbur-tester. According to the concept, that (a) duration (frequency), (b) magnitude (force, acceleration, displacement or similar) and (c) shape of the pulse (eventually expressed as the area) together determine the potential risk of damage for the particular container or package, it will be evident that the Conbur-tester has several shortcomings.

Also in recent years, some companies built inclines with the backstop as a sandwich construction of steel plates on a core of hard maple. This would give a more resistant surface and be similar to common drop test operations made against steel plates. This, of course, will again change the pulse developed, and control will be difficult to obtain.

The most interesting attempt presently being made might be supplementing the dolly of the Conbur tester with hydraulic type shock absorbers[3] that are able to generate almost any type of pulse as known from, for example, the HYGE-shock-machine[4] and the Monterey type drop tester. The pressure in hydraulic cylinders and the size and shape of orifices between cylinders determine the magnitude and shape of the pulse (4).

2.2 A true simulating device undoubtedly is being provided by using a full scale ramp with striking car and hammer car plus instrumentation. The only disadvantage here would be the space occupied and the price for establishing such a device (about 1 million dollars (5).)

Several companies do have full scale test ramps[5] that are used not only for direct measurement and simulation, but also as "calibration devices" for developing smaller scale test ramps. This last mentioned principle might be the way to go in the future, not only for calibrating laboratory scale devices according to the input in practical operations, but also according to damage observed. The only approach in that direction published for normal freight commodities seems to be du Pont de Nemour's (7) dealing with bag-damage.[6]

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3) Information obtained by the courtesy of Gaynes Testing Laboratories, 1642-45 West Fulton Street, Chicago 12, Illinois.

4) Information obtained by the courtesy of Dept. of the Army, Picatinny Arsenal, Dover, New Jersey.

5) Signode Steel Strapping Corp. Miner Corporation (5). Pullmann Standard (6). E. I. du Pont de Nemours & Co. (7). Aberdeen Proving Ground. Sandia Corp. (8).

6) Also published in ASTM Special Technical Publication No. 324 "Simulated Service Testing of Packaging".

PART A

ENVIRONMENTAL ANALYSIS AND CHOICE OF TEST DRAFT GEAR, ETC.

Statement of problems

In the earlier work with the 1/4 Scale Impact Test Plant (1), different lengths of hard rubber (Neoprene, Durometer A-Hardness 70) in the form of O-rings 3/4 in. thick and with soft rubber (Durometer-4) sponge rings 1/4 in. thick between each two of the hard rubber rings were used. These draft gear components were mounted on a hollow steel cylinder sliding against brake linings made of asbestos filled, hard textile material underneath the test car. It is possible to tighten the brake linings, but not in a very satisfactorily defined way. The rubber components accounted for most of the energy absorption capacity of the test draft gear.

There are several types of draft gears in practical usage:

(a) Standard gears characterized by a travel of 2 - 6 1/2 in. (oftentimes about 4 in.) between coupler and body in either direction. These gears are further specified as having a minimum energy absorption capacity and a maximum sill-pressure (for example, specification AAR M-901E-62: Minimum capacity 36,000 ft.-lbs., maximum reaction force 500,000 lbs.) by the use of certain qualification tests (9).

(b) Special cushioning devices, including sliding center sill, end of car devices or column-connected draft gears. These gears have combined travel of more than 5 in. up to about 36 in. or even 48 in. About 25% of 80,000 cars purchased in 1965 were cushioned cars, but still the main part of about 1.5 million railway cars in the U.S.A. consist of the conventional type equipped with standard draft gears. A cushioned car may cost twice as much as a conventional car. Several firms are manufacturing draft gears,[7)] and the Miner draft gear is

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7) Miner Corp., 209 S. La Salle Street, Chicago; Waugh, 332 S. Michigan Ave. Chicago; National Castings, 224 S. Michigan Ave., Chicago; Cartwell Westinghouse, 332 S. Michigan Ave., Chicago.

one of the most widely used. It is also used in some places in Europe. Typical characteristics for Miner type No. RF 4-29 for example are a travel of 4.33 in., capacity 42,000 ft.-lbs., and sill pressure 380,000 lbs. The characteristic pulse under standard impact conditions (hammer test) is described under (a) below.[8)]

It is clear that about 2/3 of the energy absorption by the chosen type of standard draft gear is due to friction between brake shoes, and that only 1/3 is energy-storing by spring action. In our simulation we are unable to reproduce such a high relative amount of friction; therefore, the obtained pulses show a characteristically different shape:

(a) The standard rubber-friction gears show a moderate increase in the coupler force in the beginning, then a steep increase until the maximum force, and thereafter a gradual decrease with a considerable drag out of the pulse. It is remarkable that not only Miner Corp., but also research authors such as Baillie (10) and Borgeveen (11) indicate that type of pulse as typical. On the other hand, these "clean" (noise-free) pulses are not taking into account the superimposed vibrations that occur under impact conditions (12) which may represent a damage potential for certain types of commodities.

To simulate and describe also the superimposed vibrations, instrumentation may be required such as oscillographs (whereas in the present study, we have only oscilloscopes), from which shock spectra may be derived by means of a tape-programmed electrodynamic shaker. (See reference (8) and the Bibliography (13) - (17) for shock spectra.)

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8) Information given by the courtesy of R. H. Miner, Inc.

(b) Despite the limitations touched upon , the pulse derived from rubber draft gears (1) may be made acceptable by proper arrangement for use as inputs. In the following section, the experimental results are discussed in detail for different types of spring type draft gears.

The duration of a normal impact pulse is considered by most authors to be 75-100 msec, and it was possible to obtain that duration fairly accurately by using a 6 ring rubber test gear.

The magnitude in terms of load factor (G's) goes from 2-3 g in. normal commercial operations, involving 4-6 mph coupling speed (the allowable speed in coupling is 4 mph, which is observed in most automatic or semi-automatic switching operations in switching yards) to 30 g's or more in Military operations, where a possible impact speed of 10 mph is taken into account (18). A requirement of 8 mph as design criteria and limitation of the carrier operation to a maximum of 6 mph is proposed in the NASA Report HR 1262 (19).

It was decided to present a characteristic of the test device forces as a function of impact velocities and the weight of load (consisting of wooden blocks rigidly mounted between the two bulk heads). This enables us to choose the magnitude that is desired for the particular simulation.

With the available instrumentation, it is difficult to compare test pulses with actual pulses as far as vibrations are concerned. So it was decided to clean the pulse rather than have an undefined noise superimposed on it. A felt damper freely mounted at the backstop was very effective in that respect.

It is convenient to use a standard pulse in laboratory simulation rather than the more complicated pulse shape occurring in the field, according to the opinion of several authors (20) (21). However, the more ideal method for the purpose of direct simulation would be to use the shock-spectrum derived as mentioned before from the actual impact conditions. Also extension of the shock-spectra concept covering a two degree of freedom-system has been suggested (19). When using the shock-spectra as the basic input, expensive instrumentation and highly skilled personnel would be necessary.

Experimental work and calculations

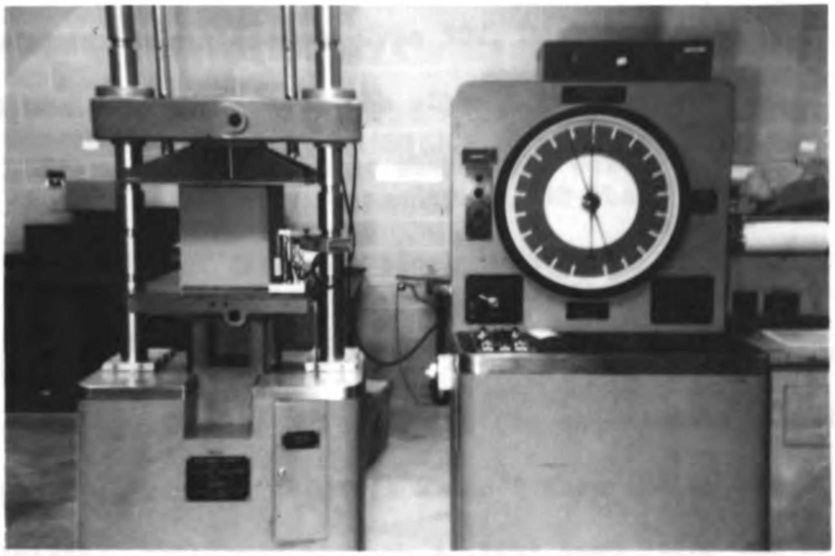
Methodology

The incline is marked with Positions 0 - 6. They correspond to vertical heights above the floor level as follows:

Position	In.
0	9.4
1	13.4
2	17.5
3	21.5
4	25.5
5	29.6
6	33.8

At a relatively early stage of the experiments, it became clear that Position 0 gave too large an uncertainty in the impact velocities V_i measured at the end of the course, and Position 6 could hardly be used because of the lack of capacity of the pulling mechanism. Position 6 corresponds to a 6.0 mph impact velocity. For the higher loads Position 5 was not used either.

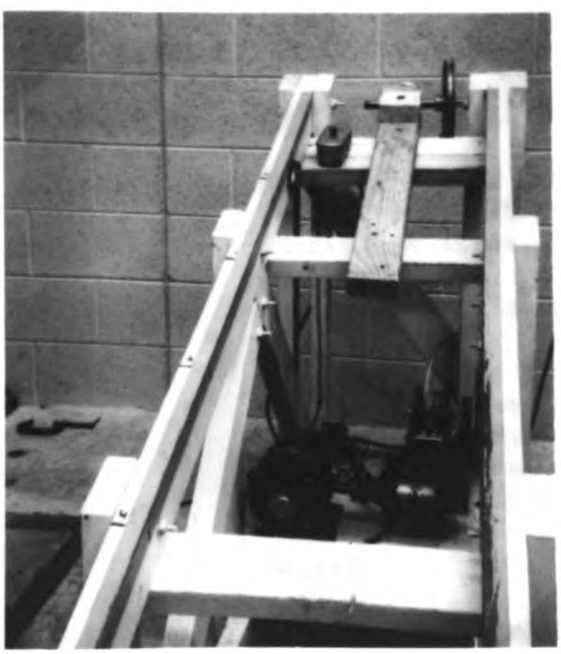
Positioning the car at the incline was accomplished by the use of a stop, positioned at a 90°-angle with the back end of the car. The theoretical impact velocities then were calculated and compared to the velocities measured (Appendix I).



BALDWIN-EMERY Compression Machine

Figure 3

Pulling mechanism of the test track.



A Tektronix Oscilloscope, Type 502, Dual Beam with a camera (POLAROID) was used to monitor the signals from strain gage load cells:

1 tubular shape load cell;
top capacity 30,000 lbs.

2 wooden block load cells;
top capacity 1,500 lbs. each.

via amplifiers PN 153 X and PN 125. The load cells with respective amplifiers were calibrated twice during the experiments at BALDWIN-EMERY Static Load Tester. Figures 2 and 3 show the apparatus arrangements. The oscilloscope, triggered through a light beam being cut by the front wheels of the car, placed the impact pulse at a convenient position on the oscilloscope screen.

Draft Gears tested

Tables 2 and 3 show the different types and sizes of draft gears tested, with the picture reference to the collection of pictures in Appendix II and Table 4. Figure 4 shows from the left to the right:

On the table:

Asbestos-filled brake lining
Tufflex pad, after testing (above)
Glassfiber rings
Hard felt pads
Standard shock absorber
Neoprene rings
Silicone-rubber rings
Blue Polystyrene (closed cells)
White Polystyrene (open cells)
Polyethylene-foam
"True Mark" felt pads

Behind:

Rubber-steel sandwich pad.

Materials used in experiments (see text).

Figure 4

Full size paper bag after impact, position 4.

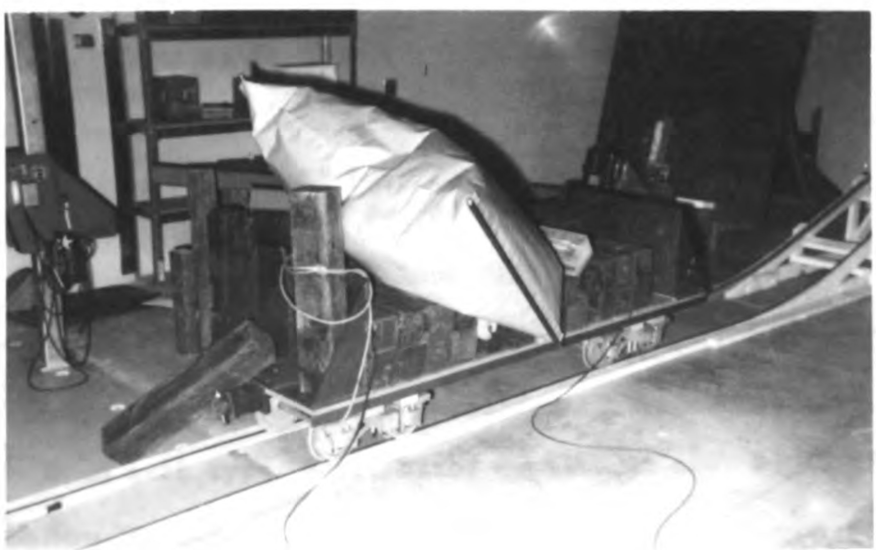


Table 2. Neoprene and Combinations with Neoprene

Draft Gears

No.	Composition	Picture Reference
1	12 Neoprene[x)] with 1/4" Sponge Rubber between	1
2	Same as No. 1, though 1/2" glass wool felt in front, only 11 Neoprene	2
3	8 Neoprene with Sponge Rubber between	3
4	Same as 3, though 1/2" glass wool felt in front	4
5	As 3, plus 3 x 1" Tufflex in front	5-6
6	As 3, plus 3 x 1" polystyrene[xx)] in front	7-9
7	As 3, plus 1" polyethylne foam[xxx)] at the backstop	10-11
8	6 Neoprene with Sponge Rubber between	12
9	Same as 8, plus 3 x 1" Tufflex	13
10	Same as 8, plus 3 x 1/2" felt pad spot-glued on backstop[xxxx)]	14
11	As 8, plus 3" felt (clothing type, soft), plus 1/2" felt pad on backstop	15-17
12	As 11, though without felt pad on backstop	18

x) Durometer hardness 70, durometer A. Thickness 3/4".

xx) Open cell type, Dow, white, density as new: 0.043 g/cm^3.

xxx) Dow "Ethafoam", density: 0.040 g/cm^3.

xxxx) "True Mark" pad for office machines, hard; Distributor: University Typewriter Co., 1912 East Michigan Ave., free to move horizontally except for the 3-4 spots of glue to the front face of the backstop.

Table 3. Other Materials in Draft Gears

No.	Composition	Picture Reference
15	10 x 1/2" felt plus 3 x 1/2" felt on backstop	24-25, 27
16	Same as 15, though 2" sandwich rubber pad x) on backstop	29
17	10 x 1" polystyrene xx)	30, 32-34
18	6 x 1" polystyrene (continued from 17)	36-37
19	6 x 1" polystyrene plus 3 x 1/2" felt damper on backstop (continued from 18)	38-39
20	Same as 19, after 48h in compression (continued from 19, half length)	40
21	8 x 1" "Ethafoam", plus 3 x 1/2" felt damper	41, 44
22	3 x 1" "Ethafoam" plus felt damper (continued from 21)	45-46
23	2 x 2 1/4" Silicone Rubber xxx)	47,48,49,50
24	Commercial types hydraulic shock absorbers	None xxxx)

x) Used on Drop-tester. One random sample of steel-rubber sandwich-construction.

xx) Dow "Styrofoam", blue, density: 0.039 g/cm^3.

xxx) Dow "Silastic 583 RTV" (selfcuring compound) prepared after 24h hardening at room temperature and 48h hardening at 100°F as recommended.

xxxx) "Volkswagen" standard absorber, 1 1/2" diameter, and GABRIEL Type 4DC (were both bottoming very fast).

Table 4. Pulse Characteristics for Selected Draft Gears

Gear No.	Picture No.	Max. Force (Faired Height) Lbs.	Duration msec t_{1o}	Duration msec t_o	Remarks x)
1	1	2300	155	160	Haversine; noisy
3	3	2750	105	120	Close to linear; noisy
8	12	3100	85	100	Parabolic; noisy up t_{oR}= 40 msec: drag-out
10	14	2750	110	120	t_{oR}= 60 msec; symmetrical; no noise
4	4	2550	110	125	Some noise, most upwards
5	6	2700	100	115	Noisy, both up and down
6	9	2750	105	120	t_{oR}= 45 msec; some noise
7	11	2400	125	140	Little noise
11	16	2600	100	120	Low frequency noise; only up
12	18	2700	100	115	Noisy; more than picture 17
5	5	2200	130	180	Haversine, almost symmetrical; low-magnitude noise
6	7	1700	145	160	t_{oR}= 120 msec, triangular pulse, noisy in 20-40 msec

x) The pulse characteristics are defined in a SANDIA Corp. proposed standard (25). In describing shape ΔV would not suffice, when comparing different materials like here. Therefore the attempt to indicate, as the remarks show. t_{oR} indicates rise time.

Comments to Table 4

Effect of length

There was a change of shape from almost haversine to almost parabolic with significant drag-out, when going from 12 to 6 Neoprene-rings. In all cases, the noise in the first 15-5 msec is of high frequency and has a magnitude up to almost the peak force.

Effect of dampers

The 1/2 inch glass wool had little influence, and the 3 inch Tufflex and 3 inch polystyrene had no influence, except in the very first impact. The polyethylene were damping some noise in the beginning of the pulse. The resilient type of rubberized felt was very effective. The clothing type felt was effective only in the first couple of impacts, where it cleaned for noise of higher frequency than that of about 100 cps.

Effect on magnitude and duration

Polyethylene drags out the duration the most. The glass-wool will do some. The inverse relationship between pulse duration and magnitude holds.

Degradation

The normally accepted cushioning materials such as Tufflex and polystyrene had an energy absorption effect (most polystyrene) and damping effect (most Tufflex) in the first blow and showed the characteristic shapes only in that first blow. The effect of high mass plus low G-factor (compared to drop) made these cushions useless, although the size-order of the force was the same as in drop-test operations.

Comments

All draft gears were tested from position 2 on the incline with an empty car (say $V_1 = 3.0$ mph, $M_2 = 575$ lbs.). The thinking behind this was that if the different types of relatively soft materials do not act satisfactorily at these relatively small impact momentums, they must be excluded. If one or more of the combinations acted good enough at the stated momentum, they ought to be tested through a whole range of momentums.

The draft gear originally supplied with the apparatus made of 12 Neoprene rings gives a pulse duration of 160 msec. See Table 4. According to the formulaes:

$$G_m = \sqrt{2\,h\,k_2 \,/\, W_2} \text{ and } T_2 \,/\, 2 = \pi\sqrt{M_2 \,/\, k_2}$$

for linear elastic systems without damping, we get:

$$F_m \cdot T_2 \,/\, 2 = M_2 \cdot \pi\sqrt{2h \,/\, g} = \text{Constant}$$

(independent of spring constant).

We therefore assume that no single elastic or nearly elastic material will be able to decrease the pulse duration without increasing the maximum force, and the maximum force divided by load (G-factor) is already fairly high compared to the actual environments.

(1) We tried softer materials having the possibility of high damping which could act in a shorter time, compared to the rubber gear, by the same energy absorption. These included glassfiber, felt materials, Tufflex and plastic-foams, and plastic materials such as lead (22) (23).

certain

(2) The other possibility is to use non-linear springs. These are able to produce square pulses or at least pulses showing considerable hyperbolic tangent elasticity which means a relatively low F_m, if bottoming does not occur. Hydraulic devices, as mentioned before, may be useful; and also, certain polyurethane constructions[11] could have the desired properties. The fact that Miner Corporation is experimenting with polyurethane-steel sandwich construction for draft gears supports the desire to try that material. Unfortunately, the time did not allow for the manufacturing of O-rings from polyurethane. The use of specially designed fluid springs, DIENE (Butadiene rubber with 30% oil from Goodyear) confined into a convenient shape, would be still another valuable attempt in the direction of square-type-pulses (24).

Choice of gear

Unfortunately, the softer materials and combinations tried showed up to be somewhat degrading (see also Appendix III) or plastic, so that they could be used only once, or a few times. The best material was felt; although, it did not give sufficient decrease in duration without increasing the maximum force. It had a very valuable effect, namely, the cleaning of the pulse for the undefined noise especially in the beginning. Three inch felt was mounted by spotgluing to the tubular cell on the backstop in all further work.

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11) For example, ESTANE 5701 Resin, manufactured by B. F. Goodrich Chemical Company, Cleveland, Ohio.

After all, the negative result for simulating pulses just by using conventional spring-type materials was not surprising, when considering, that the actual draft gear construction includes a considerable amount of energy absorption by friction.

The best available draft gear chosen for use was a 6 ring rubber draft gear with a travel of about 2.0 in. by a 2800 lb. load. The travel was measured by using the light beam-triggering system, moving it from the 0-position (the point of impact), to the position where it was not switched off at a certain impact. Table 6 shows the final characteristic values for the test track using this draft gear, and Figures 5 and 6 are corresponding curves (see Appendix IV for further discussion).

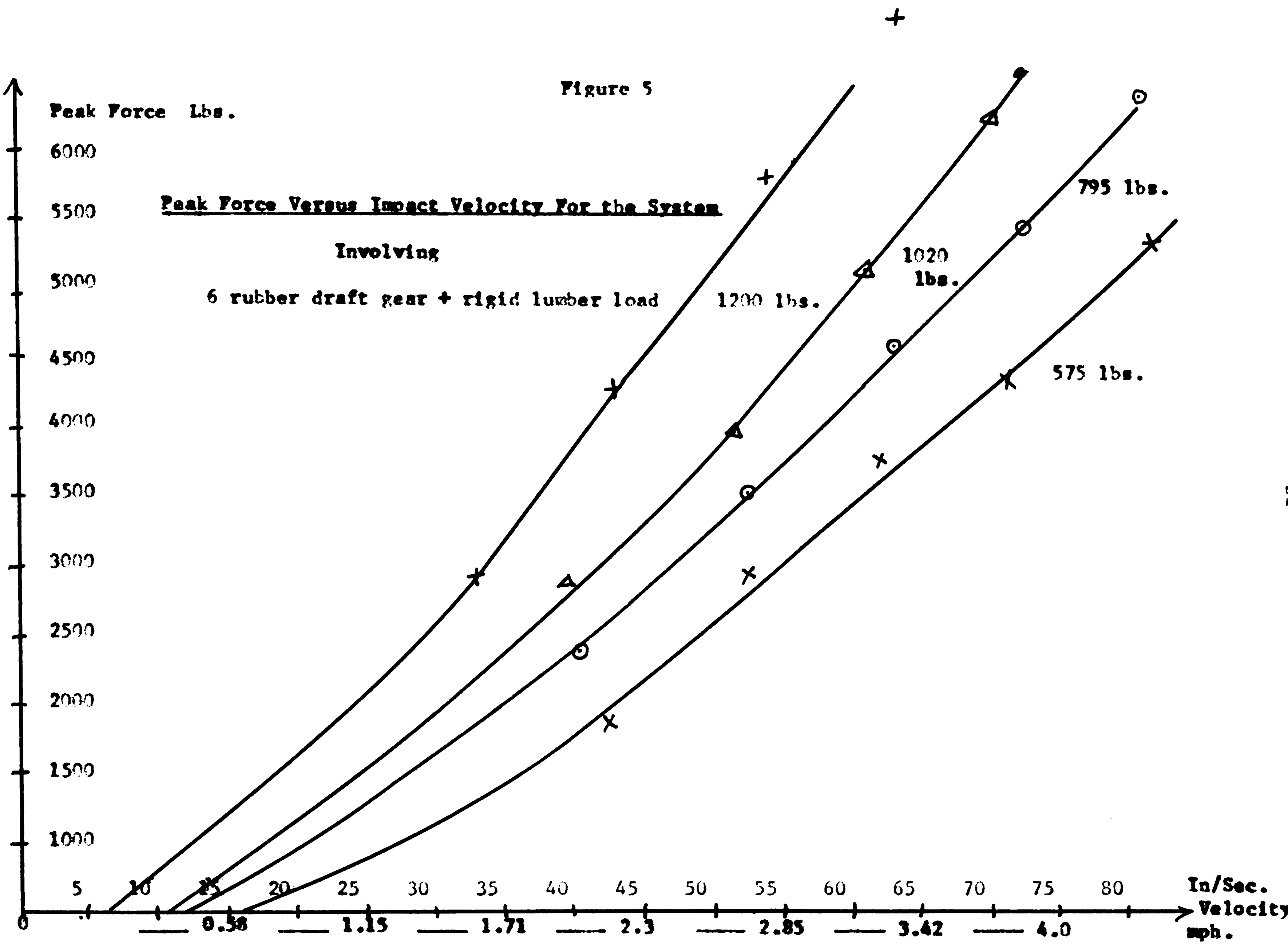
<u>Comparing measured and pulse-derived rebound velocities and maximum deflections</u>

A final comparison of velocities ($\triangle V$), derived from the photographed pulses, with rebound velocities ($V_R = V_i + \triangle V$) revealed some interesting weaknesses of the system. Especially the probable skewed forces in the system due to backstop construction was remarkable. It seems desirable to make the direction of forces and velocity truly horizontal, in order to get better comparable results as far as rebound velocities. (See Appendix V for further details.) This probably plays an important role for the load-damage potential of the system.

Table 6. Characteristic of 6-Rubber-Gear

Position on Incline	Total Load lbs.	Impact Velocity in/sec. (V_i)	Rebound Velocity in/sec. (V_R)	Measured Vel. ΔV in/sec.	Peak Force (Fm) lbs. (Faired Height)	Duration of Pulse m sec. (T/2) (10% Limits)
1	575	42.5	21.0	65.5	1850	120
2	"	52.5	27.5	86.5	2900	100
3	"	62.0	35.0	105.5	3700	90
4	"	71.5	39.5	119.5	4300	90
5	"	81.5	43.5	133.5	5300	85
1	795	40.5	23.0	63.5	2350	110
2	"	52.5	30.0	82.5	3500	100
3	"	63.0	36.0	99.0	4550	100
4	"	72.5	42.5	115.0	5400	95
5	"	80.5	(48.5)	(129.0)	6350	90
1	1020	39.5	23.0	62.5	2850	120
2	"	51.5	31.5	83.0	3950	110
3	"	61.0	37.0	98.0	5100	110
4	"	70.0	44.0	114.0	6200	105
1	1200	32.9	20.3	53.2	2900	125
2	"	43.2	27.4	70.6	4250	120
3	"	54.0	35.8	89.8	5750	115
4	"	63.5	40.5	104.0	6800	110

Figure 5

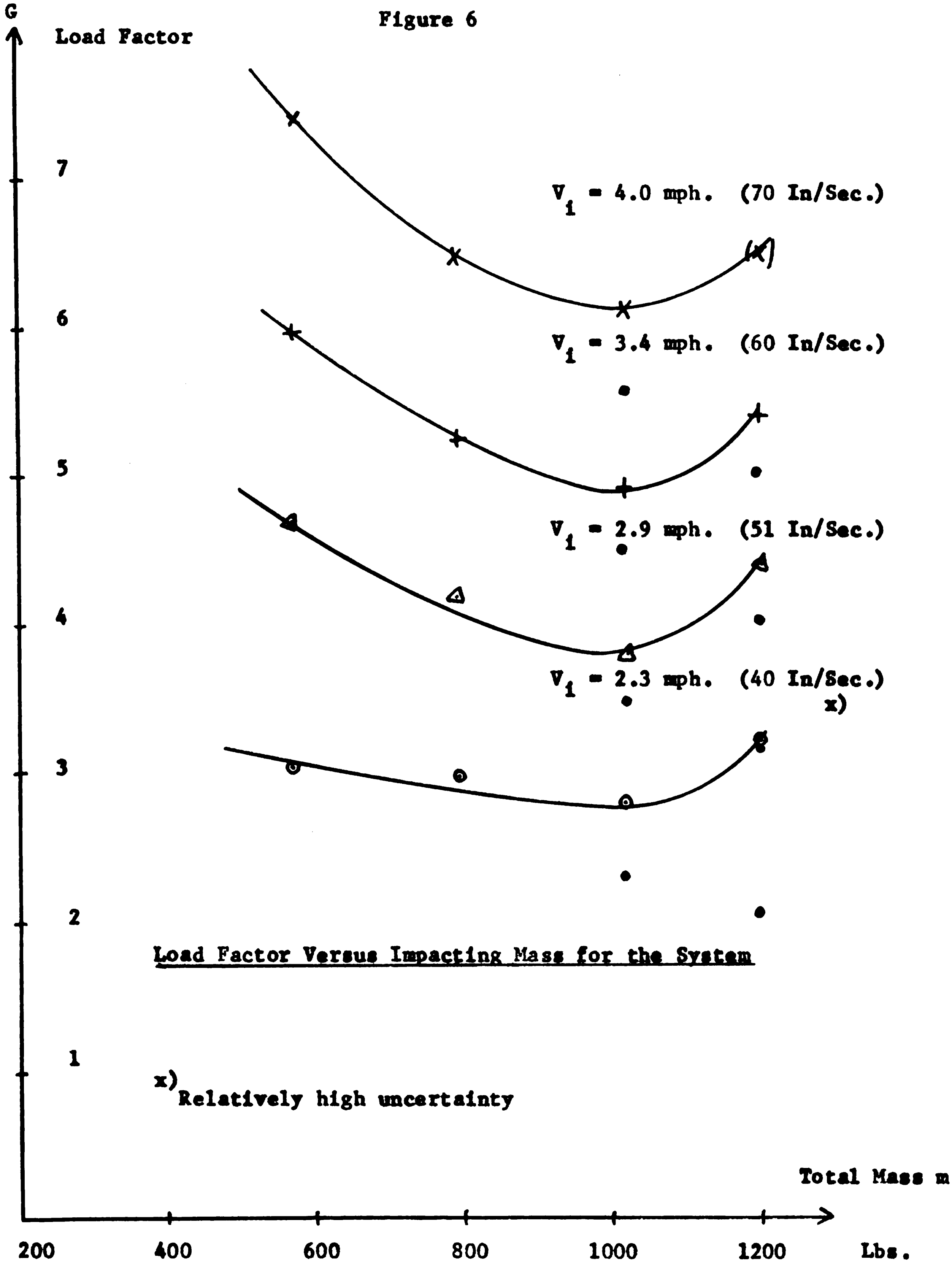


Peak Force Versus Impact Velocity For the System

Involving

6 rubber draft gear + rigid lumber load

Figure 6



<u>**Load Factor Versus Impacting Mass for the System**</u>

x) Relatively high uncertainty

PART B

TRANSIENT PULSES TO THE LOAD AND THE EFFECT OF PNEUMATIC DUNNAGE

Introduction

The following measurements were taken:

(1) Transient pulses in the load of wooden blocks, horizontal direction.

Position 4. Load: 450–490 Lbs.

(a) in front; lower and middle row

(b) in the middle; middle row

(c) in the back end; lower row.

(2) Check on forces in vertical direction.

In the earlier report (1) the peak forces all the way along and upside down the car were mapped. Rather than repeat such measurements of limited interest since they are only valid for the wooden block type of load, we went a step further to measure:

(3) Transient pulses in the load of wooden blocks, horizontal direction.

All positions and all loads used in Part A were also used in Part B. Only in front; lower, middle or upper row.

(4) Transient pulses, when 1, 2 or 3 pneumatic type cushions with different air pressures in them were placed vertically between wooden blocks along the load.

(5) General check on the validity of measurements:

(a) inter-changing the wooden block load cells

(b) turning wooden block cells 180° around.

No change of amplifiers was made. The load cell marked (2) was always connected to amplifier PN 125 and the load cell marked (4) was connected to amplifier PN 153 X.

Summary of findings

(1) There are slightly higher forces in row 1 (near the car floor level) than above in rows 2 and 3.

(2) No vertical forces of any significance compared to the horizontal could be observed.

(3) The forces decreased rapidly along the load from the front to the back end;

(4) In the back end, two impacts of the same magnitude size-order occurred; first, the rear bulk head reaction, and after 100 msec the main pulse appeared.

(5) The magnitude of the transient pulse followed monotonously the magnitude of the impact pulse. In the range of impact velocities used, 7-15% of the impact force in the front end was input force to the lading in the particular row.

(6) The pictures showing two or more peaks, observed at certain loads, were due to load cell construction rather than real input characteristics. The blurred peak picture makes the uncertainty in determining peak force close to the maximum allowable tolerance ± 15% (25).

(7) The air-cushions diminished the input dynamic forces in front with 50-65%, and extended the duration; dependent upon number and position of cushions, but almost independent of air-pressure in the range 1.5 - 4.5 psi.

(8) The feedback on impact force is negligible.

(9) In addition to dynamic forces, static pressure occurs. Generally by increasing static pressure, severe vibrations are superimposed on the main pulse.

(10) These vibrations may be characterized as random, though 60-80 and 800-1000 cps. dominated quite often. Their duration is 20-50 msec. The magnitude went up to a maximum size order that was the same as the peak force of the main pulse.

(11) The vibrations were induced by internal friction in the load of untreated wooden blocks - and normally supported by a medium high level of static pressure. As soon as the friction was brought under control by proper positioning and fastening, in addition to smooth sliding surfaces, the superimposed vibrations diminished.

(12) Coupling between load components and "feedback" of the characteristic pulses through the load was supported, when two or more cushions spaced about 50 in. were used (floating load). When only one cushion is used coupling only occurs at the highest pressure (5 psi).

The theory of air cushions

Figure 4 shows three sizes of pneumatic dunnage.

Many cushioning materials such as foam materials have their effect because of air entrapped in them (pneumatic type cushioning). However, their effect is not independent of the plastic and damping properties, etc. of the material that encloses the air.

One air-type cushioning material, classified for use in normal light weight applications, is "Air Cap."[25)] It consists of a transparent, air-tight, double sheet which encloses air-bubbles approximately 3/8 x 3/8 in., and fulfills the specification MIL-C-81013 for aircraft use, etc.

The next smallest "pure" aircushion seemed to be PNEUPACK-Luftkissen,[26)] which is used for the packaging of electric and electronic parts in Germany. It consists of high-frequency sealed PVC pillows with an air pressure of about 2.8 psi, when unloaded. It comes in 10 x 10 cm. and 24 x 24 cm. pillows with a height of 4-6 cm., respectively 8-12 cm. Maximum allowable static load is indicated to 1.4 psi (10 kg pr. 10 x 10 cm.), by which load the deflection is approximately 2 cm. The material itself contributes considerably to the damping property of the cushion (the enclosed air is principally an undamped spring).

Pneumatic dunnage

Dividers between sections of the load, mainly the wooden dunnage type and pneumatic dunnage, are used in transportation by rail, and eventually by truck. They are not used inside the packages.

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25) Sealed Air Corporation, 179 Goffle Road, Hawthorne, New Jersey.

26) Erich Gericke Plastic-Verarbeitung, Hedemannstrasse 11, 1 Berlin 61, Germany.

Most of the time these devices are used in the doorway part of the car for the double purpose of spacing the load, filling voids between load-sections, and providing a "floating load" type of protection with complete recovery (26). It is indicated too, that air-cushions, besides decreasing the acceleration response in longitudinal direction, will damp vibrations in a vertical direction and lower the frequencies transmitted (27). Standard sizes are 36 x 60 in., 48 x 48 in. and larger.[27)] Manufacturers indicate placement of the cushions not only vertically in different numbers along the car, but also horizontally at the floor or between layers.

Most commercial types of pneumatic dunnage are made from rubber.[28)] One disposable type of bag is made from paper-polyethylene:[29)] An inner bag made of high-density polyethylene, approximately 40 mil. thick and a sturdy outer bag; six layers of 100 lb. kraft paper.

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27) Interlake Steel Corporation, ACME STEEL Products, 135th Street & Perry Ave., Chicago, Illinois.

28)
- (1) ACME Steel: Nylon reinforced, textured Neoprene.
- (2) B. F. Goodrich Corporation, 3135 Euclid Ave., Cleveland, Ohio (distributed through Rubber Fabricators, Inc., Grantsville, West Virginia): Rubber.
- (3) RFD Comp. Ltd., Godalming, Surrey, England: Rubber.
- (4) Firestone (5) Goodyear (6) U. S. Rubber, Plastic Product Division (Reference: Modern Materials Handling, June 1966, 1966 Directory, p. 286).
- (7) TRELLEBORG's Gummifabriks Aktiebolag, Trelleborg, Schweden: Butyl-rubber inner-bag and Neoprene-polyamide outer texture.

29) Manufactured by International Paper Company, 220 East 42 Street New York, New York, Sale through Interlake Steel Corporation.

Static pressure

A test was conducted for the paper bag: (the company indicates that the bag is able to withstand a pressure of 30 psi, but does not say for how long a time). All tests at constant temperature and humidity: 70°F and 50% RH.

19/7-66	11 psi
21/7-66	OK
27/7-66	OK, inflated to 15 psi
1/8-66	exploded

Type of break:

It seemed remarkable that the bag could take the pressure of 15 psi for four days before exploding. The break-starting point was the middle of the front side in the long direction, from which it goes to the end seams, where it breaks along the steel tubes (photograph, Figure 4).

Reasoning:

The strength of paper probably determined the moment of explosion (non-stretchable paper was used). The polyethylene, stretchable to about 200%, can be made slightly smaller than the paper, (if this is not already the case) in order to prevent too much pressure on the paper.

The rubber manufacturers indicate that the bag should be able to withstand 10 psi, under a load of about 8 psi. (Trelleborg) and in free condition to not more than 6 psi (Trelleborg). These data seemed more realistic than the data for paper bags, according to the test above. It was remarkable, too, that the Trelleborg type cushion was reinforced with a steel tube woven into the rubber at the middle in the long direction of the front side, exactly where the explosion starts in the paper bag.

Rules of application

The American Association of Railroads (28), military authorities (29) (30), as well as neutral literature (26), give rules and suggestions for the application of inflated air cushions which are summarized in the following:

(1) Used mostly for rigid types of loads, bricks, blocks, etc., the pressure is 2-8 psi. For fragile items such as glass, china, and fruits, use very low pressure and be especially careful in loading the car (compressibility). For semirigid goods like cans in fiberboard, determine the pressure after "a rule of thumb"; weight of load behind the cushion divided by the effective area between load and cushion.[30)]

(2) The main objective is to fill voids by using one bag for each 2-12 in. of void with the center of the top of the bag slightly above the top of the load; and the bottom of the bag placed about one inch above the floor. An even surface must be provided against the load (effective area).

(3) The dunnage must be filled after being placed between loads. For filling, it is, therefore, necessary with transportable air-compression equipment, pressure gage, etc.

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30) The rule is used by AAR (37). It may be criticized because the G-factor normally is 2-3 g, so that the pressure and compression will be large. On the other hand, the slow response in moderate pressure air cushions accounts for a low amplification, so that the approach might be acceptable in most cases - also, it is very practical.

Calculations for air springs (31)

Air springs have low natural frequency combined with zero statical permanent deflection. When combined with masses, the statements are modified to some extent.

The behavior can be determined by looking upon the dunnage as a cylinder having area A and length dimension x.

$$PV^{\gamma} = P_o V_o^{\gamma} \quad \text{(gas law for adiabatic change)}$$

where P, V, P_o, V_o indicates pressure and volume after and before deflection, and γ is the ratio of specific heat C_p / C_v. [31)]

$$dP / dx = -\gamma P_o V_o^{\gamma} V^{-(\gamma+1)} \cdot dV / dx \text{, where x is deflection.}$$

$$dV / dx = - A \quad ; \quad V = V_o - A \cdot x$$

Stiffness k is the force per unit displacement:

$k = A \cdot dP / dx = \gamma P_o A^2 / V_o \cdot (1 - Ax / V_o)^{-(\gamma+1)}$, which implies cubic elasticity.

Simplified, when the deflection is small:

$$\underline{k} = \gamma P_o A^2 / V_o = \underline{\gamma WA / V_o}$$

Natural angular frequency and period is:

$$\omega = \sqrt{k / m} = \sqrt{\gamma Ag / V_o}$$

$$T_o / 2 = \pi\sqrt{m / k} = \pi\sqrt{V_o / \gamma Ag}$$

No damping will occur for an air spring since air is acting as an ideal gas. By using a big volume V_o (by means of a connecting tank with a valve release for a certain value of pressure), it is possible to obtain a soft spring with an almost constant k-factor. A throttle valve in the air-line may provide for an additional damping to the system.

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31) γ for different gases: H_2 : 1.41   A : 1.67   He : 1.67 CO_2 : 1.31
Atmospheric air: 1.4

Experimental work and discussion

According to the rules of application, a trial cushion may not extend beyond the top of the load.[32] Therefore, cushions 18 x 30 in. were used, which gave an effective area against the load of about $A_o = 200$ in.^{2} (area of 2 rows wooden blocks is $2 \times 5.5 \times 30 = 330$ in.^{2}; but the shape of the pillow will prevent it from contact at the whole area). TRELLEBORG was the only company responding to the request for cushions in that special size.

The simple formula for $T_o / 2$ has the consequence, that duration is independent of pressure. The formula does not hold. The area becomes greater during the impact, though with a lesser increase at higher pressure (where deflection is less). Therefore by higher pressure, average A is relatively small; and duration should be relatively high. The experiments showed the opposite (Tables 9 and 10).

Now we consider the more complicated expression for k which accounts for the increase in k, when travel is not negligible; in fact, travel in this case is about 4-2 in. at 2-5 psi.

With a thickness (void) of 8.5 in., the pillow is approximately a cylinder, which volume is $V_o = 3300$ in.^{3}

$(\pi \cdot \text{diameter} = 2 \cdot \text{width gives } \pi \cdot d = 2.18;$

$d = 12;\ A^1 = \pi / 4 \cdot d^2 = 110;\ V_o = 30 \cdot 110).$

The factor:

$$(1 - A \cdot x / V_o)^{-(\gamma + 1)} = (1 - 2 \cdot 200 / 3300)^{-2.4} = \underline{1.35} \text{ for } x = 2 \text{ in.}$$

$$\text{and } (1 - 4 \cdot 200 / 3300)^{-2.4} = \underline{1.75} \text{ for } x = 4 \text{ in.}$$

shows an expectedly lower k at higher pressure, where deflection is small.

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32) What happens, if it does, may be seen from Figure 4.

The main factor $\gamma P_o A^2 / V_o$ will contribute to a higher k, (an increase of 70% from 3 to 5 psi), whereas the decrement from the other factor is only about 25%.

Total calculation at 5 psi as follows:

$k_o = 1.4 \cdot 5 \cdot 200^2 / 3300 = 85$ lbs/in.

$k_{max} = 85 \cdot 1.35 = 115$ lbs/in.

$k_{av} = 100$ lbs/in.

$T_o / 2 = \sqrt{m / k_{av}} = \sqrt{360 / 386} \cdot 100 = 330$ msec

"m" is used as the total mass behind the cushion in position 2, and comparison in duration might be to the load cell in position 1; in the back end of the car, the duration will be zero according to the same sort of calculation. Now the coupling effect between 2 + 12 cushions will even out the difference, and we noticed the <u>sum</u> of $T_o / 2$ front and $T_o / 2$ back to be approximately 260 msec (Table 9), only 22% below calculated value. The "qualified" average between front and back duration is $\sqrt{(130^2 + 130^2)} / 2 = 180$ msec; which is only 55% of the calculated value.

Conclusively, our model is far too simple. We cannot neglect:

(1) the reaction from the backstop as a stiffening factor that sometimes overlaps the main pulse transmitted.

(2) the friction forces in the load, which probably would give longer pulses (see Tables 9 and 10, cardboard trials).

(3) the multiple degree of freedom-system that are involved, including not only harmonic forces, but also friction forces that are extremely difficult to treat analytically in a multi-degree of freedom system.

(4) the influence of distributed mass, and last but not least:

(5) our model includes, to some extent, the odd shape of the cushion, although we had to assume constant A. But, the elastic forces from the rubber material, when working together with the air-cushion, is not considered - thus we assume an additional stiffening factor, not from the point of impact since the load is incompressible, but from the places where the cushion is not loaded and the air therefore is able to expand. To completely investigate that effect would require measurements of deflection and forces in different directions of the inflated rubber air-cushion in the load, and the effect depends upon the compressibility of the load. Before such a study of restraint-influence on duration - and mainly on forces - is undertaken, it has to be remembered that in actual use the fraction of the cushion area free from the load to the cushion area under the load is loss than in our simulation. If we want to investigate these effects at the 1/4 scale railroad, we therefore ought to adjust to the right proportion of $\frac{\text{area free}}{\text{area loaded}}$. Beforehand, it is unknown if a restraint will occur, or if the rubber material will correspond to the normally involved dynamic impact forces and durations so that no additional factor of importance will occur.

Tables and results

Tables 7, 8, 9 and 10 are attempts to describe the observed pulses under the indicated conditions. The vibrations could have been described by use of the spectral density concept and probability distributions, especially if the reproduction was refined by using the oscillograph - records instead of just photographs. However, the descriptions - preferably held together with the corresponding pictures in Appendix VI - should support the findings that have already been indicated.

Generally the dynamic input forces are diminished with at least 50%,[33] when air cushions are used in the front end of the car. We noticed that forces do not necessarily decrease along the car. At positions far from the cushions with wooden blocks at both sides (for example, 2 + 12, Position 5), the input force may even be about 60% of the input force without cushions, which is about twice as much as in the front position (2 + 12, Position 1).

Usually, there is no clearly directed influence on the peak force of either pressure or the number of cushions as long as the volume of the air cushion is maintained. It is believed that with other types of loads, and especially when internal friction is brought under control, we might be able to see such a difference.

The self-excited vibrations due to the floating load of non-surface-treated wooden blocks that slide against each other producing friction forces with randomly varying friction coefficients will also

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33) 1000 lbs. (Table 8) used in the denominator, probably ~10% higher, due to load hold-down blocks adding to the weight.

determine the obtainable peak forces; however, the phenomena are not easily described by differential equations. Even slight nonlinearity in the terms of the equation of movement will influence the solution considerably.[34]

The experimental evidence for self-excited vibrations can be seen from the last four trials in Table 10 (with cardboard between rows of wooden blocks) compared to trials without cardboard; otherwise conditions are the same as in Table 9.[35]

Also, when there are no wooden blocks to move, as in 2 + 12, Position 1, there are practically no vibrations. The vibrations from other parts of the car are being damped due to the cushion in Position 2, except for a high static pressure which gives high support for oscillations (2 + 12, Position 1, 5 psi).

Coupling is said to occur when there is a clear connection between the values of peak force in the front and in the back of the car; that is, when the value of corresponding load cells are approaching each other relatively and with significance, when going from one pressure to the next highest.

Peak forces, values in parenthesis, are static pressures exerted by the air cushions. Vibrations are being maintained partly due to high levels of static pressure, because oscillations then will take place around the static level represented by the cushion itself which has a

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34) Jacobsen & Ayre: "Engineering Vibrations," 1958, pp. 274-77.

35) It is a usual thing to place cardboard in the bottom of railway cars before loading (32).

negligible mass. The static pressure has been measured once at 4.5 psi: pillows in position 1 + 7 + 13, load cell in Position 4: 430 lbs.; load cell in Position 10: 390 lbs. Average 410 lbs. corresponding to an effective area of 410 / 4.5 = $\underline{90\ in.^2}$ for each of the two rows.

Durations, values in parenthesis, are rebound durations in sequence indicated without letter. If a letter V is indicated, severe vibrations (up to magnitude of approximately the static pressure) occur; if a letter W is indicated, vibrations occur both before and after the main pulse; if letter i, the vibrations clearly superimpose the main pulse; if the letter V does not occur in parenthesis, the pulse cannot be distinguished from vibrations; if letter (w) or (v), inside another parenthesis, only medium vibrations occur.

Table 7. Input to Lading, Front End of Car

Position	Total Load Lbs.	Peak Force F_e (Faired Value) Lbs. x) xx)	Duration t_o, msec	F_e/F_m	Picture Reference
1	795	180	100	0.03	-
2	-	330	90	0.11	78
3	-	620	95	0.14	79
4	-	920	95	0.17	80
5	-	1250 (+90 st.)	100	0.20	81
1	1020	350	140	0.12	82
2	-	480	130	0.12	83
3	-	620 (560)	120	0.12 (0.11)	84
4	-	820 (700)	115	0.13 (0.11)	85
1	1200	70	160	0.02	86
2	-	330	140	0.08	87
3	-	600	120	0.10	88
4	-	620	110	0.09	89

x) (+st.) indicates a static load left over from the preceding impact or induced from pneumatic dunnage.

xx) Numbers in parenthesis indicate the lower peak, when two peaks of about 20 msec in between occur. Otherwise faired.

Table 8. Forces Along and Upside-Down the Car

Position 4 1050 Lbs. Total Load	960 Lbs. 115 msec	-	190 Lbs. 85 msec
(Pictures No. 74, 75, 76)	1030 Lbs. 110 msec	480 Lbs.	(300 Lbs. 15 msec) (reaction from bulk head)

Comments to Tables 7 and 8

The percentage of transmitted load to the lading is most consistent in the 1020 load area, where it is measured in the middle row. In the lower row, it varies from 8 - 20%, higher by higher severety of the impact; this is consistent with some indications in part A of higher amplification factors by higher load factors: The lower row is more close to the floor and the impact force. The 12% corresponds fairly close to the values in the bag-damage-simulation tests (7).

In the upper row, the transmission factor is a little lower than in the middle row, and from Position 1 of the incline the response is very small.

The decrease of the magnitude of the main pulse along the car from 1000 lbs. to 200 lbs. and the time delay of the main pulse, about 100 m S, is a characteristic of the particular lading.

Because of the relatively small forces in the back end of the car and the additional blurred pulse, that the rear bulk-head introduces, it seems advisable always to use the front end of the car for damage simulations.

Table 9. Pulse-Characteristic for Pneumatic Dunnage - Void 8 1/2 Inch Maintained. Position 4 ~1050 Lbs. Total Load.

Cushion Position	Load Cell Position	Pressure psi	Peak Force (Faired Value) Lbs.	Duration (t_o) msec	Coupling + ⊖	Velocity Check xx)	Picture Reference
2+12	1	5	270 (+s450)	130 (+120) (i 100v)	+	1229	53, 54
"	5	5	490 (+s450)	120 (+200v)	⊖	1174	51, 52
"	8	5	180 (+s450)	120 (+200v)	⊖	1174	51, 52
"	13	5	140 (+s450)	130 (+110) (i 50v)	+	1229	53, 54
2+12	1	-	-	-	-	-	-
"	5	4	590 (+s360)	140 (+v)	⊖	1227	55
"	8	4	200 (+s360)	80 (+v)	⊖	1227	55
"	13	-	-	-	-	-	-
2+12	1	3	330 (+s270)	160 (+70)	+	1228	56
"	5	-	-	-	-	-	-
"	8	-	-	-	-	-	-
"	13	3	100 (+s270)	150 (+100v)	+	1228	56
2+12	1	2	320 (+s 180)	150	+	1225	57, 58
"	5	-	-	-	-	-	-
"	8	-	-	-	-	-	-
"	13	2	70 (+s180)	150 (+50v)	+	1225	57, 58
2+12	1	1	420 (+s90)	220	⊖	1160	59
"	13	1	50 (+s90)	-(+30v)	⊖	1160	59

Table 9. Continued.

Cushion Position	Load Cell Position	Pressure psi	Peak Force (Faired Value) Lbs.	Duration (t_o) msec	Coupling + -	Velocity Check xx)	Picture Reference
2+12	1	0	370	220	-	1224	60
"	13	0 xxx)	30	-	-	1224	60
7	1	5	420 (+s450)	120 (+80)	+	1238	61
"	13	5	170 (+s450)	80 x) (iv.)	+	1238	61
7	1	4	490 (+s 360)	120 (+60)	-	1231	62
"	13	4	150 (+s 360)	90 x) (iv.)	-	1231	62
7	1	3	440 (+s270)	160 (+30) (iv.)	-	1226	63
"	13	3	150 (+s270)	70 v	-	1226	63
7	1	2	480 (+s180)	180 (+50) (iv.)	-	1228	64
"	13	2	80 (+s180)	70 v	-	1228	64
7	1	1	580 (+s90)	150	-	1232	65
"	13	1	80 (+s90)	50 v	-	1232	65
1+7+13	4	5	470 (+s450)	120 (+90+60)	+	1150	72
"	10	5	170 (+s450)	130 (+20 v)	+	1150	72

Position

0 1 2 3 4 5 6 7 8 9 10 11 12 13

Key for Position Numbers

x) The first reaction from the backstop not indicated (only a few vibrations and magnitude about the same as for the transmitted main pulse).

xx) Reading at counter in 10^{-4} sec. by constant distance, that is the higher value, the lower velocity.

xxx) Void fill-out is maintained and valve closed.

Table 10. Pulse Characteristics for Pneumatic Dunnage - Variation of Void, Check on Rubber Material, Impact Chock, Paper Bag, and Cardboard Material in Load.

Cushion Position	Load Cell Position	Pressure	Void, in. x)	Peak Force	Duration	Velocity	Picture
7	1	4	7 1/2	470 (+s360)	160 (+200) (i.v.)	1220	66
7	13	4	7 1/2	120 (+s360)	50 (i.v.)	1220	66
7	1	4	11	530 (+s360)	200 (+80)	1236	67
7	13	4	11	100 (+s360)	25 (i.v.)	1236	67
7	1	0	0 (rubber)	1320	120	1158	68
7	13	0	0 (alone)	90	30	1158	68
2+12	1	3	8 1/2	300 (+s270)	160	1181	69
2+12	0 (impact)	3	8 1/2	5200	130	1181	69
7	5	~2	~10 (paper bag	580	170	1231	70
7	8	~2	~10 large)	~0	~0	1231	70
xx) 2+12	5	5	8 1/2	500 (+s450)	110 (+80) (150 (V))	1234	71
xx) 2+12	8	5	8 1/2	170 (+s450)	90 (+ (w))	1234	71
xx) 1+ 7+13	4	4 1/2	8 1/2	370 (+s410)	100(+100+70)	1236	73
xx) 1+ 7+13	10	4 1/2	8 1/2	150 (+s410)	110 (+100..)	1236	73

x) 7 1/2 in. is about the maximum compression possible with a rope tie-down at the particular impact; neither steel-straps nor plastic-straps could hold the load down.

xx) Cardboard, coated type, is placed between rows no. 1 and 2 in addition to the C-flute corrugated in the bottom of the car.

PART C

REMARKS ON SIMULATION AND REPRODUCTION OF DAMAGE

As a matter of fact, very little information was available about the correlation between the input characteristics and the damage on particular goods caused by the input. The bag simulation, mentioned before, is one of the few attempts. (7)

In order to simulate damage in the laboratory, it is necessary to know the damage potential of the input compared to practice. The attempt in the present study has been devoted to

A. Producing well defined impact conditions and pulses that in duration and travel are comparable to practice.

B. Measuring the relationships between the impact and the input to a certain type of load.

Simulation of forces

The shortcomings of the laboratory-type simulation might be first of all:

(1) the lack of force that in size-order would be similar to practice.

(2) the lack of reproducibility of vibrations and friction which are also involved in practical operations.

Now concentrating on the shortcoming (1) and thereby restricting ourselves to discuss matters of importance for goods that are not vibration-sensitive (which we do not know too much about), we have produced input forces at about 2000 lbs. total in the front end of the car. It is 2000 / 330 = 6 psi dynamic force (in fact we have no information regarding how the input force is distributed over the cross-direction of the car, neither in practice nor in the laboratory; at present, we must assume even distribution of forces horizontally across the car). In practice, input forces to the lading (though a lading of bags, probably giving less peak force than wooden blocks) have been measured to 1800 lbs. total force with a cross section area of 6x5' = 4300 in.^{2}, is 1800 / 4300 = .42 psi dynamic force (for bags).

In this respect, too, our simulation seems fairly good, if the causes for damage really are unidirectional forces centrally located and evenly distributed over each container. How is this in practice?

Simulation of damage

We decided to further investigate the canned goods; namely, carbonated juices in cans. According to (33), this type of goods is one of the most susceptible to damage; about 11% of revenues are paid back in claims, whereas the overall average is only 1%. Causes for damage are indicated as follows:

Poor arrangement of the load	25%
Improper handling of the car in transit	23%
Shift in loading due to a loose load	15%

Unfortunately, these firures suggest bad loading as the main source for damage. In other words, they refer to material handling methods, not to packages. However, the figures imply that internal shift in lading (in addition to forces acting on an integral type of load) are causing the damage. Internal shift is difficult to simulate in a laboratory device where we do not have the height, number of layers and packages necessary to produce a random shift in different directions.

In a way, drops of packages in the car seem to be the cause of much of the damage. This brings us to the question. Would drop-testing be sufficient for investigation of damage? In fact, the forces acting in vertical impacts (switching) have the same size-order as forces acting in vertical impacts (drops).[36] The answer might be denying, because the durations are really different (about 100 msec versus $<$ 10 msec), resulting from the long travel of draft gears. Therefore, elements of the lading, that would not be involved by a drop test because amplification factors (due to low value of $\frac{\text{element natural frequency}}{\text{input natural frequency}}$) will be $\ll 1$, might be involved in impact tests with the approximately 10 times as low "input natural frequency."

Also the horizontal shift in lading that produces damage by a second impact against other lading or the ends of the car ought to be studied in a proper device with room for distribution of load, devices for measuring influence of friction, etc., although it might be studied analytically to some extent (34). A drop-testing-device will not suffice for investigations of that kind.

(1) Direct simulation involves the use of the test item (package) in the front end of the car with a rigid and unmovable mass behind it. The mass behind may consist of the same type of items (packages) or a dummy load.

(2) Indirect simulation takes into account the smaller forces due to smaller masses behind in the simulation than in practice, by adjusting the test item (package). If it is found, for example, in a load of bags that the input force is only one half in the simulation in the front of the car

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36) For example, reference (24) (pp. 26-28) uses a Neoprene rubber spring similar to ours, and a mass of 227 lbs. impacting with a velocity of 50 in/sec. gives a peak of 38 g's, it is 8-9000 lbs. (compared to Table 6: 3-5000 lbs.).

as it is in practice (in the position of a full scale car that is under investigation), then the knowledge of package strength as a function of the size of the package, the thickness of bag material, etc., will make it possible to adjust the package to the forces occurring in simulation (for example, to make a bag with twice the dimensions of the package in practice).

Maybe the only simple example describing that is a wooden or plywood box which can be treated according to the beam concept (35); say a wooden board of length L, thickness T, width D:

S (Bending Stress) =

(Bending Moment) $M \cdot T / 2 \cdot I$ (Moment of Inertia) =

$$= \frac{F_m \cdot L \cdot T / 2 \cdot 12}{8\ DT^3} = \frac{F_m \cdot L}{DT^2} \cdot \frac{3}{4}$$

In terms of maximum deflection:

$$Y_{max} = \frac{5}{384} \cdot \frac{F_m \cdot L^2}{E \cdot I} = \frac{F_m\ L^2 \cdot 60}{DT^3 \cdot E \cdot 384}$$

where E is the modulus of elasticity.

In order to perform a simulation we choose to build a model $2 \cdot L$ of the original container. With the other dimensions unchanged, the stress for the test container will be the same by a simulation peak force of F_m, as it is for the original container in transportation with a peak force of $2 \cdot F_m$, etc. The static concept could be modified by the influence of the amplification factors. To check this the natural frequency for the beam:

$$f_n = C \sqrt{E I g / F_m L^3}$$

C = 3.56 for fixed beam may be calculated (25) and compared to the input force frequency.

Friction forces that have been brought under control and fulfill the simple equations (for dry friction: $F_{fr} = \mu \cdot W$, where μ is a constant friction coefficient, and W is the weight component rectangular to the sliding surfaces), may be simulated separately. Say the weight in vertical direction above the car floor is only 1/4 in the simulation than it is in practice (possible height 1/4 of full scale, same type of commodity up and down in the carload), then by choosing four times as high friction coefficients as in practice between the load and floor (choose a fairly rough or "sticky" floor material), the friction force will be the same. It is noticed, that it would be impossible to combine the latter method with the former dimension-adjusting method in one test.

The case of resilient lading

(a) If the lading or parts of it have space for moving in the horizontal direction (free slack), the input velocity to the lading relative to the car is $V = V_i + V_R$. (Law of inertia, friction 0). In practice, since the coupling between cars will stop the striking car immediately after impact, the input velocity is only V_i [37)] or it might be V_R

37) The statement is slightly modified by the fact, that if only a few of free slack is available, that is if the lading will already, during the first half part of the impact move sufficiently to hit other lading or bulk head (V_i / 2 x t = 3.5 in. for V_i = 70 in/sec. and t = 0.1 sec.) then the input velocity is less than V_i even at friction zero, but in that case the damage potential is low too.

for slack in the other end of the car or the unit and is then directed in the opposite direction. It was noticed then, that although impact velocities are the same in the simulation as in practice, then with the same relationship V_R / V_i, (namely $\sim 37 / 61 = 60\%$, Appendix V compared to 53% measured in practice (11)) the impact simulation is more severe than impact in practice for slack in the front end of the car or the unit, whereas slack in the back end has no effect. We may be able to adjust for this by changing to a lower impact velocity at the same time considering forces involved, when possible.[38] But the best solution would be to build the test track to include a coupling device similar to practice.

(b) If the load is to some extent floating or resilient, the input velocity may be something between V_R and $V_i + V_R$, eventually 0, and we would have a very bad simulation, when friction has to be taken into consideration (Part B).

(c) We reached the conclusion that our work is successful only in the case of a rigid load, otherwise coupling, etc. will have to be included on the test track in order to perform a direct or indirect simulation procedure.

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38) Different types of friction forces occur.

Experimental work and calculations

A fiberboard box 27 x 5.5 x 10 in. (L x W x D), B-flute, 200 lbs. (Mullen), double faced, RSC, containing 40 cans, tin-coated, 412 x 210, each containing 12 oz. MAVIS Club Soda-beverages. Soldered, seamed ends, 3/8 in. space above liquid. Total weight: 35 lbs. Taped over the flaps.

The package was exposed to impacts, V_1 = 70 in/sec.

W = 975 lbs. (F_m = 6000 lbs. for wooden block load,

F_e = 1700 lbs. in front position) and higher (table 11).

No damage except for small indentations under the lid occurred (cans turned around occasionally, so that new indentations could be observed), although dynamic forces as high as 1800 / 27.10 = 6 psi were encountered.

Because we were unable to reproduce damage in the box, cans were taken out and exposed to impacts in three directions (table 11). Some authors indicate strength of cans (static) to about 1000 psf = 7 psi (6) and here two cans with a top area of 2 x 5.5 in.^{2} were exposed to a force of about 850 lbs.; 850 / 11 = 77 psi (if a pulse shape similar to the earlier observed pulses are assumed) without a break.

It is quite complicated to consider the strength of the can analytically. It involves both elastic, plastic and viscous properties for a rather complex item. However, the maximum force acting on the end of the filled can may be calculated approximately by assuming that the can is acting as a liquid spring, without damping (adiabatic), where the force rises very fast to a constant value. The law of conservation of energy applied internally in the load then gives:

$$F \triangle S = 1/2\ M\ V_1^2 \text{ , where}$$

F is the constant reaction force from the can,

$\triangle S$ is the corresponding maximum deflection of the can,

(= displacement of the row of wooden blocks relative to the car),

M is the mass of the row of wooden blocks, and

V_i is the total decrease in the velocity of the wooden blocks during the impact, relative to the car.

If the time for total deflection of the can is not too much smaller than the rise time for the input pulse (it can hardly be believed, that it is longer), V_i is the same as the measured impact velocity, and friction forces will hardly be able to brake the movement at a distance of 3/8 in. From the trials (1 can), it seems as if the minimum breaking point is about $V_i = 70$ in/sec., and then we get

$F \cdot 3/16 = 1/2 \cdot 205 / 386 \cdot 70^2$; $F = 7000$ lbs.; $P = 700 / 6 =$ __117 psi.__

The friction force involved and assumed constant means that $F = F_{can} + F_{fr}$; with cardboard underneath, the friction coefficient is less than 1; $F_{fr} < 200$ lbs.; only slight difference in the calculation will therefore occur.

Impact from the side gives 1/8 in. deflection at 70 in/sec., that means an area of 9 in.^{2} for the two cans in the trial gives the (constant) force $F \cdot 1/8 = 1/2\ 205 / 386 \cdot 70^2$; $F = 10{,}400$ lbs.; $P =$ __115 psi__.

The present example is only meant as a rough calculated case demonstrating that the present test track can be used to produce damage and data relating to it, although it requires a lot of force even to break cans on the test track. The observation supports that other causes for can-breakage during railroad transportation are more important than unidirectional forces.

Table 11. Impacts on Lading Including Cans.

Load Lbs.	Position of Box/Can Cushion	Number of Impacts	Remarks
975	7 (box)	1	-
	13 (box)	2	-
	1 (box)	3	Beginning indentations under the lid where can is touching another can.
1175	1 (box)	2	Beginning indentations under the lid where can is touching another can.
1050	4 (box)	5	-
5 1/2 psi	1,13 (cushion)		
cushion x)	1 (box)	5	-
	2,13 (cushion)		
1250	3+4 (box) xx)	5	-
5 1/2 psi	1,13 (cushion)		
1020	1, from end	4 (10 cans)	No damage up to 70 in/sec.
	1, from end	4 (2 cans)	No damage up to 70 in/sec.
	1, from end	4 (1 can)	At 70 in/sec. crushed 3/16 in. and leaking a little.
1020	1, from side	3 (2 cans)	At 70 in/sec. 1/8 in compressed in first blow. (2) a little more (3) no further compression.

Table 11. Continued.

Load Lbs.	Position of Box/Can Cushion	Number of Impacts	Remarks
780	1 (40 in/sec)	1 (1 can)	slight impression, 1/16 in.
(One layer of wooden blocks resting on cardboard)	1 (52 in/sec)	1 (1 can)	1/8 in. impression (measured across).
	1 (63 in/sec)	1 (1 can)	7/8 in. impression, leak.
	1 (70 in/sec) diagonally xxx)	1 (1 can)	Total break and leak

x) Static pressure exerted for 1 hour before dynamic test. Effective area: 96/270 gives 2 psi static pressure on box.

xx) Box placed horizontally in the middle row.

xxx) Special jig secured the can in 45^{o} position (Figure 8).

Figure 8

Cans after exposure to impact. Shown from left to right: impact from the side, diagonally (3 stages) and from the end.

Jig used for diagonal testing of cans shown in place in the front end of the car.

6. SUMMARY AND CONCLUSION. SUGGESTIONS

With the objective of predicting damage potential caused by humping and switching operations performed in practice on railway freight cars, a laboratory scale railway test track with car and lading was investigated as the model system.

Being interested in an output (readings concerned with damage) from the model similar to, or with a simple calculatory relationship to the real system during operation, we tried to compare the components of the input to the two systems.

Impact conditions, input-forces to the lading and self-excited movements in the lading all constitute factors, that must be assumed to be important, with a lack of comparable information on the detailed behavior of the lading as a function of the switching operations alone, our attempt was to discuss and make suggestions in order to adjust the model from an environmental analysis-standpoint, involving parameters such as

- impact velocity
- magnitude, shape and duration of impact forces
- impact quantum (equivalent mass of striked cars)
- rebound velocity
- movement during and after impact
- magnitude and duration of input forces.

Having reached the input to the lading quite successfully, we proposed direct and indirect simulation procedures using items or model items from which it should be possible to describe damage. Correlation to practice is practically unknown, and parameters such as

- superimposed vibrations on impact
- self-excited random vibrations in the load caused by internal sliding, etc.

seemed impossible to simulate in the laboratory. But we still suggest that proposed procedures - after proper adaptation of the test track - such as:

- securing strictly horizontal impact and movement of all outer components
- rebuilding backstop to perform as 3 or 4 striked cars in actual operations, preferably with no rebound travel
- reinforce construction and hauling devices to operate at least masses of 1200 lbs. securely

will make a valid prediction of damage possible for a wide range of commodities being transported by the railroad.

Although it might be outside the direct scope of the present work, it is suggested that trials including the analysis of damage potential be made on full scale test tracks also, because this is believed to be the only way to steadily adjust a laboratory scale device to account for special phenomena and new developments in railroad transportation (7).

Air-cushions included to divide lading into sections were investigated and showed up to have a decreasing effect on the magnitude of input forces to the lading. Further investigations are suggested including other types of loads than wooden blocks, preferably with a controlled type of internal friction, using unitized loads and smoother sliding surfaces. Also the effect of unitizing, bracing and fastening means (36) (37) such as steel accessories, non-skid plates, anchor plates and retarder plates in connection with steel strapping and wooden type dividers can be investigated and compared to air-cushions.

An attempt to reproduce the damage of canned goods in fiberboard boxes were not successful, but damage was obtained by exposing one or two cans to the maximum obtainable input-forces.

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APPENDIX I

Impact Velocity Versus Position on Incline, Theoretical and Measured; Uncertainties Involved

1. Theoretical impact velocity:

$$V_{it} = \sqrt{2g \sum W_n \, h_n / W}$$

derived from the law of conservation of energy:

$$1/2\ M\ V_{it}^{\ 2} = 1/2\ W/g\ V_{it}^{\ 2} = h_1 W_1 + h_2 W_2 + h_3 W_3 + h_4 W_4 + h_5 W_5$$

W_1 = Weight of load; h= corresponding height - height at the floor level (of center of gravity).

W_2 = Weight of rear bulk head, 135 lbs., h_2 as above.

W_3 = Weight of front bulk head, 135 lbs., h_3 as above.

W_4 = Weight of car floor with steel-angles, 150 lbs.

W_5 = Weight of wheels and suspensions, 80 lbs.

The weights are figured out from the information given in (1). The heights of center of gravity were measured for each of the components.

<u>Sample Calculation</u>

Position 2, 445 lb. load:

$$V_{it} = 27.8 \sqrt{(5.0 \cdot 445 + 12.0 \cdot 135 - 2.0 \cdot 135 + 5.0 \cdot 150 + 11.5 \cdot 80)/1020}$$

(Only backwheels undergo a decrease in height.)

Table 1. Impact Velocities

Position	Theoretical Velocity by Load: in/sec				Measured Velocity by Load: in/sec			
	575 lbs.	795 lbs.	1020 lbs.	1200 lbs.	575 lbs.	795 lbs.	1020 lbs.	1200 lbs.
1					42,5	40,5	39,5	32,9
2	65,2	63,8	62,3	63,1	52,5	52,5	51,5	43,2
3					62,0	63,0	61,0	54,0
4	83,2	82,5	-	83,3	71,5	72,5	70,0	63,5
5					81,5	80,5	-	-

<u>Comments to Table 1:</u> <u>Dry Friction</u>

The truly obtained velocity is less than the available, except when no friction occurred. The available velocity has been calculated for Positions 2 and 4 (Table 1).

According to the equations for dry friction:

$$W/g \cdot \text{acc.} = W \sin x - \mu W \cos x$$

Where x is the angle of the car with the horizontal, we get:

$$V_F = (g \sin x - g\mu \cos x)\, t + V_{oF}$$

μ is the friction coefficient that we want to determine. When integrating we assume that cos x is constant, in fact it varies with the time; for Position 2 from 0.985 to 1.0, for Position 4 from 0.96 to 1.0. If there is no friction, we get:

$V = g(\sin x)t + V_o$ where V is the friction-free velocity.

$V_o = V_{oF} = 0$ (initial velocity), and then

$$V - V_F = \mu g \ (\cos x) \ t$$

t was measured to ca. 3.0 sec. (by a stopwatch) for position 2 (and 4), which gives

$$\mu = (64.0 - 52.0) \ / \ 386 \cdot 0.995 \cdot 3 = 0.0104$$

The corresponding friction coefficient μ' between bearings and axles is determined by

$$\mu = r \sin \phi \ / \ R \cos x$$

where r is the radius of journals, R is the radius of the wheels, and ϕ is the angle of friction between bearings and axles:

$$\sin \phi = 0.0104 \cdot 0.995 \cdot 8 \ / \ 0.8 = 0.104$$

$$\mu' = \operatorname{tg} \phi = 0.105$$

which is quite usual for a railway truck. Therefore, this point in simulation is quite well justified.

2. Velocity was measured as follows: Just before the car hits the backstop, the time interval t between the front wheel hits the first and the second switch was measured at a Hewlett Packard electric counter. The distance d between the switches was determined carefully by rolling the car very slowly over them, and the following formula used:

$$V_i = d/t \ (\text{in/sec}) = d \cdot 60 \cdot 60 \ / \ t \cdot 12 \cdot 5278 \ (\text{mph})$$

Because of the small friction and the small distance (about 8-10") just before impact, we get a high degree of accuracy.

As a result of approximately 70 measurements, we found a double standard deviation (2σ) of 1.2-1.8% for velocities less than 60 in/sec.

and 2.5 - 3% for velocities 65-30 in/sec. with a sample size of 2-3, for longer series from one position as in Position 4, though less than 1.5%

As a rule, the velocity was checked each time in Parts A and B of the experiments. This is considered to be a very important parameter that goes into energy calculations as $(V_i)^2$, and even more emphasis could be placed on a higher degree of accuracy.

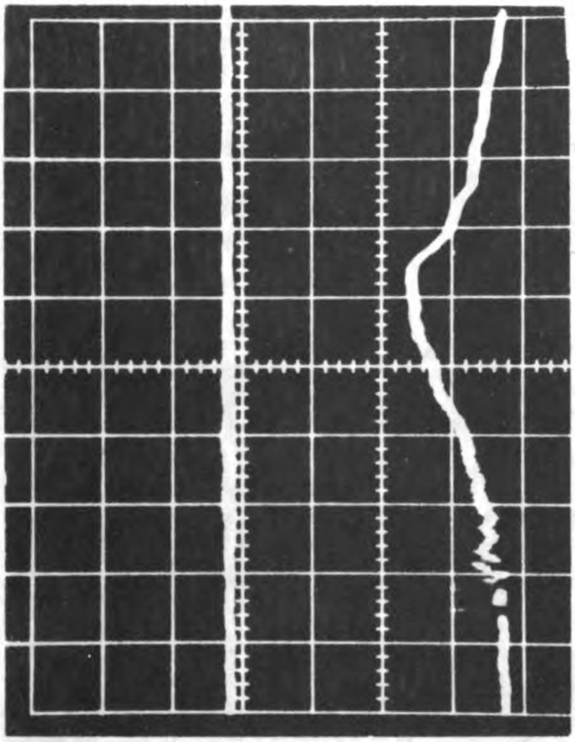
With or without load up to and including 1020 lbs., the impact velocities are fairly constant with the same height and are consistent with the conservation of energy-concept discussed under calculation of theoretical velocity. There is no influence of distributed mass (two or higher lumped) before the impact.

The significant jump in impact velocity at 1200 lbs. for same height is believed to be the result of the overloading of the suspension and supporting system of the car, thereby increasing friction. In other words, we are close to the maximum capacity of the car, and normally do not want to use as high a load for that car.

Another observation made by high load and high position at the incline revealed that the back end of the car almost lifts itself from the rails. The reasons for this jump that the suspension system is hardly able to prevent, is the rotational moment, that develops just after the impact because of the different level of the attack point of the impact force and the center of gravity of the car.

APPENDIX II.

Selected photographic series for draft gear pulses, Tables 2 and 3.



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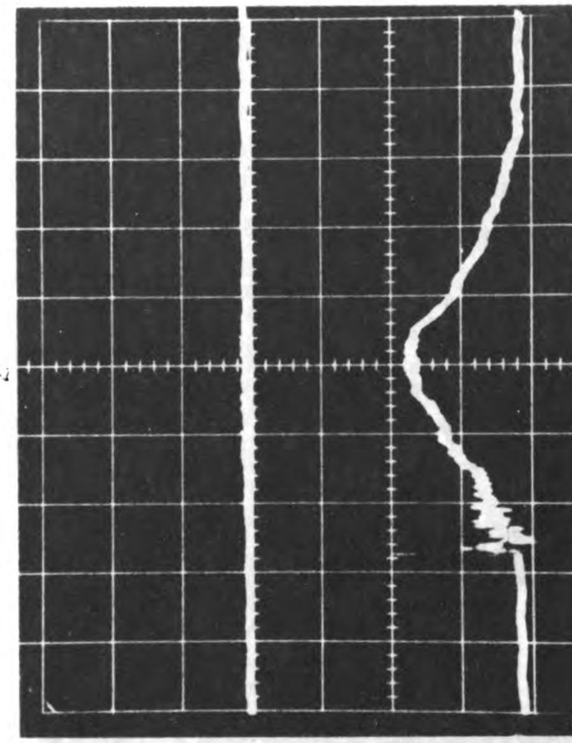

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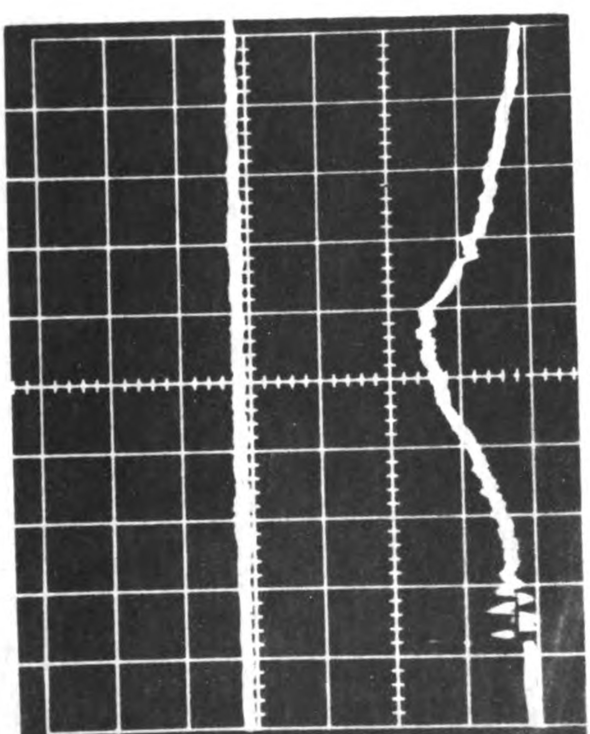

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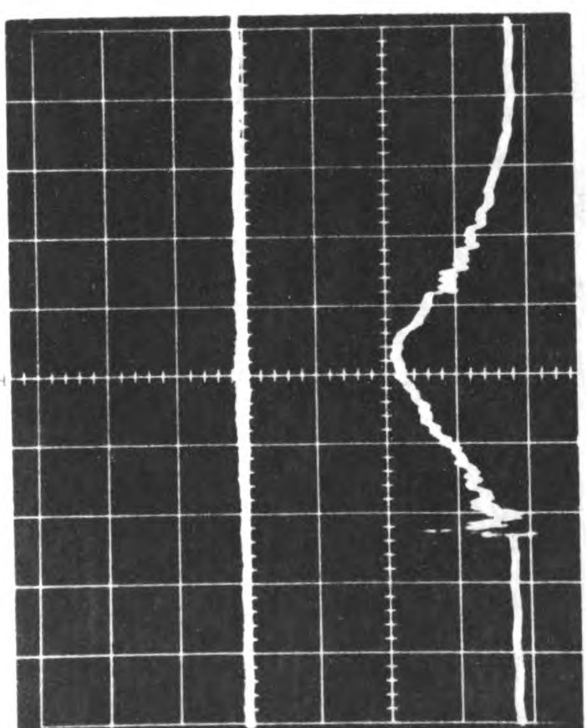

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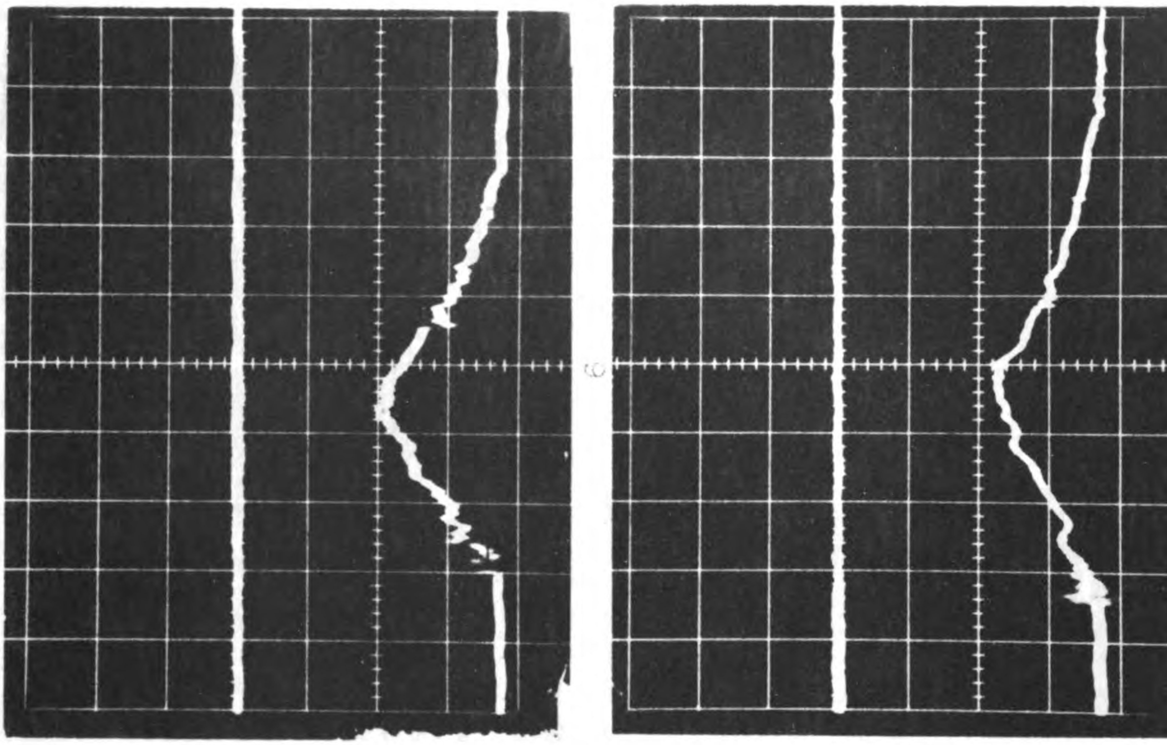

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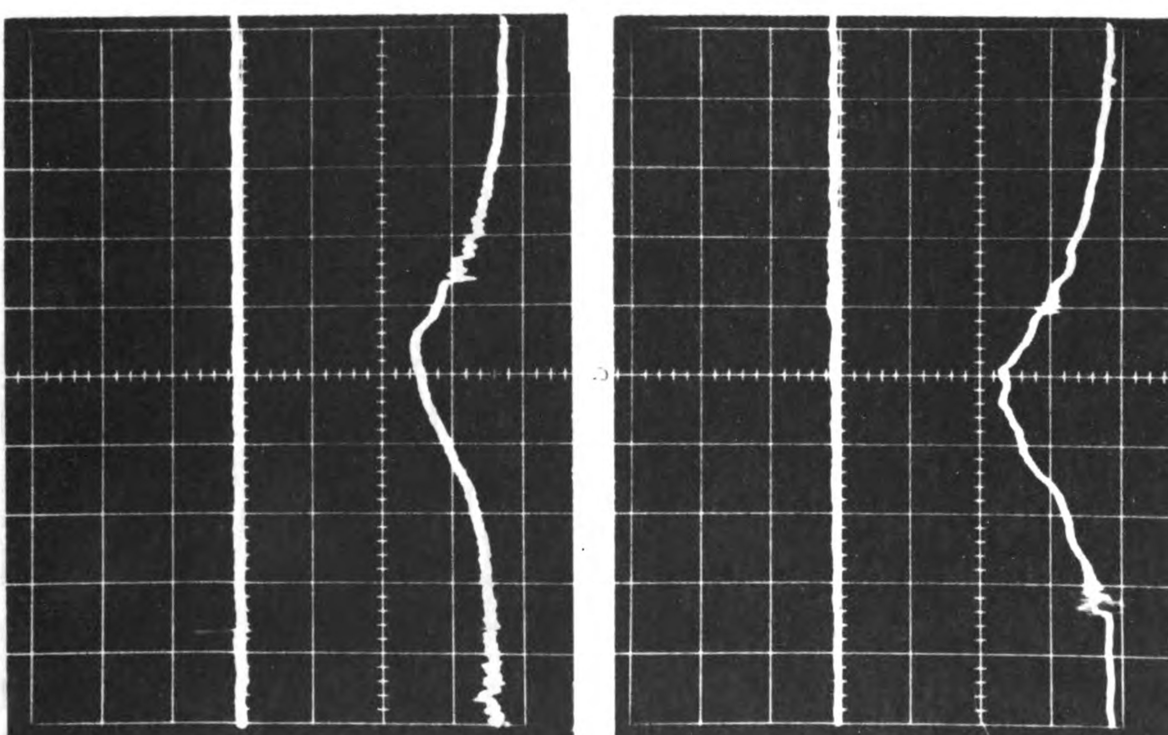

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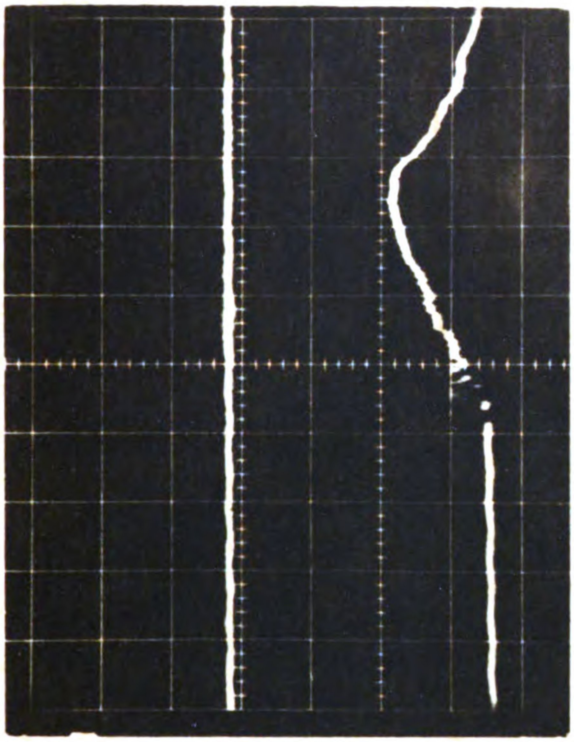

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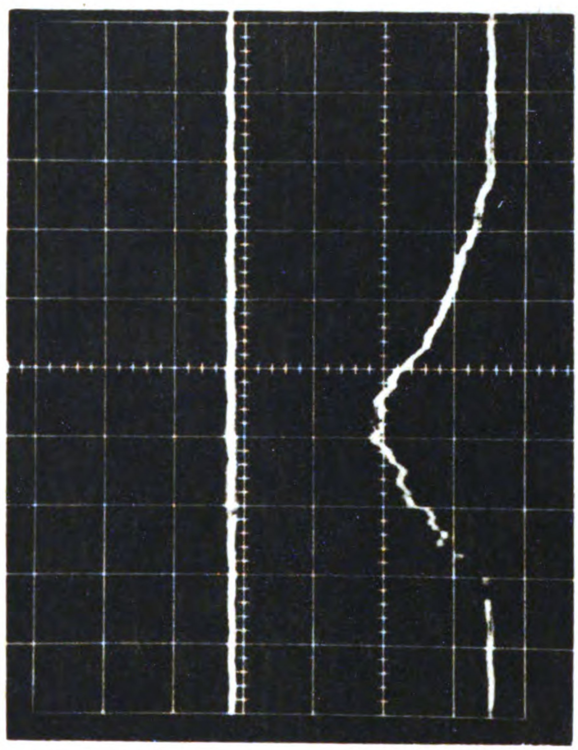

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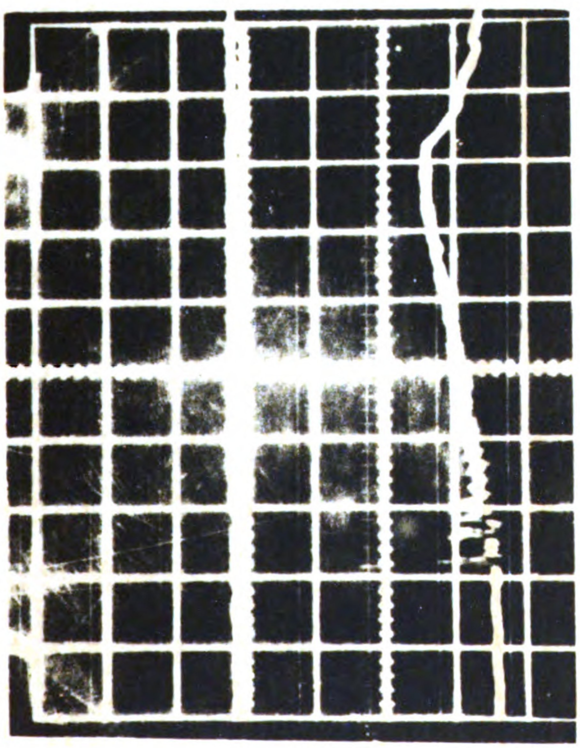

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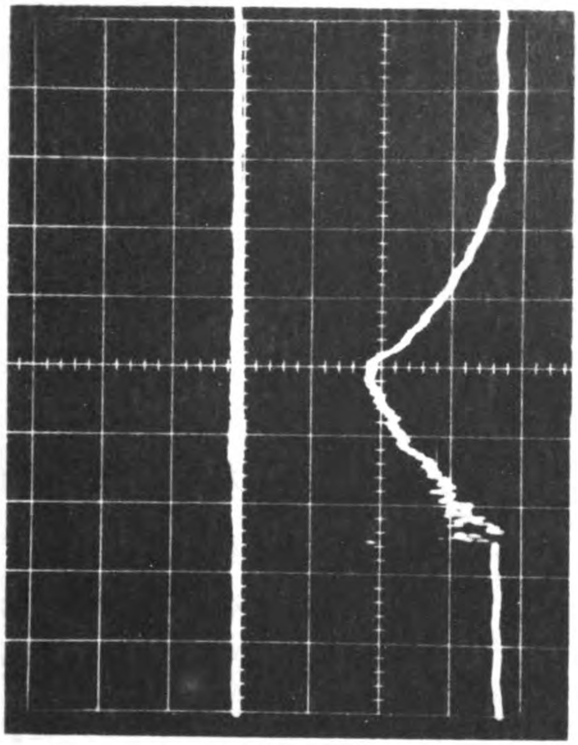

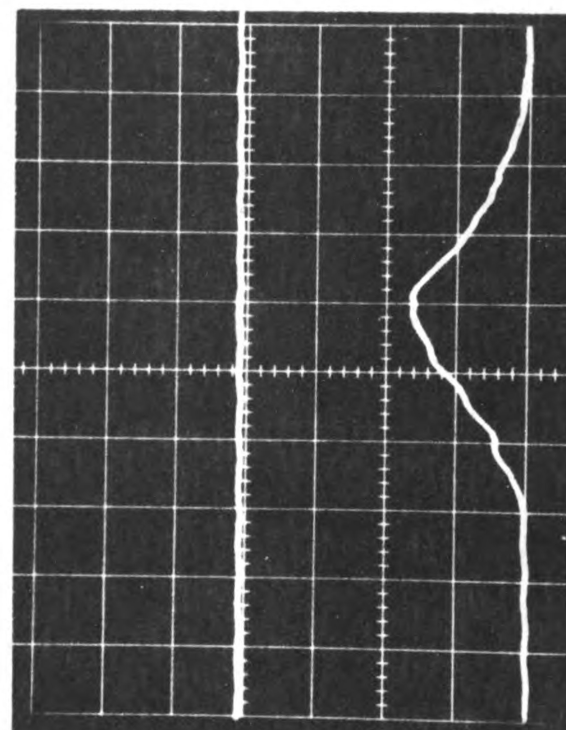

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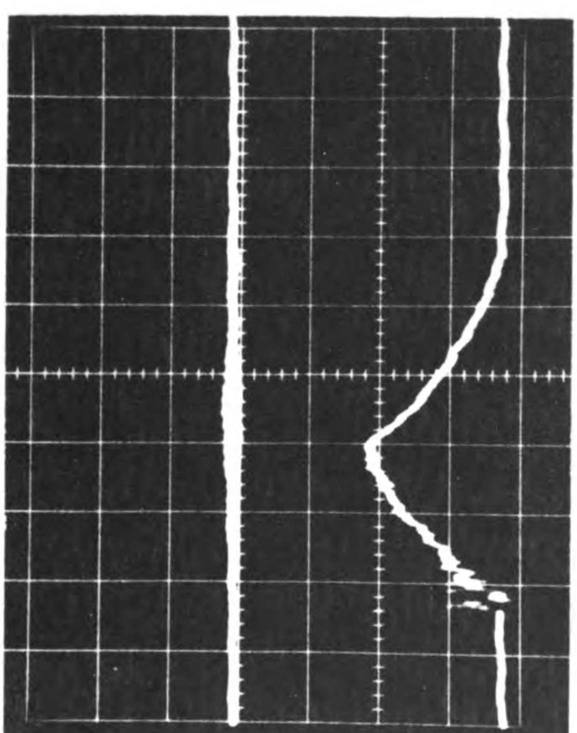

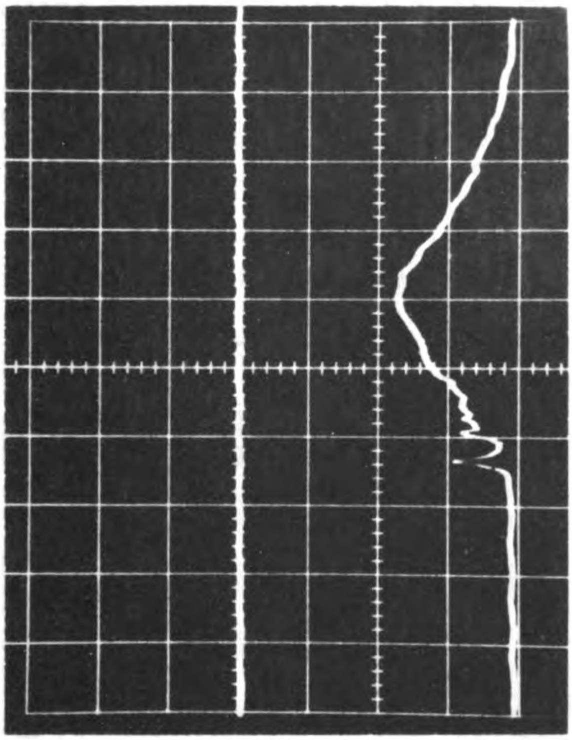

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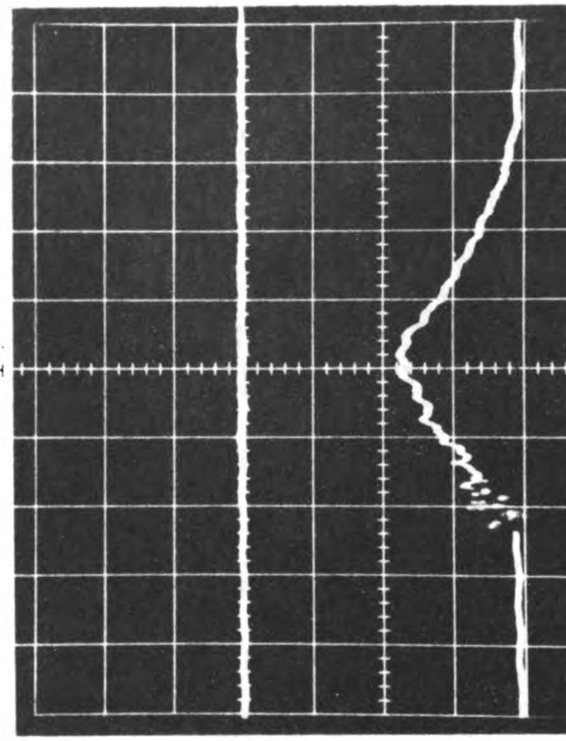

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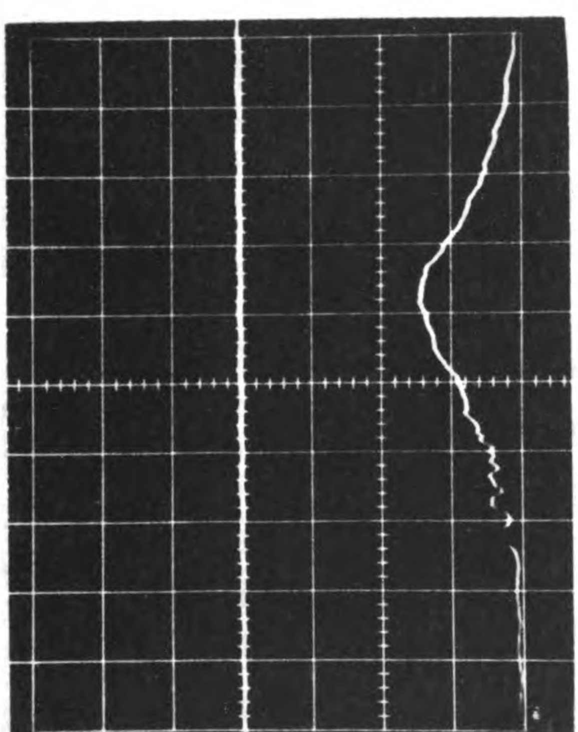

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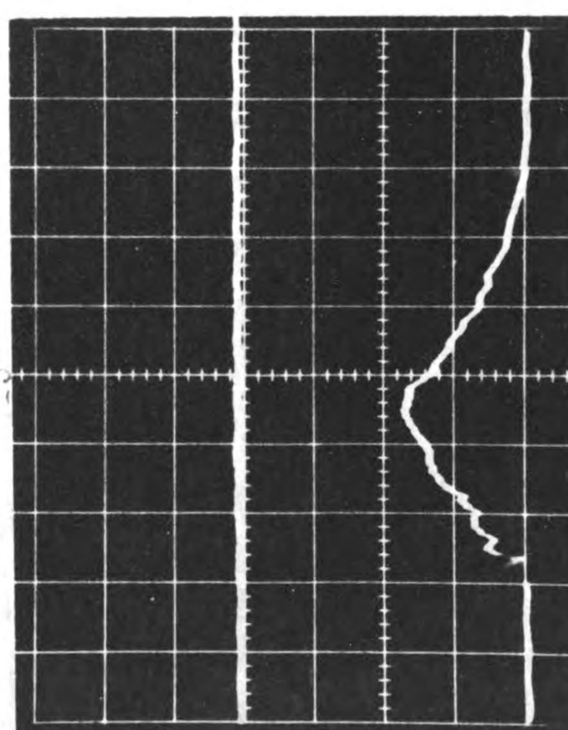

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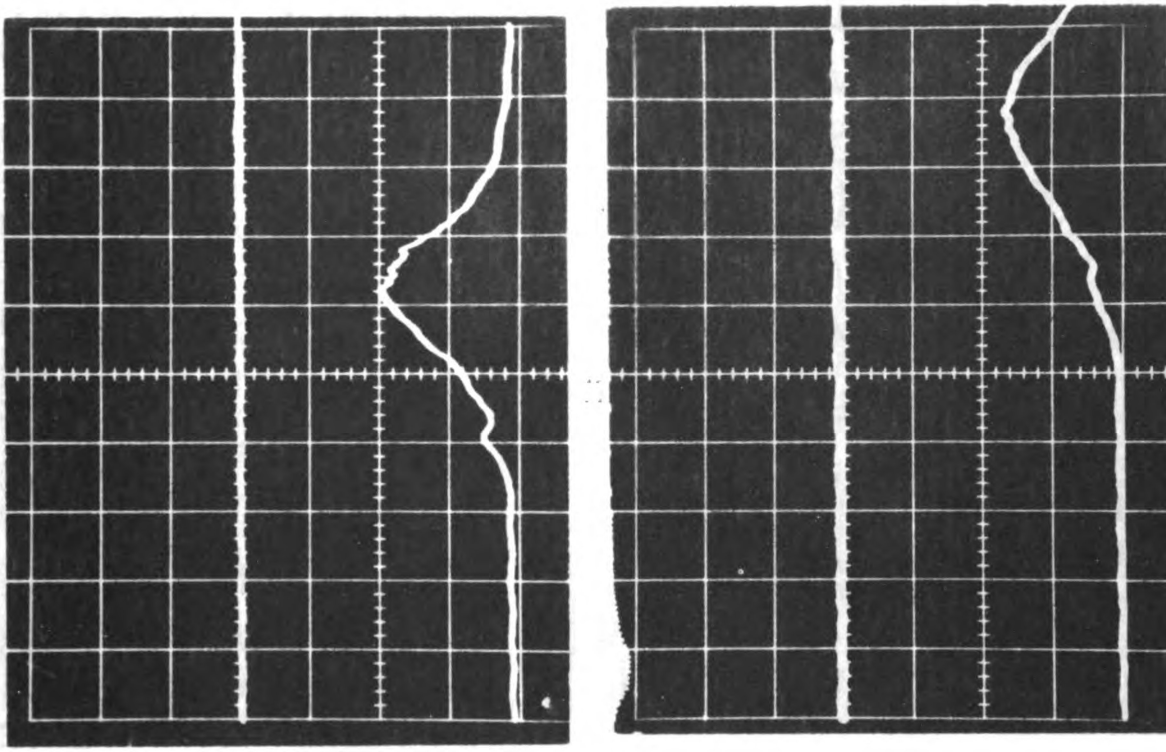

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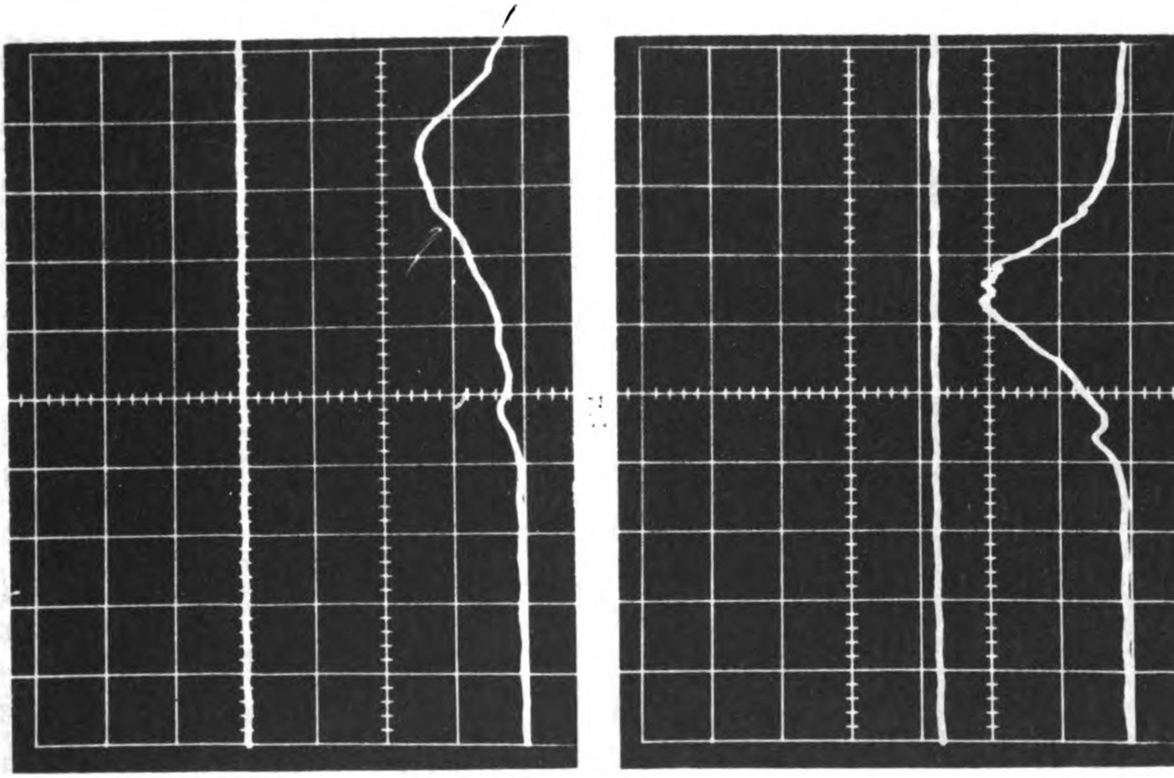

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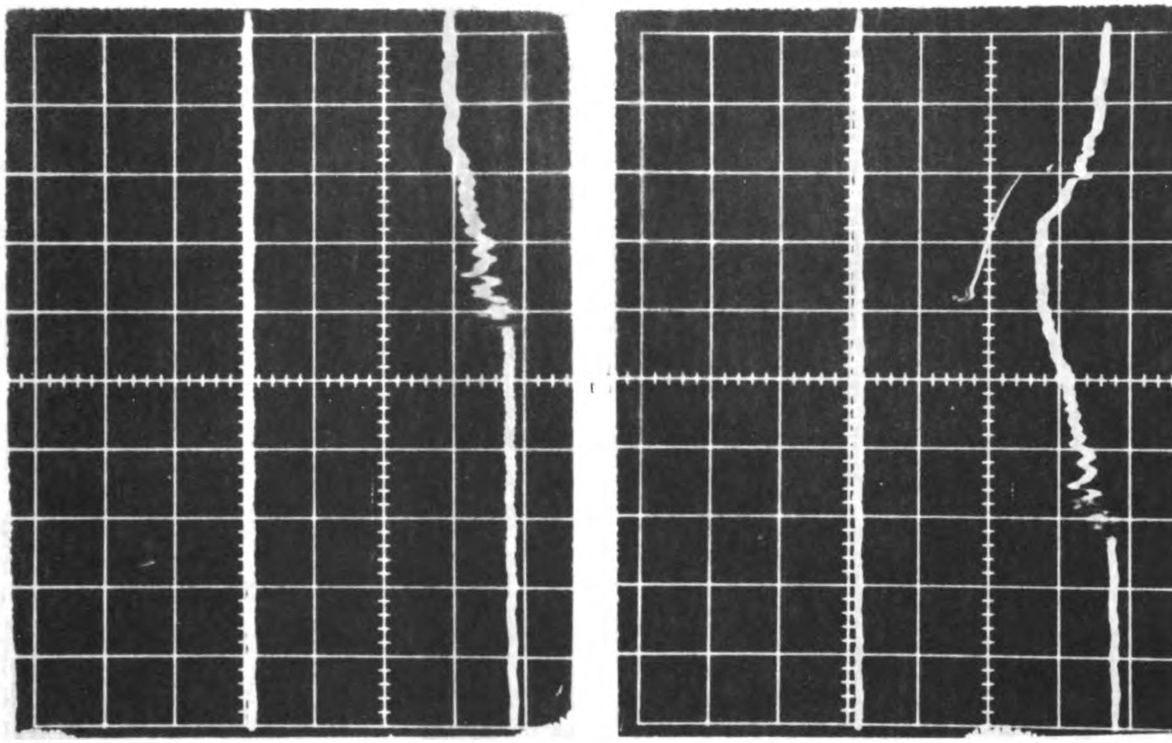

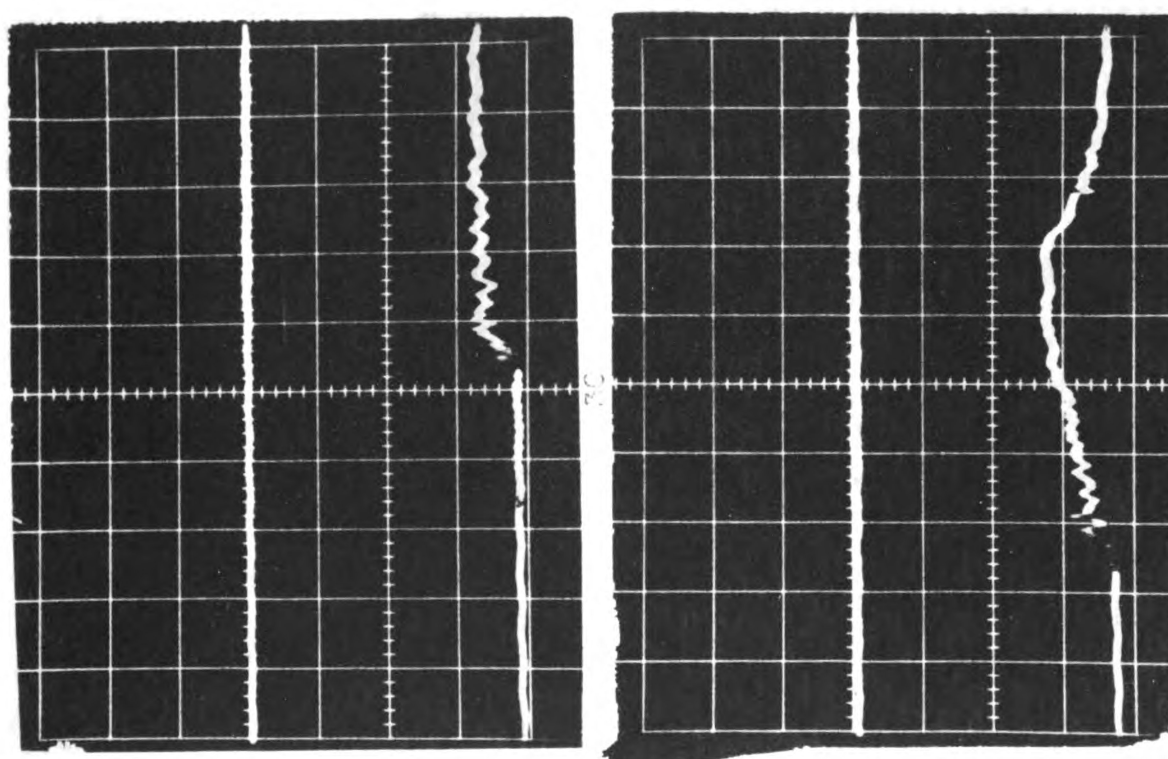

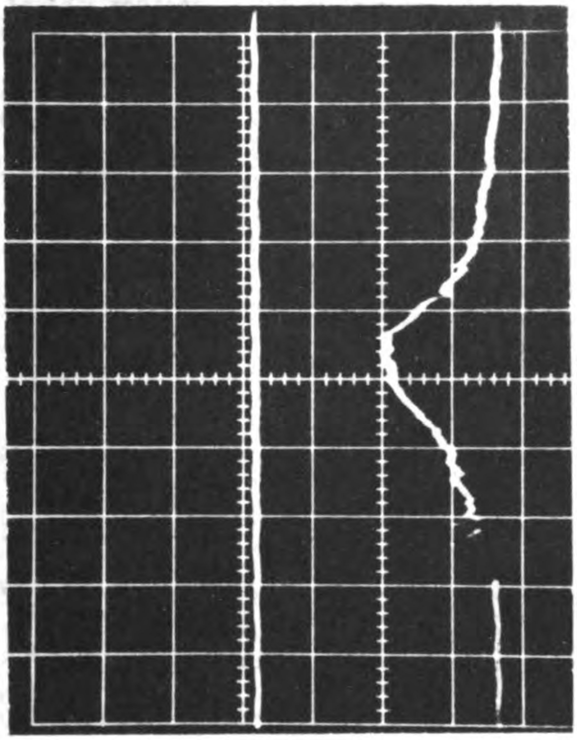

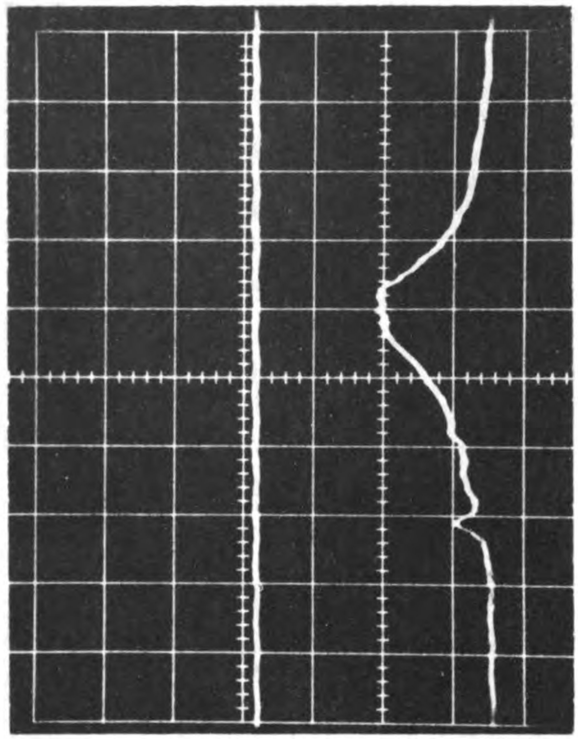

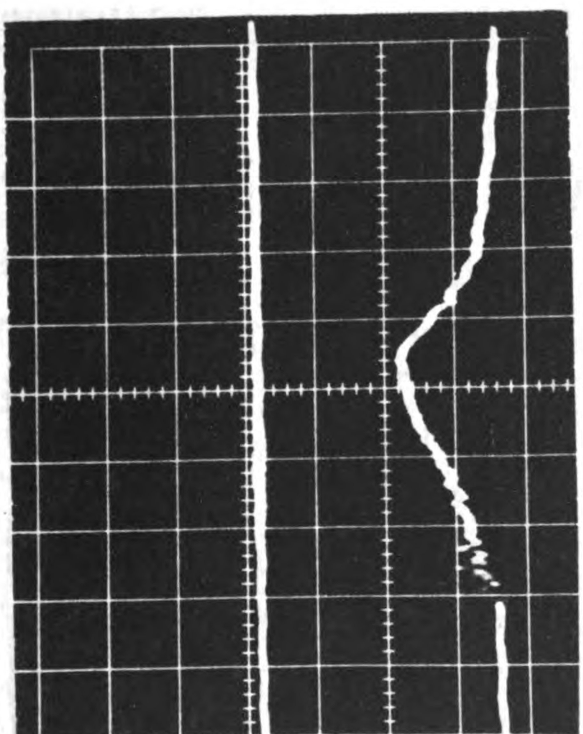

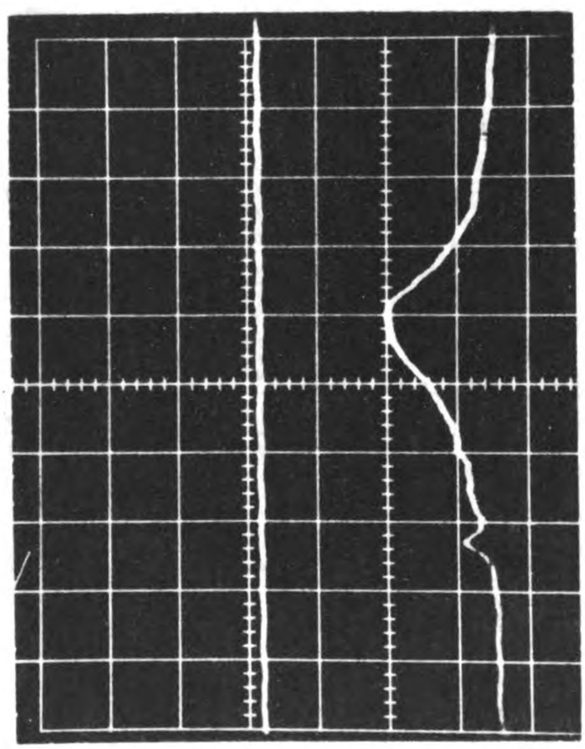

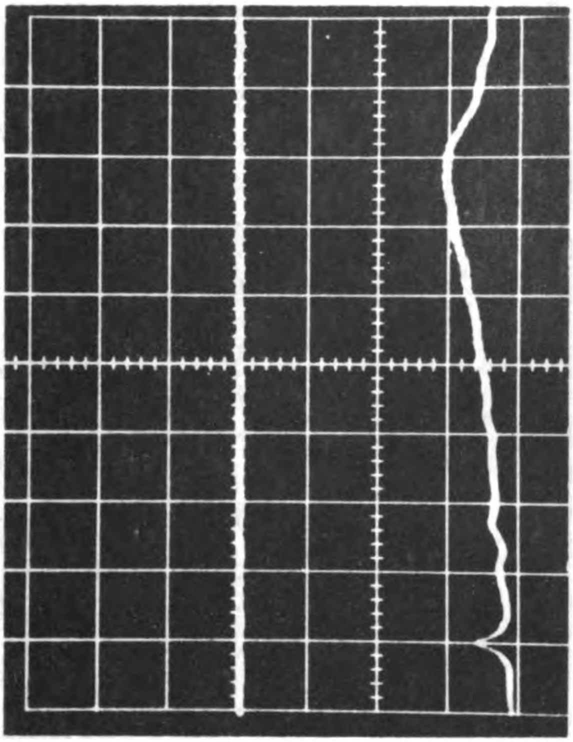

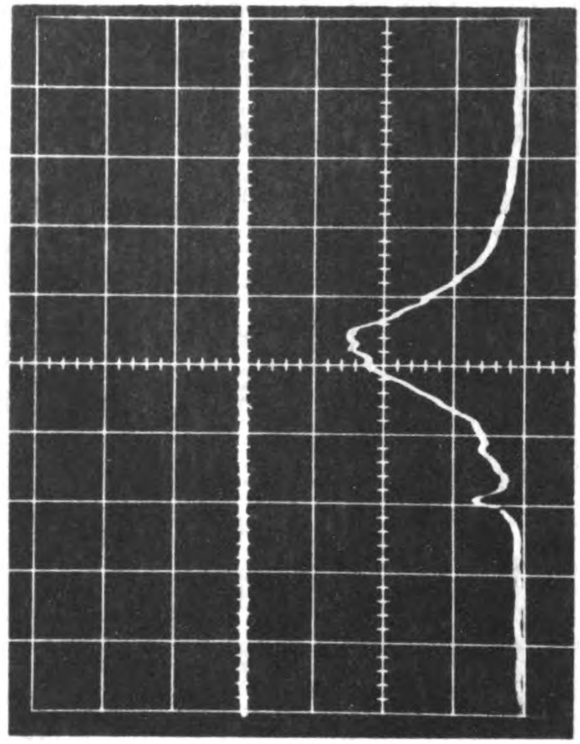

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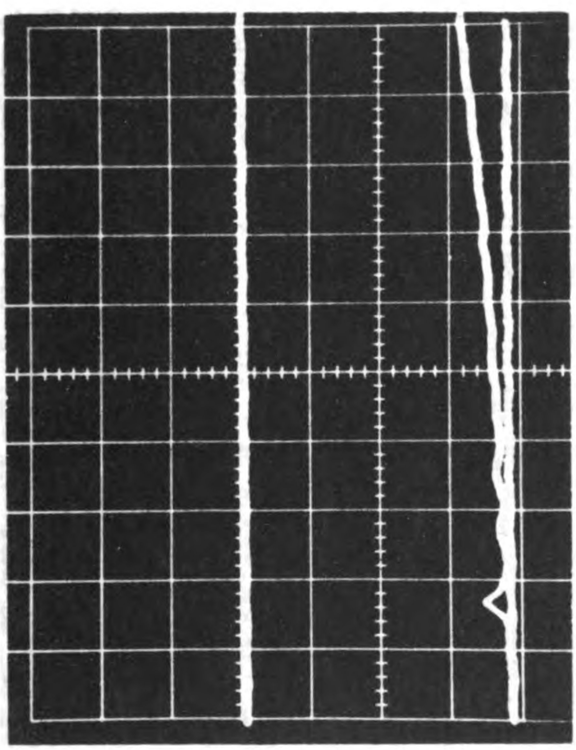

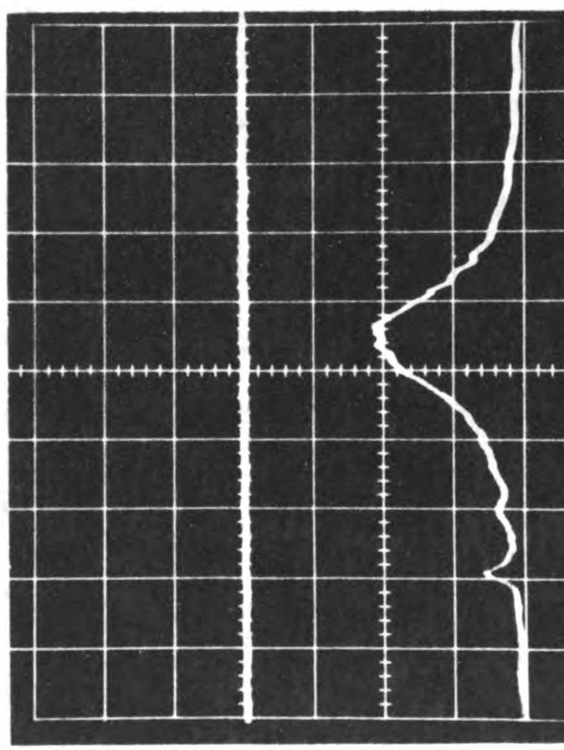

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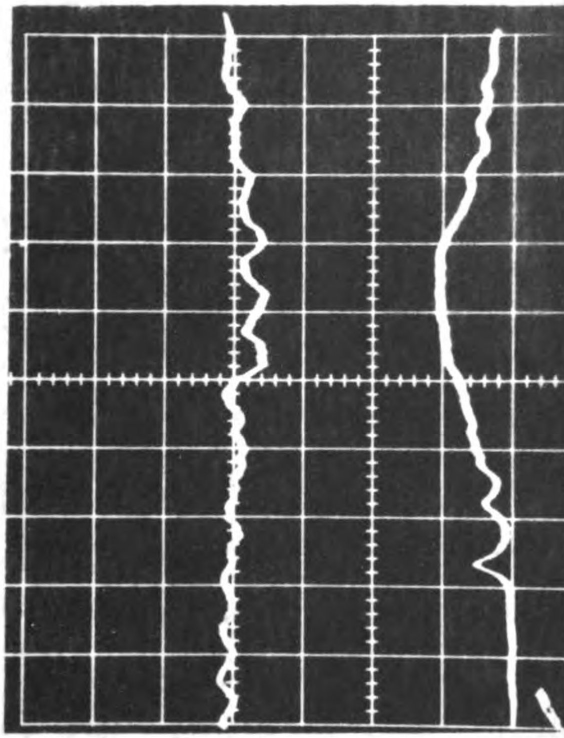

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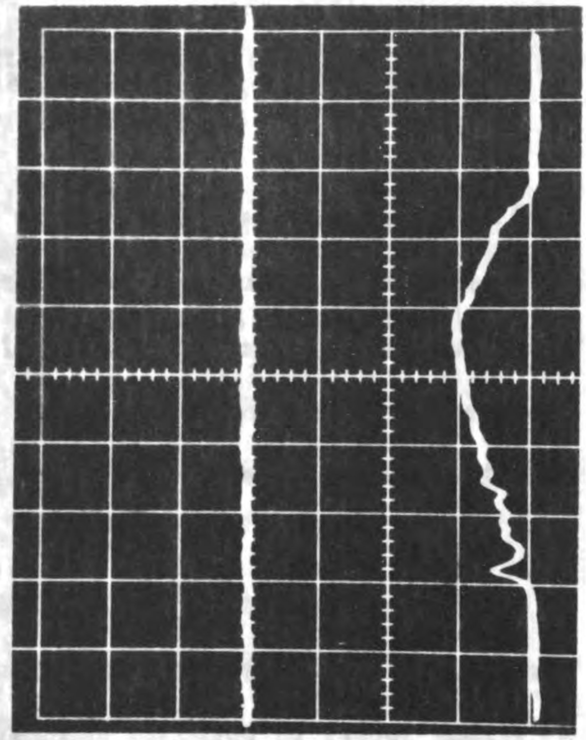

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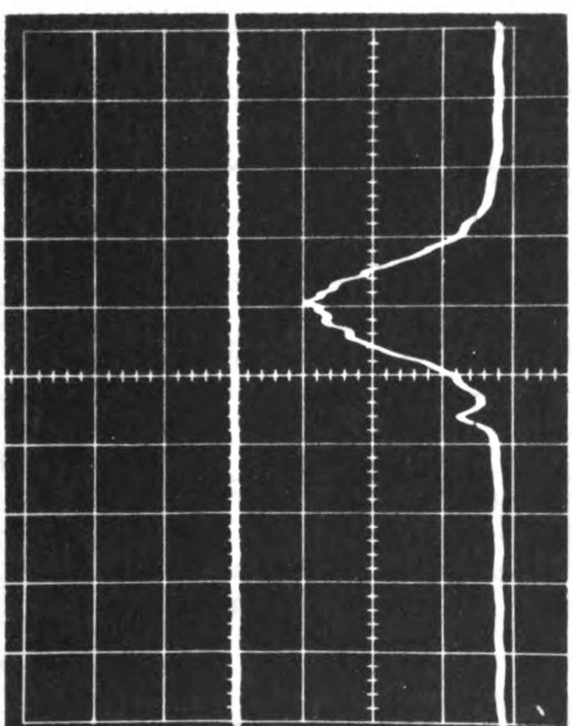
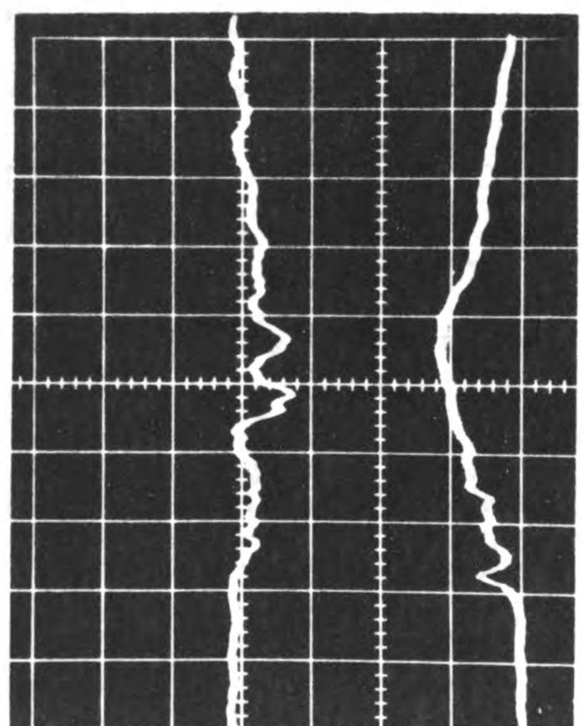




APPENDIX III

Behavior of Draft Gears

With Reference to Tables 4 and 5

1. The 10 in. Polystyrene-gear is more damping than the 5 in. felt gear, which might be seen from the rebound velocities measured. For the same thickness, the damping is almost the same for the two materials, both having felt dampers, which also might be seen from the area of the pulses after stabilizing.[39] (This measure is more difficult to obtain than rebound velocities.)

2. It takes 3-4 impacts to stabilize both felt gear and styrene gear, and the styrene has less mechanical strength (easy to tear, etc.).

3. None of the gears have preferences compared to rubber gears; therefore, they have not been used any further.

4. Measuring the rebound velocities is a handy way to follow the stabilization of a certain gear. For example, after 24 h and 48 h compression the stability of polystyrene was measured as follows:

Impact No.	1	2	3	4	48 h
Reb. Velocity	20.0	21.0	22.5	21.5	20.0
(in/sec.)	After 24 h				48 h

[39] ΔV measured by the pulse area is a monotonous function of the damping capability: when ΔV increases - for the same conditions of mass and impact velocity - then the damping of the gear decreases (more precisely: damping is the decrease in kinetic energy during the impact).

Some regeneration - although not sufficient for practical application - is occurring during rest-time.

5. The felt damper at the backstop has same damping effect at the noise vibrations as observed by the rubber gear, though a little higher effect in the case of felt gears as by the styrene gear. A rubber sandwich pad, normally used by drop-testing, showed up to have a damping effect on vibrations too, but at the same time it drags out the duration of the pulse again.

6. The polyethylene gear 8" has the better dimension stability of the 3 non-rubber draft gears tested:

	Thickness As New	Thickness After 5 Impacts	% Decrease
Felt	5"	2 7/8"	43%
Polystyrene	10"(6)	5 1/2"(3)	45% (50%)
Polyethylene	8"(3)	5 3/4" (1 1/2)	28% (50%)

Although felt and polystyrene has about the same permanent deflection, polystyrene has the lower decrease in the beginning, but continues to decrease at a higher rate than felt after the 5. impact.

When using the 3 in. polyethylene gear, (giving approximately the right pulse-duration) the permanent deflection becomes about 50%, as for the other non-rubber gears tested. It seems as though the rear rings of the non-rubber gears are shielded in the beginning and gradually will come into action by the longer gear; whereas, they contribute to the energy absorption immediately by the shorter gears.

7. Because of the permanent set, the smaller mechanical strength (mostly by polystyrene) and the not very outstanding damping capabilities, respectively energy absorption capability by the travel of gear, by a proper pulse duration (especially polyethylene), it was decided to use only truly resilient type gears like rubber, and to go no further with non-rubber gears.

8. The silicone rubber gear broke at the third impact along a radial plane in the O-ring-cylinders. The break surface was uneven. According to the commercial information:

Durometer A-Hardness, 45, maximum 65

Tensile strength, psi 400, maximum 700

Elongation, % 180, maximum 200

It seems improbable that any type of Silastic RTV (the self-curing type) can withstand the loads involved, if they are not strictly centered so that the whole cross-section area of the gear - about 10 $in.^2$ - is involved. We ought to have a safety factor such as 3 in tensile strength, which means a requirement of ca. 2000 psi. The urethane-rubber, mentioned previously, fulfills this requirement. Even if we have this safety factor, we do not know for sure whether the dynamic type of load will break it or not.

With 1750 lbs. peak force and t_o = 180 m S, the 4 1/2 in. silicone rubber has properties similar to the 8 in. polyethylene gear (Table 5).

9. To divide the total length of a draft gear into several smaller rings - instead of using one cylinder-ring - may have the effect that eventually skewed forces at the point of impact will be directed in the proper direction perpendicular to the gear after having passed 1 or 2 rings.

Table 5. Degradation of Selected Materials Used in Draft Gear Tests

Felt-Gear 5 Inch

	Impact Number	Thickness After Impact Inches	Peak Force (Faired Height) Lbs.[x)]	Duration (10% Limits) msec	Rebound Velocity in/sec.	
	1	3 1/2	-	-	17.5	
	2	3 1/4	2700	120	-	
	3	3	3100	100	19.0	
With	4	3	3400	90	20.0	
Felt	5	2 7/8	3300	90	20.5	
Damper:	6	2 3/4	3400	80	21.0	xx)
	7	-	-	-	-	
	8	-	-	-	22.0	
	9	-	-	-	21.0	
	10	2 5/8	-	-	21.0	
With Rubber Pad:		-	2700	110	-	

x) The uncertainty in force is about 50 lbs. (3%), in duration about 10 msec.

xx) ΔV (area) of No. 6 measured to 4.5 $\pm$ 0.2 (cm^2).

Table 5. Continued.

Polystyrene-Gear 10 Inch Respectively 6 Inch

	Impact Number	Thickness After Impact Inches	Peak Force (Faired Height) Lbs.	Duration (10% Limit) msec	Rebound Velocity in/sec.	ΔV (cm^2)
10 Inch:	1xxx)	8	1000	160	10.8	
	2	7	-	-	12.7	
	3	6 1/4	-	-	14.2	
	4	5 3/4	1500	-	15.8	
	5	5 1/2	1550	150	16.8	
	6	5 1/4	1750	140	-	
	7	5	1800	140	15.7	
6 Inch:	8	3 1/4	2300	130	-	
	9	3	2600	110	17.5	
Felt	10	3	2650	110	20.0	4.3
Damper:	(11)	2 7/8	2700	110	22.0	4.3
1 1/2 In:		1 1/4	4650	60	20.0	4.1

(After 48 h in Compression)

xxx) Characteristic pulse with flat top lasting more than 80 msec.

Table 5. Continued.

Polyethylene-Foam Gear 8 Inch Respectively 3 Inch

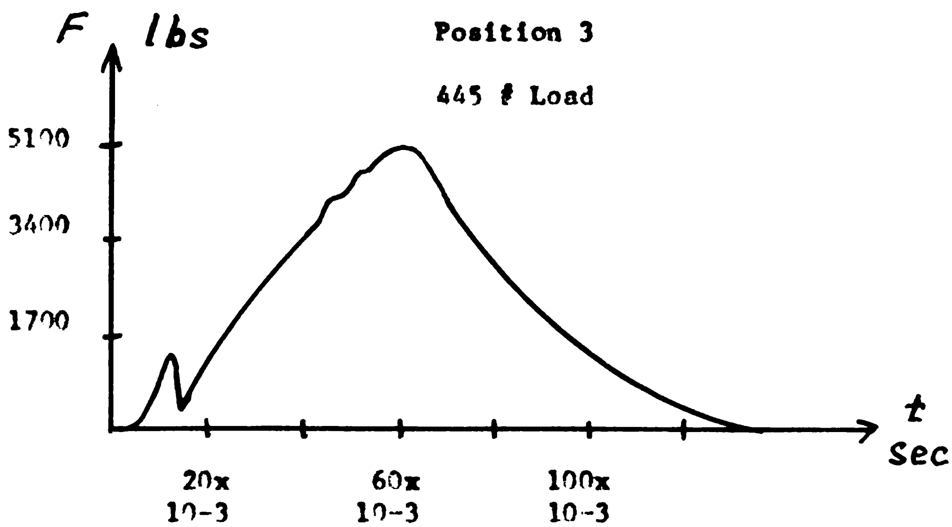
	Impact Number	Thickness After Impact Inches	Peak Force (Faired Height) Lbs.	Duration (10% Limit) msec	Rebound Velocity in/sec.
8 Inch:	1	6 1/4	1150	200	25.5
	2	6	1400	200	22.5
	3	6	1750	200	23.5
	4	6	1750	200	23.5
	5	5 3/4	-	-	24.5
3 Inch:	6	1 1/2	3900	120	24.5
	7	1 1/2	4100	90	23.5

Figure 7

6 Neoprene Gear + Felt Damper

Position 3

445 # Load



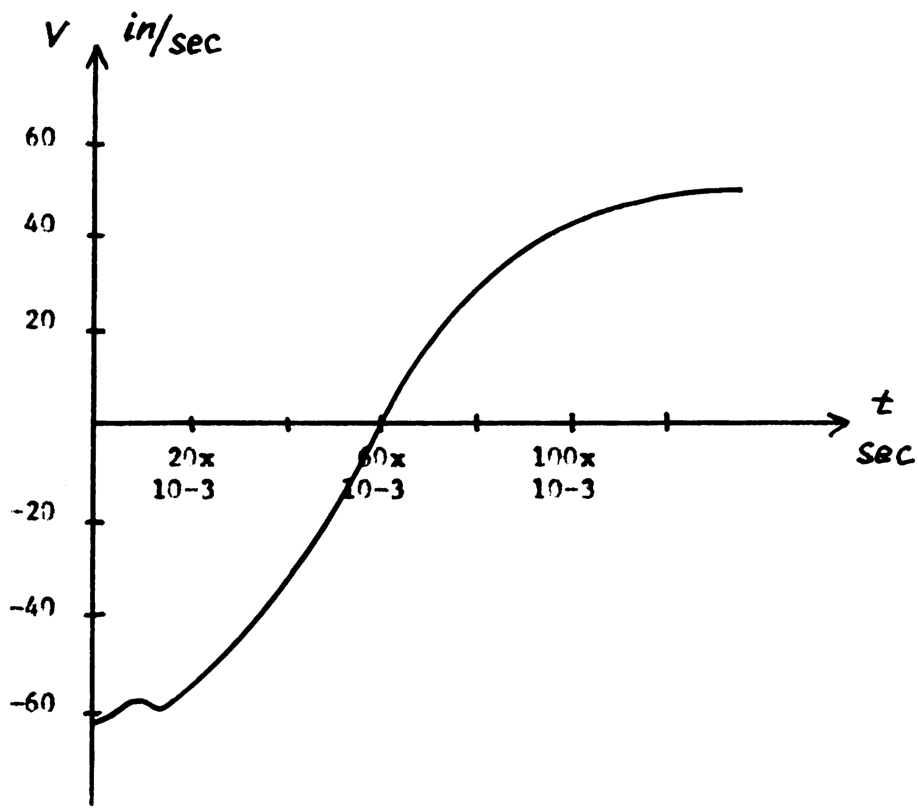
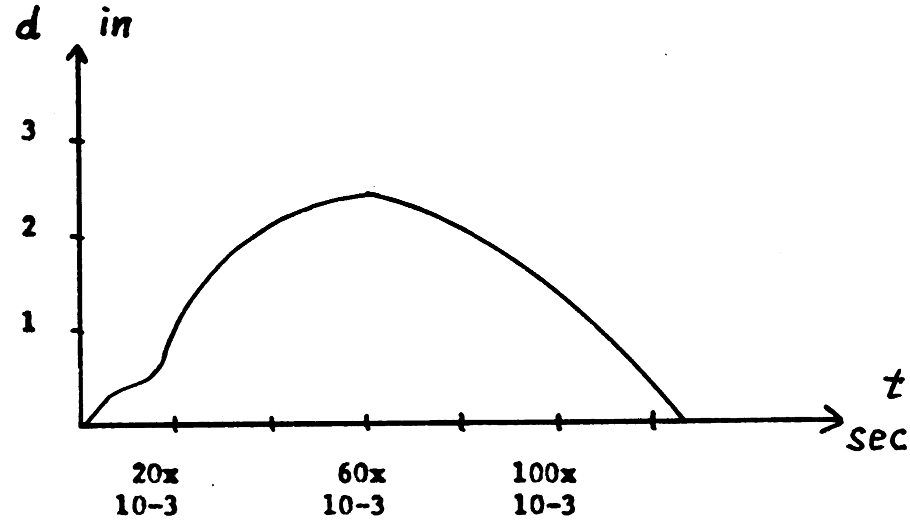
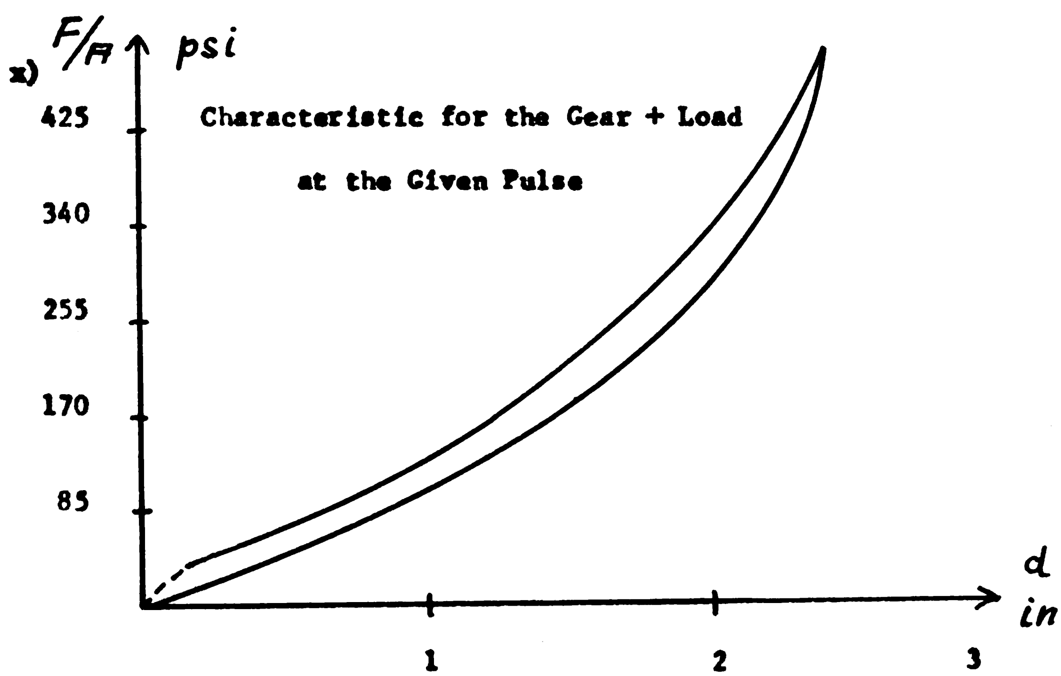


Figure 7 Continued





x) Calculated for the Rubber Part of the Gear Having a Cross Section Area of 10 in.^{2}

APPENDIX IV

Characteristics for the Test Track System

After commenting on the travel-characteristic of the draft gear (Figure 7) the characteristic diagrams for the whole system (Figures 5 and 6, Table 6) will be discussed.

<u>Travel of the draft gear</u>

The combined travel 2.4 in. of the 6 ring rubber gear under the stated conditions is composed of:

travel of felt damper	$\sim$	3/4 in.
backstop-movement	$\sim$	1/4 in.
due to rubber-spring, etc.	$\sim$	1.4 in.

With 6 rubber rings each 3/4 in. = 4.5 in. the available travel is about 0.80 · 4.5 in. = 3.5 in. So we still have some available travel of the draft gear. If we want considerably more energy absorbed without changing the duration of the pulse, the rubber or spring type gear must be combined with or replaced by other types of gears, as discussed before, or the cross-section area must be increased.

The characteristic for the draft gear (Figure 7) has uncertainties because of the several links involved in deriving it. Although, it might be better than a characteristic taken by <u>static</u> load application as presented in the earlier report (1).

Some correspondence is obtained between the static load characteristic and the present characteristic. For example, in the earlier report a force of 2800 lbs. corresponds to a travel of 2.0 in. of a 7 ring gear, while here a force of 2800 lbs. corresponds to a travel of 1.4 in. of 6 ring, thus a smaller travel by the dynamic load is indicated.

Duration

The formula $T/2 = \pi\sqrt{m/k}$ (also valid for a rather high damping) does not hold, eventually because of the two-lumped mass or rather the distributed mass (discussed below). When comparing 0 load (575 lbs.) at 2900 lb. force with 625 load (1200 lbs.) at 2900 lb. force, the k-factor should be the same, and duration should therefore deviate with a factor of $\sqrt{1200 / 575} \cong 1.5$. They only differ with a factor of 1.2, and the observation therefore supports the concept of distributed mass in the simulation. Several authors account for at least a two-lumped mass (11).[40)]

If the mass is assumed to be lumped,

$$k_{1200,\ 2900} = (1.5 / 1.2)^2 \cdot k_{575,\ 2900} = \underline{1.56} \cdot k_{575,\ 2900}$$

The apparent k-factor becomes higher, as the impacting mass is increasing, although the force is the same. This is not consistent with the concept of k as a staticially determined factor (the spring itself can be considered as massless), and we interpret the result in terms of distributed mass 100 / 1.56 = 65% of the actual mass relative to 0 load.

---

40) When considerable amount of cubic elasticity, multiply with the factor:

$$\sqrt{2 / B\ (1+B)\ (-1+\ \sqrt{1+B})}\text{, where}$$

$$\underline{B} = 4\ mghr / k^2 = \underline{2\ V_m^2\ m \cdot r / k^2}$$

where r is the parameter appearing in the approach equation for the characteristic curve:

$$F = k \cdot x + r \cdot x^3 \quad \text{(Mindlin, pp. 11-13)}$$

Only a small amount of cubicity appears by moderate loads.

On the other hand, when comparing 220 load (795 lbs.) at 4550 lbs. force with 625 load (1200 lbs.) at 4250 lbs., there is no such difference. There is no difference in the relative distribution of mass in these two cases, (as far as effect upon duration is concerned) as the factor $\sqrt{1200 / 795} = 1.23$ explains the different observed durations well within the uncertainty of the measurement ($\pm$ 5 m S).

The rise time for the pulses is about 40% of the total pulse length for all pulses.

Load factor-static-load-curve (Figure 6)

For some impact velocities, the load factor ("G"-factor) is plotted against total mass m ($F_m = m \cdot G_m = m \cdot G \cdot g$). It is interesting to compare to the well known curves for cushioning materials, where the peak acceleration is plotted against static load of the dropping head of the testing machine for different drop heights (equivalent to different impact velocities).

Expectedly the curves would go down ($G_m = V_i \sqrt{k/m}$ except for damping) as long as k remains constant or rises less than the mass. Not only stiffness and damping, but influence of distributed mass (and eventually amplification factors) complicate explanation of the curves.

Assuming the damping to be approximately the same at 1020 lbs. as at 575 lbs. (about the same percentage decrease in G relative to G_o, which is justified by comparing hysteresis areas under the load-displacement curves, Figure 7), we get:

$$(G_{1020,\ 40} / G_{575,\ 40})^2 = (575 / 1020) \cdot (k_{1020,\ 40} / k_{575,\ 40});$$

$$k_{1020,\ 40} = (2.8 / 3.05)^2 \cdot (1020 / 575) \cdot k_{575,\ 40} = \underline{1.5} \cdot k_{575,\ 40}.$$

$$k_{575,\ 40} = k_{1600\ \text{lbs.}} = \underline{1760\ \text{lbs/in.}}\ \text{(stress-strain curve)}$$

$$k_{1020,\ 40} = (k_{2900\ \text{lbs.}}) = 1.5 \cdot 1760 = \underline{2640\ \text{lbs/in.}}$$

According to the static curve:

$$k_{2900} = \underline{4800 \text{ lbs/in.}}$$

Therefore, we may assume that the static k-concept is not valid in this case, or we may calculate an "equivalent mass" [41] different from the actual mass by working back:

$$(2.8 \ / \ 3.05)^2 = (575 \ / \ W_{eq}) \cdot (4800 \ / \ 1760); \ W_{eq} = 1840 \text{ lbs.}$$

$$\text{Amplification} \ \underline{A_m} = 1840 \ / \ 1020 = \underline{1.8}$$

Now the static characteristic is not worked out for higher loads; it is suggested to determine the stiffness-characteristic more accurately for further calculations and comparisons.

Also, another method is tried, as follows:

By merely crossing the curves in Figure 5 and assuming k constant by constant peak force:

$$F_m = V_i \sqrt{km} \ ; \quad F_m{}^2 \ / \ k = \underline{V_i{}^2 \cdot m} = \underline{\text{Constant}}$$

(still assuming damping as a certain percentage of the total work being done), we get to (for 1020 lbs. load):

$$W_{eq} \ (2900 \text{ lbs.}) = 575 \cdot (53^2 \ / \ 40.5^2) = \underline{930 \text{ lbs.}}; \ A_m = \underline{0.96}$$

$$W_{eq} \ (4400 \text{ lbs.}) = 575 \cdot (71.5^2 \ / \ 44.5^2) = 1480 \text{ lbs}; \ A_m = \underline{1.45}$$

---

41) The mass participating in the particular impact divided by static mass may be thought of as the "amplification factor" in this case (Mindlin, p. 71).

It is reasonable to believe, the $A_m \sim 1$ by lower loads. The value of W_{eq} is consistent with the finding $k_{2900} = 2640$ lbs/in. The earlier characteristic then seems to be wrong, but only for higher values of force such as for the 2900 lb. load. The foregoing calculation of $A_m = 1.8$ then must be wrong too.

It is therefore suggested to calculate equivalent load or the amplification factor according to the crossing method, instead of giving up the constant k-concept.

At load factors less than about 3, the amplification effect is negligible. By higher load factors such as 4.5 (4400 / 1020), the amplification is about 1.5, etc.

The influence of distributed mass cannot be distinguished from the influence of amplification by these measurements.

Characteristic Curves (Figure 5)

The shape of the peak-force curves are fairly consistent with the fact that the characteristic of the draft gear is somewhat cubic, in other words, shows increasing stiffness (increasing k-factor) with increasing load (Figure 7):

$$F_m / m = V_m \cdot w = V_m \sqrt{k/m}$$

$$F_m = V_m \sqrt{km}$$

Approximately right for small damping,[42] small cubicity of the characteristic [40] and amplification factors close to 1.[41] The duration of pulse is between a low of 85 and a high of 125. At the velocity, we want to use in the further work: 4.0 mph., 70 in/sec., the duration is between 90 and 105 msec which corresponds fairly close to the duration found in the field (75-100 msec).

---

42) For a matter of interest, when knowing damping, energy loss $\psi = 4\pi\beta$ or logaritmic decrement $\Delta = 2\pi\beta$ for the spring, and by using

$$F_m = F_{om} \cdot e^{-\beta\pi} = V_m \sqrt{k/m} \cdot e^{-\beta\pi}$$

for small damping (see Mindlin p. 50), we may calculate stiffness from the measured values of F_m and V_m, or reverse (assuming also that there is no influence from amplification factors).

APPENDIX V

Sample Derivation of Velocity, Displacement and Stress-Deflection-Curves for the 6 Ring Rubber Gear with Felt Damper

For the test gear No. 10: 6 ring Neoprene gear + felt damper on backstop, Position 3, 445 lb. load (1020 lbs. total), a picture was taken of the pulse (No. 14), that is redrawn in Figure 7. From that we derived the velocity - pulse by graphical integration using the formula:

$$V_t = V_i + \int_0^t (F/m)\, dt = V_i + 386 / W \int_0^t F\, dt$$

$$V_i = -\ 61.0 \text{ in/sec}$$

$$W = 1020 \text{ lbs}$$

V_{120} (the final velocity) = <u>48 in/sec.</u>

ΔV_{120} = 61.0 + 48.0 = <u>109.0 in/sec.</u>

which is compared to the <u>measured</u> velocity:

61.0 + 37.0 = 98.0 in/sec.

The uncertainty involved in the graphical method of integration is hardly more than 2%.

Also we derived the displacement-time curve by using

$$d_t = d_o + \int_0^t V_t\, dt \ ; \quad d_o = 0.$$

d_m (the maximum deflection) occurring at t = 60 m S is <u>2.4 in.,</u> which is compared to the <u>measured</u> combined travel of the draft gear: 2 3/8 in.

The travel is determined by successive movement of a light beam with increments of 1/4 in., until the beam would not be switched off by the impact. Then the distance from the backstop at floor level was subtracted from the distance at the point where the draft gear was just touching the backstop at $V_i = 0$.

The rebound velocity was measured in the same manner as the impact velocity, changing the switches around and coupling the counter into the system just after the impact began. The switches were hit by the proper front wheel just after the impact was over.

Discussion

(1) Directional Forces and Velocity

Dissipation of energy in the rubber gear itself and in the felt damper is relatively small. Absorption of energy in the backstop is due to the movement of the backstop, which is not totally rigid. According to the law of momentum:

$$V_i \cdot m = M \cdot V_2 + m V_R$$

$$M = (V_i - V_R) / V_2) \cdot m$$

where

$V_i \cdot m$ is the impact quantum (IQ).

$V_R \cdot m$ is the rebound momentum for the car.

V_2 is the velocity of backstop just after the impact.

M is the equivalent mass of the backstop, which is expressing the degree of rigidness.

Now the maximum displacement of the backstop is about 1/4 in. at the point of impact (measured by a pencil fastened to the backstop), and when a symmetric shape movement of the backstop is assumed, the final velocity V_2 will be determined by:

$$\Delta V_2 = (2000 / t_o \text{ (m S) }) \cdot d_{m2} = (2000 / 120)\ 0.25 = 4.2 \text{ in/sec.}$$

Substituting gives:

$$\underline{M} = (61 - (-48)) / 4.2 \times m \quad = \underline{26\ m}$$

The pencil test also revealed that the backstop moves at an angle of about 20^{o} with the horizontal, partly because the cylindrical gear hits the tubular load cell excentrically (it is observable by empty car and small load at 220 lbs.), and probably the impacting surfaces are not exactly parallel either (which might be very difficult to obtain). Also the backstop as a whole seems to be too weak and the bolts in the floor are not able to hold it down.

From this it is clear that the impact is acting partly in another direction than the horizontal, and when we are <u>measuring</u> the rebound velocity only in the horizontal direction, we come to a result which is less than that derived from the pulse.

Other Sources of Errors Discussed:

(2) <u>Influence of Non-lumped Mass</u>

As pointed out before, the load is not acting together with the car as one mass. Now in this case the wooden blocks are rigidly fastened and fill out the entire car length.

(3) The Error in Measuring Rebound Velocity

The several measurements of rebound velocity show that the error 2σ is less than 5% even with one single measurement. The measurement is taken just after the impact.

(4) The Static Calibration Method

For strain gage accelerometers one may expect that under dynamic conditions, the response would be somewhat lower so that the real force is higher than measured. This error will force the result in the opposite direction of our finding for ΔV, and therefore there is no indication for that kind of error.

(5) Influence of Distributed Mass

When the mass is not acting as one lumped mass, we may expect that the mass participating in the impact is different in different stages and durations of the impact (the shock wave concept, also Part B) and less than the constant mass we assume in calculation of ΔV. Then the real ΔV would be larger than calculated, whereas we found ΔV already too large compared to the measurements. Therefore, the observation gives no indication for this error either.

(6) Determination of Weight of Components

It is believed that the weight of the wooden blocks and the car components is still the same as that measured about two years ago (before the author). The time and facilities did not allow for controlling that information. Wooden blocks could have dried out to some extent.

The very fine correlation between measured and calculated displacement does not necessarily indicate accuracy. In two other cases (gear No. 15, felt gear, and gear No. 10, at Position 2 and 0 load) was found:

	Δ V		dm	
	calc.	meas.	calc.	meas.
Felt gear	94	76	1.8	2.4
Rubber gear (0 load)	99	80	2.2	1.8

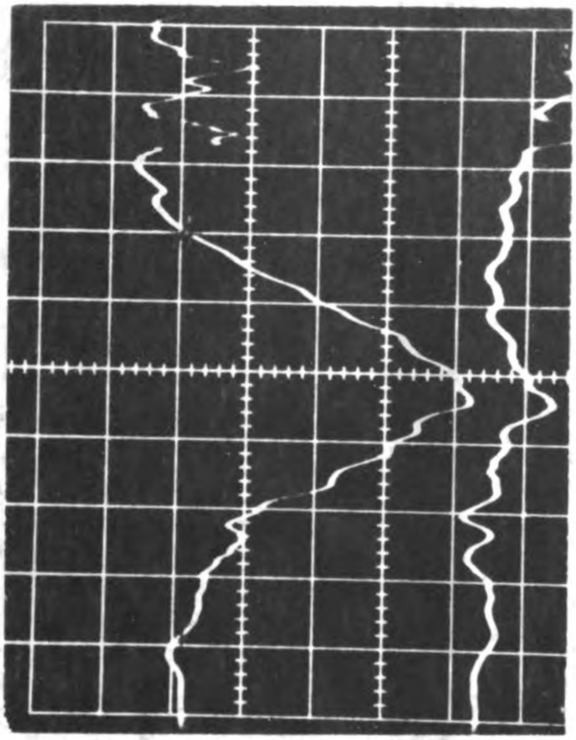
In both double integration and in the beam method for determining travel there is a considerable uncertainty.

Despite the deviations, it is concluded that the comparisons described above prove the general validity of the methodology and the size order of the measurements.

APPENDIX VI.

Selected photographic series for

lading pulses, Tables 7, 8, 9 and 10.



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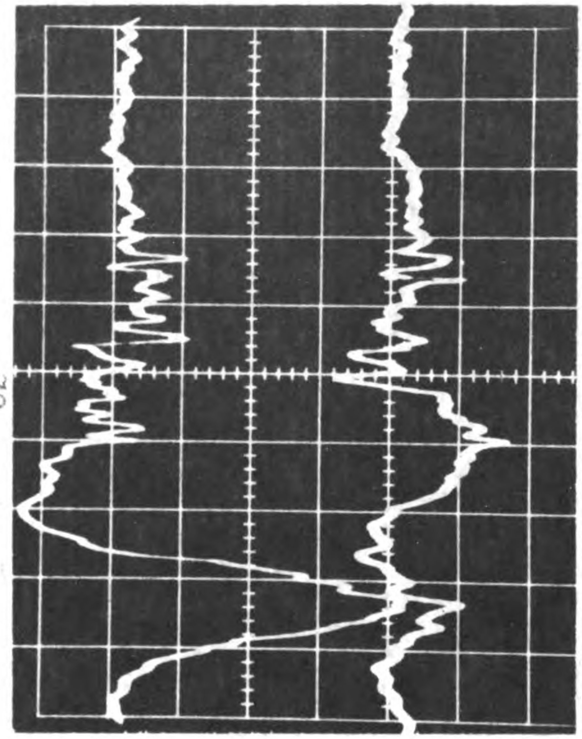

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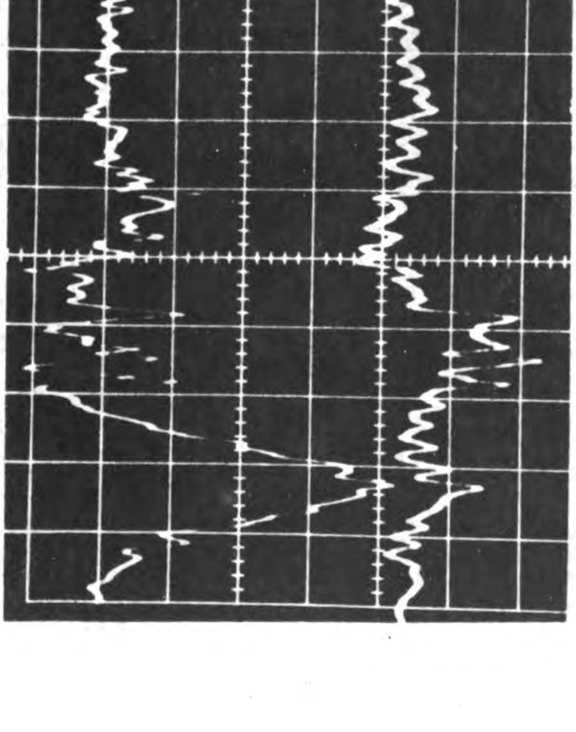

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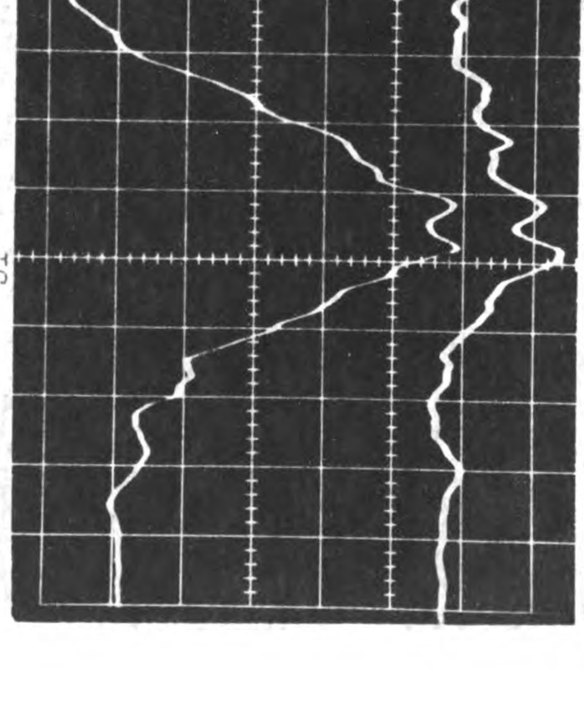

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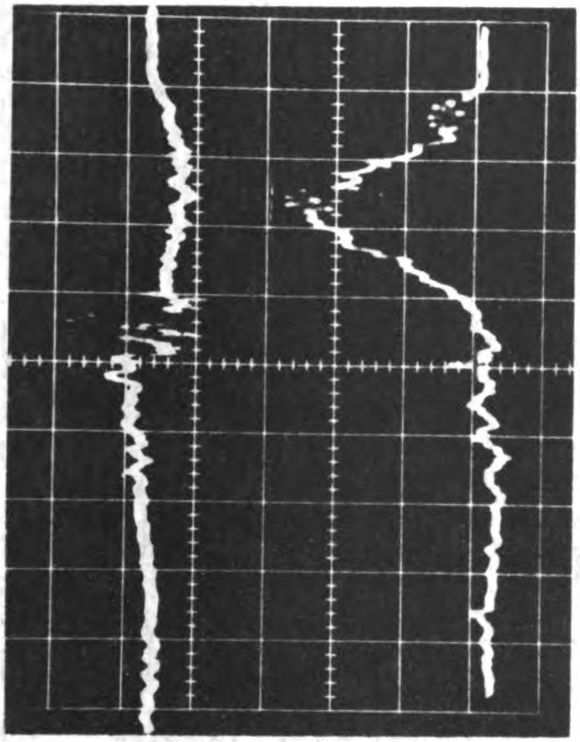

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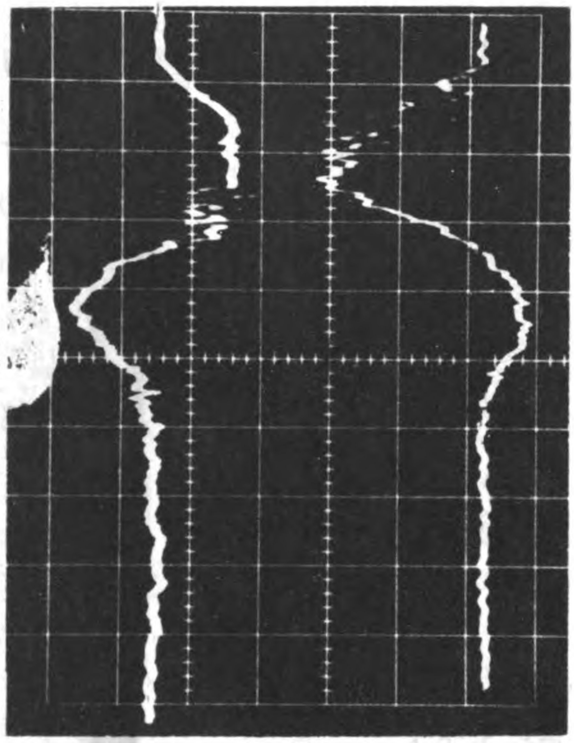

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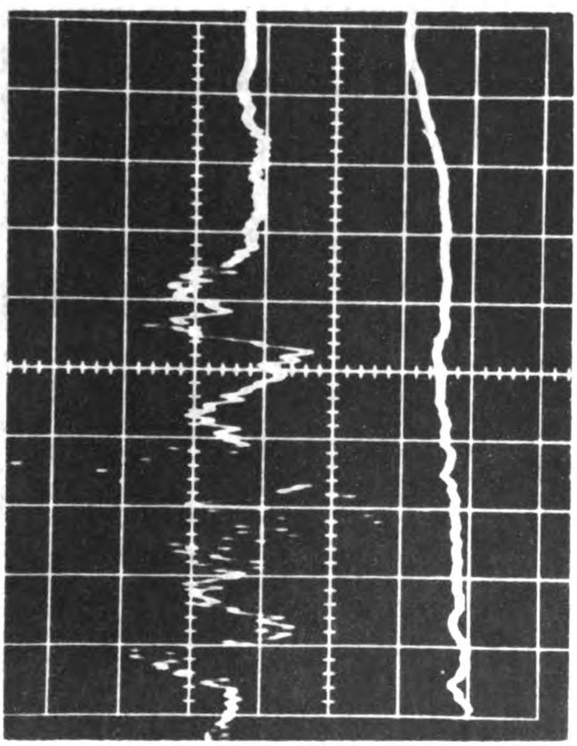

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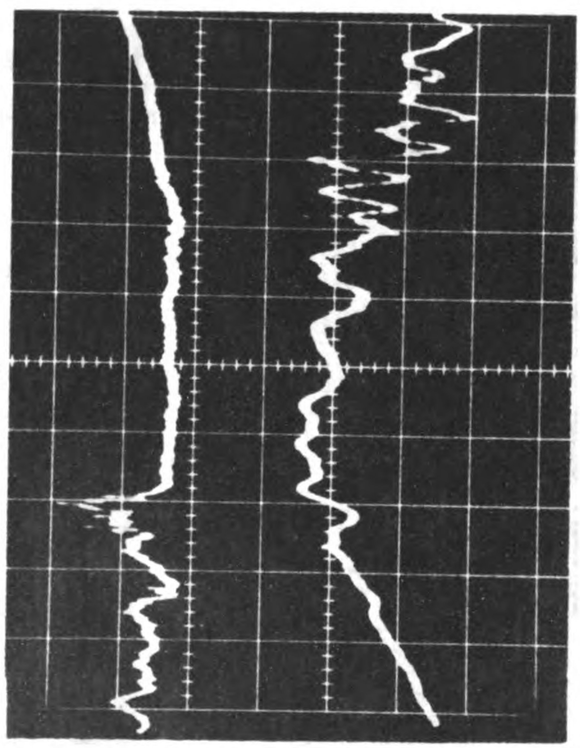

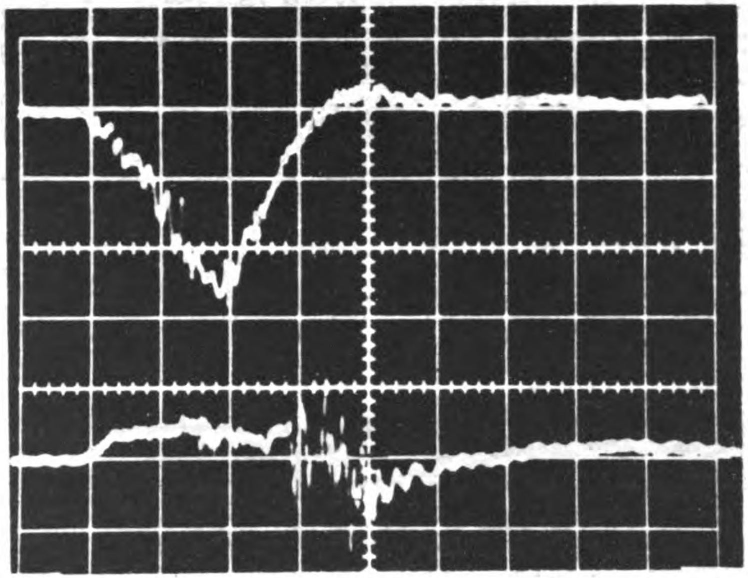

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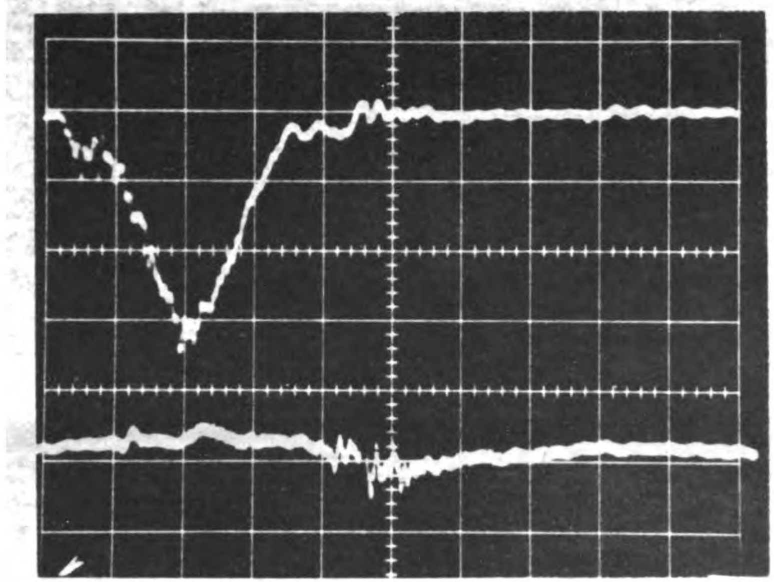

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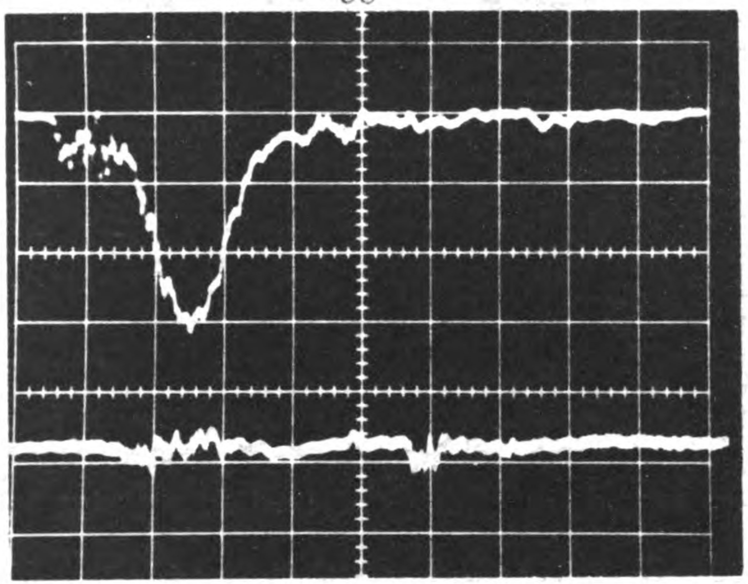

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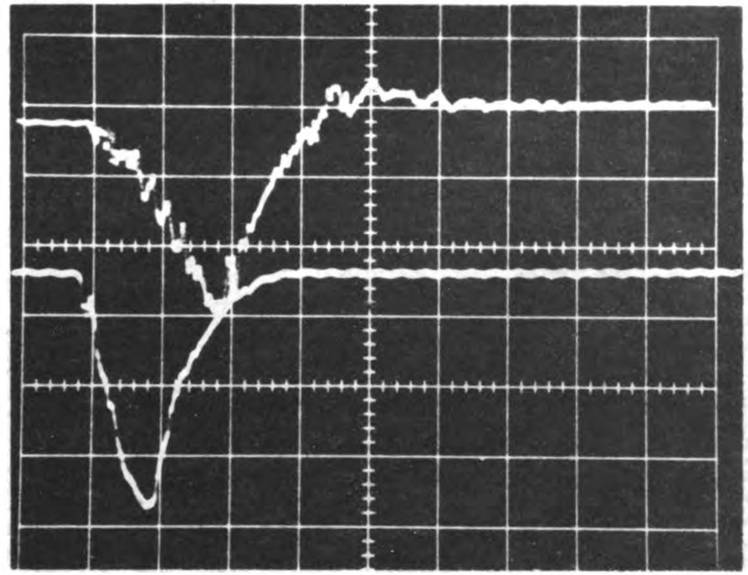

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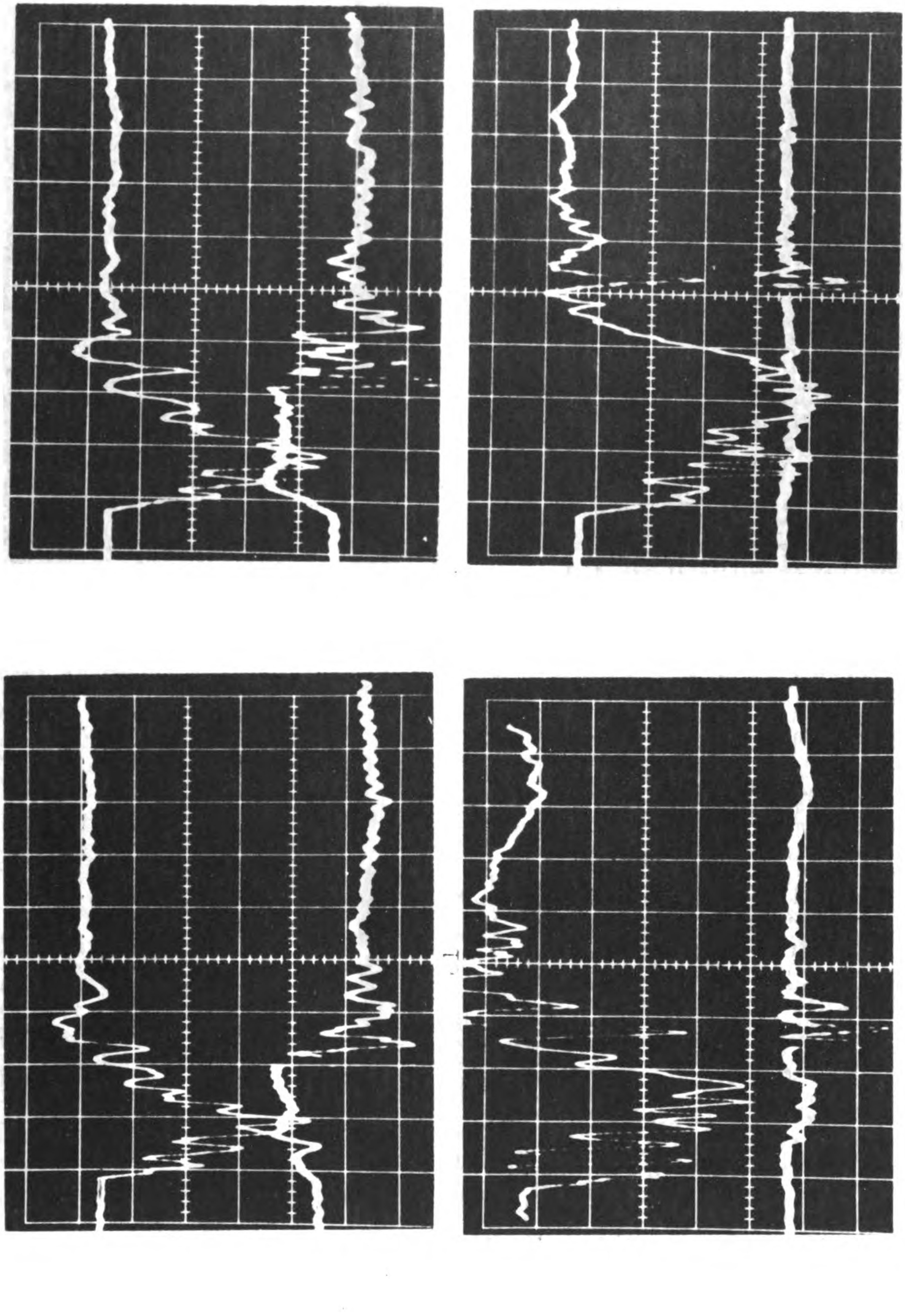

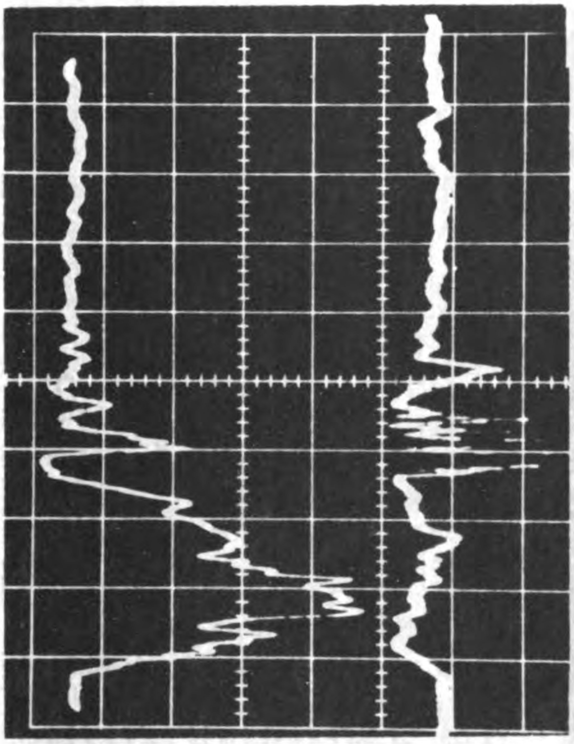

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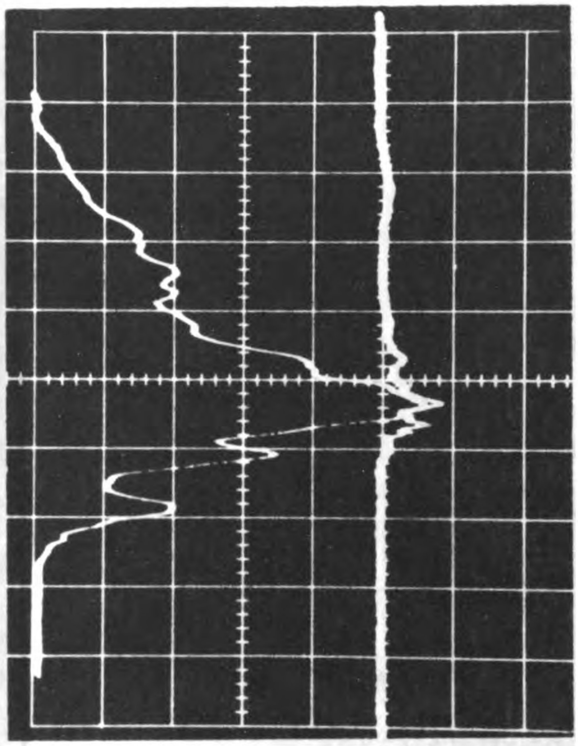

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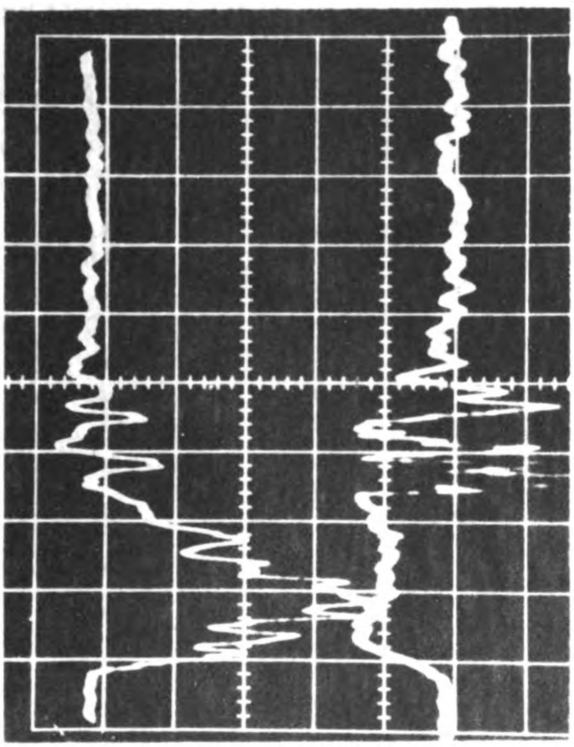

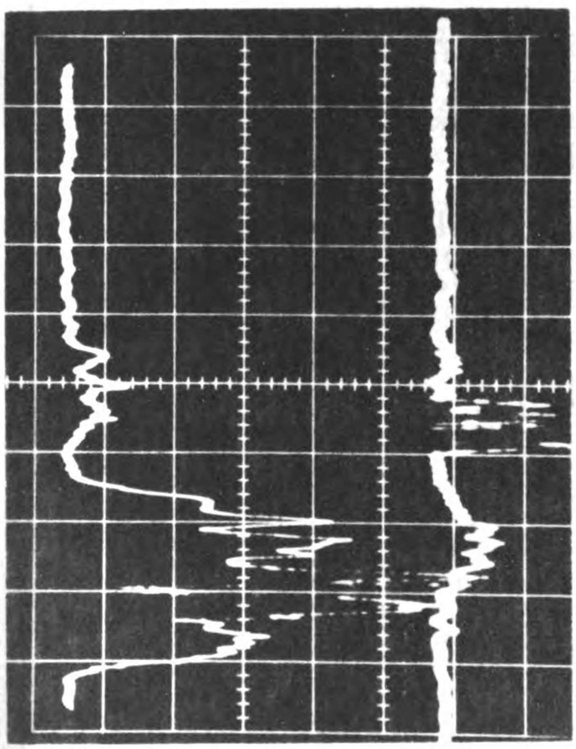

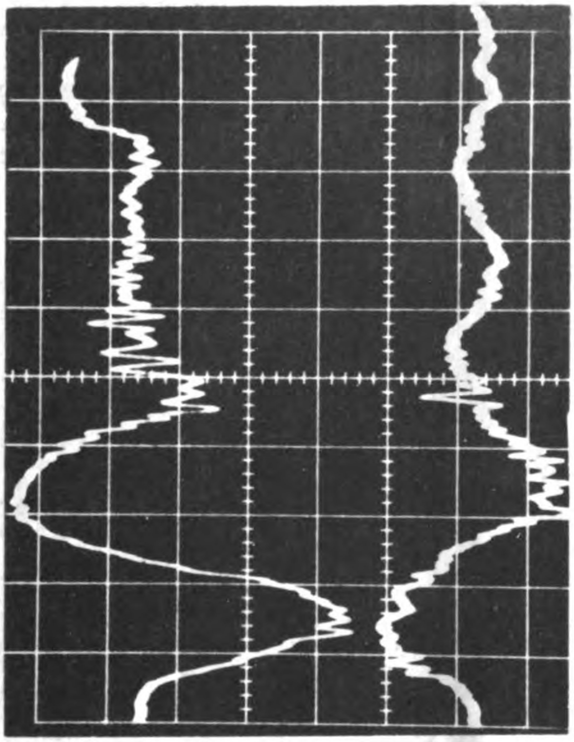

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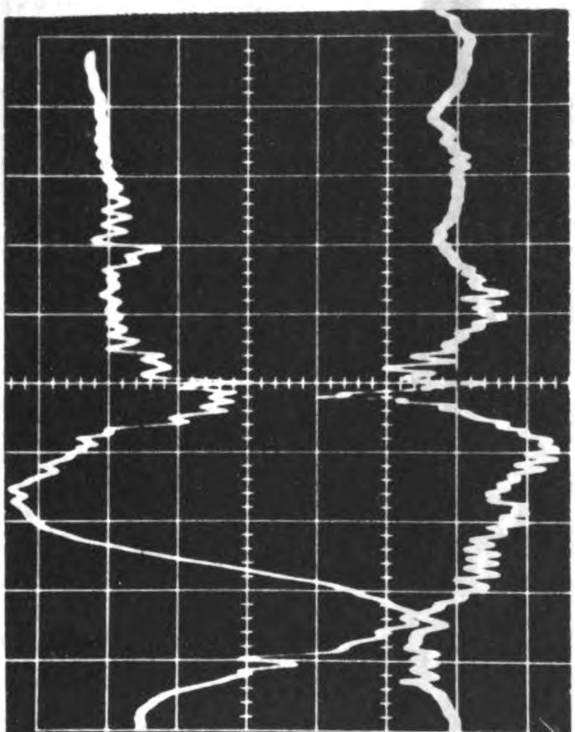

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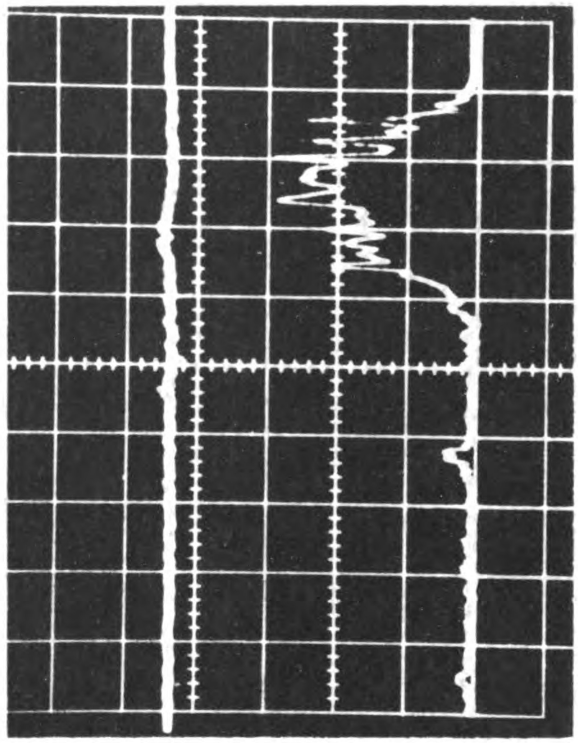

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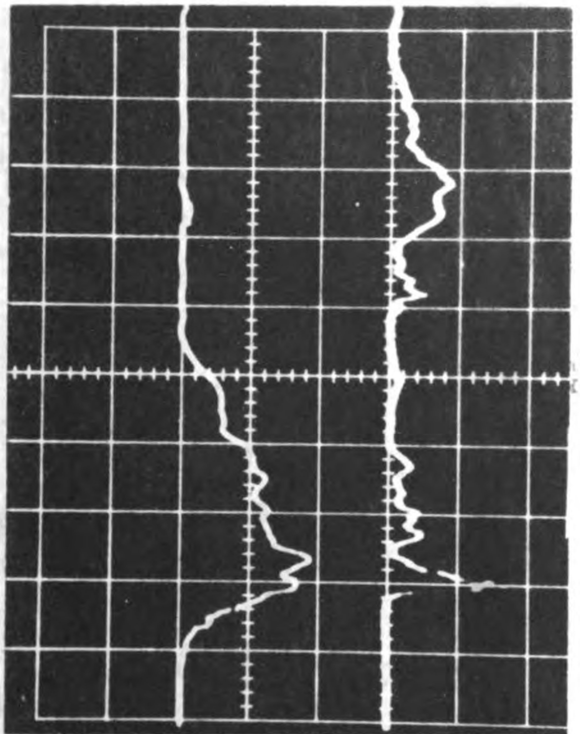

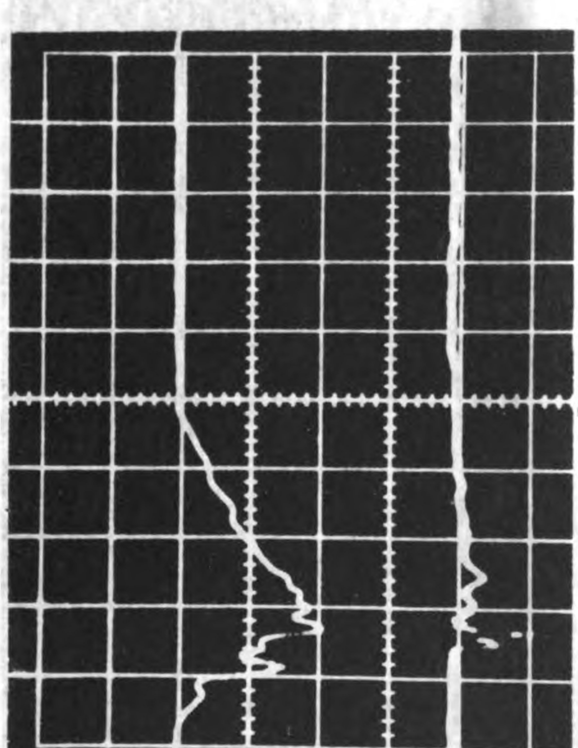

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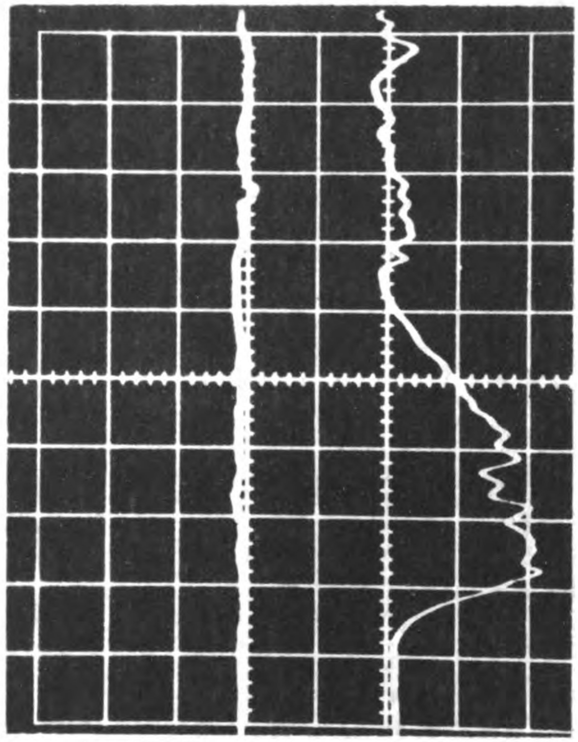

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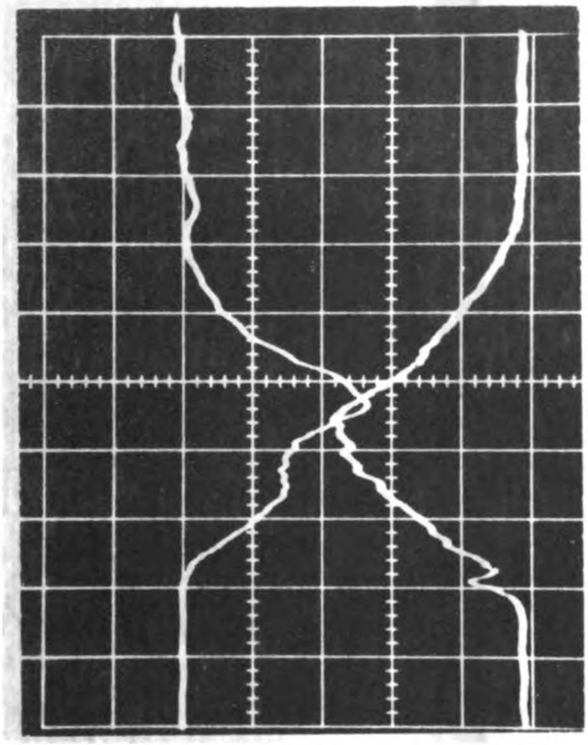

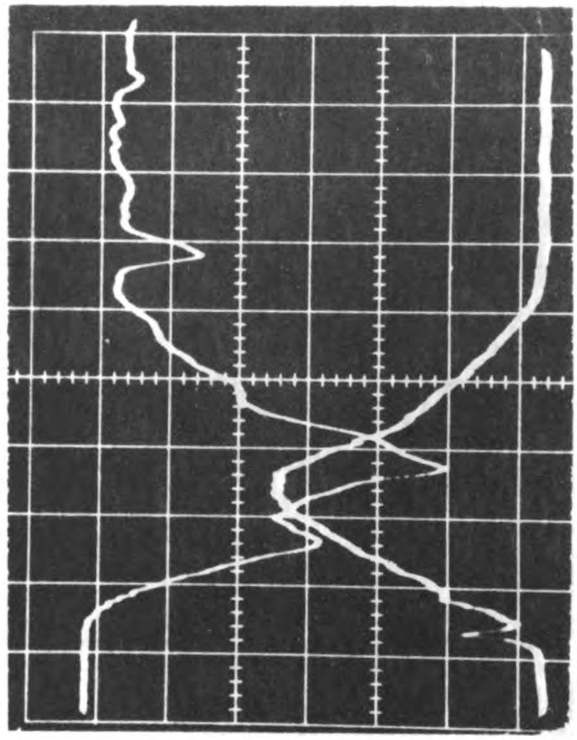

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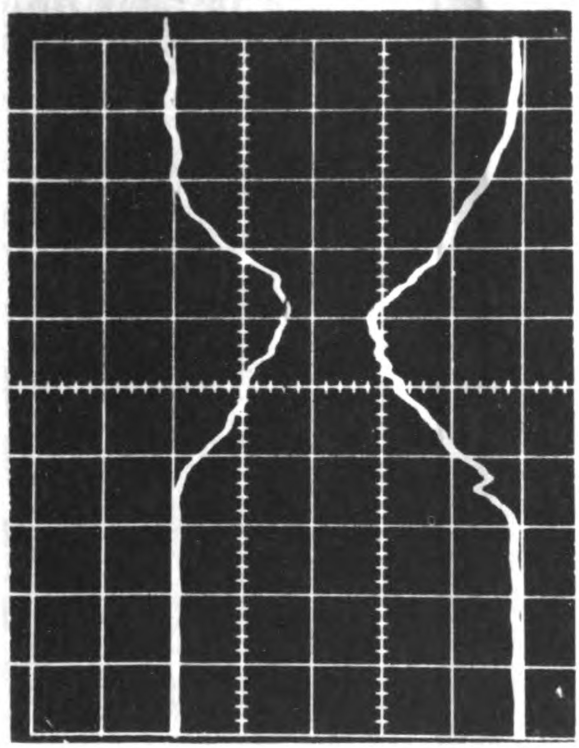

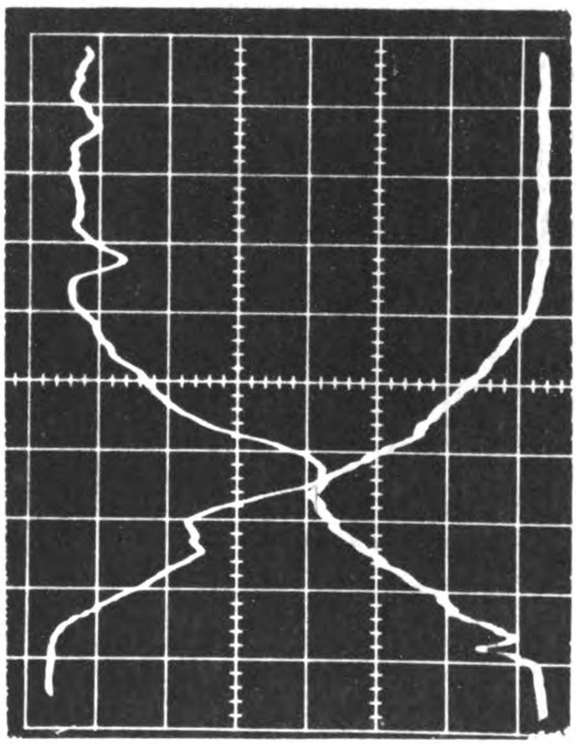

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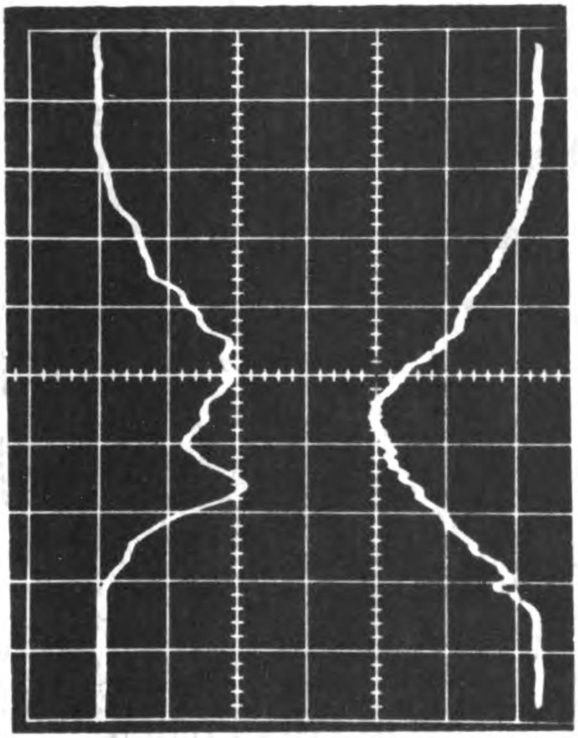

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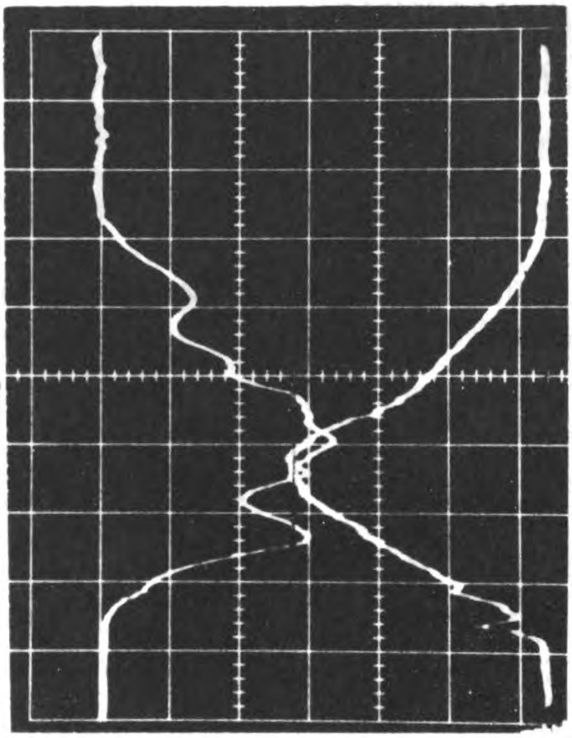

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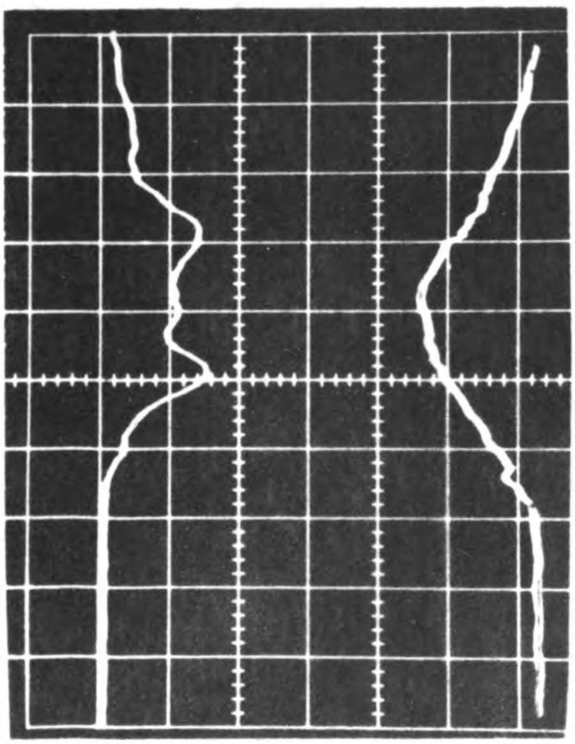

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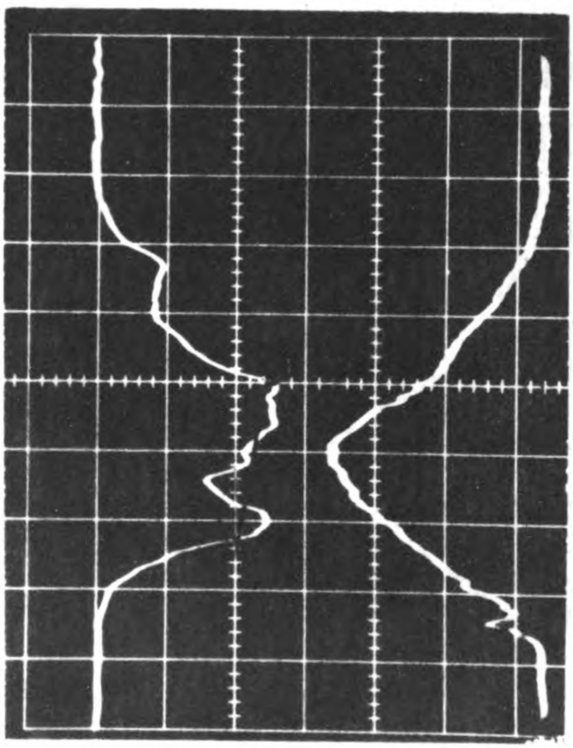

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