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by The Participants

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USATACOM

TACOM

MOBILITY SYSTEMS LABORATORY

U.S. ARMY TANK AUTOMOTIVE COMMAND Warren, Michigan

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November 1975

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FOREWORD

This symposium is part of the continuing information exchange program between the ground mobility research organizations of the Federal Republic of Germany and the United States. The objective of this type of exchange is to reduce or eliminate replication of effort in an area which does not see the expenditure of excessive sums of money in any one country. Primarily this is so because there is precious little interest in high speed, cross-country mobility of any kind and even less so when the vehicle under consideration is much heavier than a light truck. That there is no existing commercial market is witnessed by the fact that there is not a long line of contractors knocking on our respective doors with information and proven products in hand. In point of fact it is often difficult to find the necessary help in the commercial field when constructing prototype equipment requiring specialized manufacturing expertise. Thus, it is desirable to assure that the limited assets available are used to the greatest benefit of all concerned.

These open exchanges also allow the experts of both countries to benefit from the experiences, the thinking and the constructive criticism of their compatriots. That they all go home with refreshed viewpoints has been shown in the many cases where they have saved their respective governments many dollars by avoiding repeating the work of another individual. To the continued free flow of such information and continued advances in ground mobility, this collection of the material presented at the April 1975 Symposium is dedicated.

AMC '75 MOBILITY MODEL

Presented By

P. HALEY
U.S.

THE U. S. ARMY MOBILITY MODEL (AMM-75)

by

M. P. Jurkat, C. J. Nuttall, and P. W. Haley

Abstract

A primary goal of U. S. Army mobility research is the development of validated, objective methodology to support decision processes relative to the design, procurement, and deployment of military vehicles. As a step toward that end, a comprehensive analytical model for evaluating the mobility of ground vehicle systems has been implemented in a large-scale digital computer simulation. The model employs existing vehicle mechanics technology to predict individual facets of system performance and new analysis and programming techniques to account for their interaction.

In 1971, the then state-of-the-art was collected in a first version called AMC-71. This paper briefly describes the second generation of that model, AMM-75, and the modifications that distinguish the two versions.

Introduction

Rational design and selection of Army ground vehicles require objective evaluation of an ever-increasing number of vehicle and vehicle system options. Technology, threat, operational requirements, and cost constraints change with time. Current postures must be reexamined, new options evaluated, and new trade-offs and decisions made. In the single area of combat vehicles, for example, changes in one or another influencing factor might require trade-offs that run the gamut from opting for an air or ground system, through choosing wheels, tracks or air cushions, to designating a new tire.

The Mobility Systems Laboratory of the U. S. Army Tank-Automotive Command (TACOM), the U. S. Army Engineer Waterways Experiment Station (WES), and the U. S. Army Cold Regions Research and Engineering Laboratory are the government laboratories responsible for conducting ground mobility research for the U. S. Army Materiel Command (AMC). In 1971, a unified AMC ground mobility program was implemented that specifically geared the capabilities of all three laboratories to achieve common goals.

As a first step in the unified program, a detailed review was made of existing vehicle mobility technology and of the problems and requirements of the various engineering practitioners associated with the military vehicle life cycle. One basic requirement was identified as common to all practitioners surveyed: the need for an objective analytical procedure for quantitatively assessing the performance of a vehicle in a specified operational environment.

In theory, a single methodology can serve the needs of all major practitioners, provided it relates vehicle performance to basic characteristics of the vehicle-driver-terrain system at appropriate levels of detail.

Three principal categories of potential users of the methodology were identified: the vehicle development community, the vehicle procurement community, and the vehicle user community (Figure 1). The greatest level of detail is needed by the design and development engineer (vehicle design and development community) who is interested in subtle engineering details--for example, wheel geometry, sprung masses, spring rates, track widths, etc.--and their interactions with soil strength, tree stems of various sizes and spacings, approach angles in ditches and streams, etc. At the other end of the spectrum is the strategic planner (user community), who is interested in such highly aggregated characteristics as the average cross-country speed of a given vehicle throughout a specified region--the net result of many interactions of the engineering details with features of the total operational environment. To be responsive to the needs of all three user communities, the methodology must be flexible enough to provide compatible results at many levels and in an appropriate variety of formats.

Interest in a single, unified methodology applicable to the needs of these principal users led to the creation of a cross-country vehicle computer simulation combining the best available knowledge and models of the day. Much of this knowledge was collected in Reference 1. The first realization of the simulation was a series of computer programs known as the AMC-71 mobility model, called AMC-71 for short.[2] This model first became operational in 1971; it was published in 1973. It was conceived as the first generation of a family whose descendants, under the evolutionary pressures of subsequent research and validation testing results, application experiences, and growing user requirements, would be characterized by greater accuracy and applicability. A relatively current status report may be found in Reference 3, after which this presentation is patterned.

The first descendant, to be known as AMM-75, is in the final stages of preparation. Planned for release by the fall of 1975, its major features are highlighted in the description that follows.

Modeling Off-Road Vehicle Mobility

In undertaking mobility modeling, the first question to be answered was the seemingly easy one: What is mobility? The answer had been elusive for many years. Semantic reasons can be traced to the beginnings of mobility research, but there was also a pervasive reluctance to accept the simple fact that even intuitive notions about a vehicle's mobility depend greatly on the conditions under which it is operating. By the mid-1960s, however, a consensus had emerged that the maximum feasible speed-made-good* by a vehicle between two points in a given terrain was a suitable measure of its intrinsic mobility in that situation.

This definition not only identified the engineering measure of mobility, but also its dependence on both terrain and mission. When, at a suitably high resolution, the terrain involved presents the identical set of impediments to vehicle travel throughout its extent, mobility in

* Speed-made-good between two points is the straight-line distance between them divided by total travel time, irrespective of path.

that terrain (ignoring edge effects) is the vehicle's maximum straight-line speed as limited only by those impediments. But when, as is typically the case, the terrain is not so homogeneous, the problem immediately becomes more complex. Maximum speed-made-good then becomes an interactive function of terrain variations, end points specified, and the path selected. (Note that the last two constitute at least part of a detailed mission statement.)

AMC Mobility Model Approach

The AMC mobility model deliberately represents real terrain as a mosaic of terrain units within each of which the terrain is considered sufficiently uniform to permit use of the simple, maximum straight-line speed of the vehicle to define its mobility in, along, or across that terrain unit.

Maximum speed predictions are made for each terrain unit without concern for whether or not distances within the unit are adequate to permit the vehicle to reach the predicted maximum.

This vehicle and terrain-specific speed prediction is the basic output of the model. The model, in addition, generates data that may be used to predict operational vibration levels, mission fuel consumption, etc., and provides diagnostic information as to the factors limiting speed performance in the terrain unit.

The speed and other performance predictions for all terrain units in an area can be incorporated into maps that specify feasible levels of performance that a given vehicle might achieve at all points in the area. At this point, the output is reasonably general and is essentially independent of mission and operational scenario influences.

The basic data constituting the maps must usually be further processed to meet the needs of specific users. These needs vary from relatively simple statistics or indices reflecting overall vehicle compatibility with the terrain, to extensive analyses involving detailed or generalized missions. At present only one output processer is considered a standard part of AMM-75. This post-processer accumulates a statistical picture of maximum feasible speeds in the terrain, and of the terrain-driver-vehicle interactions that account for speed limits or NOGO situations. (AMC-71 includes a path selection model, which chooses the minimum time path from a network of possible paths, based on speeds along the network links predicted by the mobility model. While this model is not a standard part of AMM-75, it can be used with AMM-75 for special studies.)

Overall Structure of AMC Mobility Model

In formulating AMC-71, it was recognized that its ultimate usefulness to decision makers in the vehicle development, procurement, and user communities would depend upon its realism and credibility.[4] These perceived requirements led to several more concrete objectives related to the overall structure of the model. It was determined that the model should be designed to:

a. Allow validation by parts and as a whole.

b. Make a clear distinction between engineering predictions and any whose outcome depends significantly upon human judgment, with the latter kept visible and accessible to the model user.

c. Be updated readily in response to new vehicle and vehicle-terrain technology.

d. Use measured subsystem performance data in place of analytical predictions when and as available and desired.

These objectives, plus the primary goal of supporting vehicle decision making at the several levels, clearly dictated a highly modular structure that could both provide and accept data at the subsystem level, as well as make predictions for the vehicle as a whole. The resulting gross structure of the model is illustrated in Figure 2.

At the heart of the model are three independent computational modules, each comprised of analytical relations derived from laboratory and field research, suitably coupled in the particular type of operation:

a. The areal patch module, which computes the maximum feasible speed for a single vehicle in a single areal terrain patch or terrain unit.

b. The linear feature module, which computes the minimum feasible time for a single vehicle, aided or unaided, to cross a uniform segment of a significant linear terrain feature such as a stream, ditch, or embankment.

c. The on-road module, which computes the maximum feasible speed of a single vehicle traveling along a uniform segment of a road or trail.

These three modules have been and are still able to be used separately or with output superimposed. A new feature of AMM-75 is the ability to simulate travel from terrain unit to terrain unit in the sequence given by the terrain input file. In this mode, known as the traverse mode, sufficient output data can be provided so that the user may calculate acceleration and deceleration times and distances between and across terrain unit boundaries, and thereby determine actual travel time and speed-made-good over a chosen route.

All three modules draw from a common data base that describes quantitatively the vehicle, the driver, and the terrain to be examined in the simulation. The general content of the data base is shown in Table 1.

Model Inputs and Preprocessers

Terrain

For the purposes of the model, each terrain unit is described at any given time by values for a series of 22 mathematically independent terrain factors for an areal unit (including lake and marsh factors), 10 for the cross section of a linear feature to be negotiated, and 9 to quantify a road segment (Tables 2 and 3). General-purpose terrain data also include separate values for several terrain factor values that vary during the year. For example, at present such general data for areal terrain include four values for soil strength (dry, average, wet, and wet-wet seasons) and four seasonal values for recognition distances in vegetated areas. Similar variations in effective ground roughness, resulting from seasonal changes in soil moisture (including freezing) and in the cultivation of farm land, can be envisioned for the future. Further details on the terrain factors used are given in Reference 5.

As discussed earlier, the basic approach to representing a complex terrain is to subdivide it into areal patches, linear feature segments, or road segments, each of which can be considered to be uniform within its bounds. This concept is implemented by dividing the range of each individual terrain factor value into a number of class intervals, based upon considerations of vehicle response sensitivity and practical measurement and mapping resolution problems. A patch or a segment is then defined by the condition that the class interval designator for each factor involved--22 areal, or 10 linear, or 9 road--is the same throughout. A new patch or segment is defined whenever one or more factors fall into a new class interval.

The terrain data base contains, for each uniform patch or segment, a series of numbers specifying the value for each of its factors. A sample of such a listing for areal terrain, and of the terrain factor complex map to which it relates is shown in Figure 3.* As suggested by Table 2, the terrain data base is in fact different for the three types of terrain (areal, linear, and on-road).

Before being used in the three computational modules, the basic terrain data are passed through a terrain data preprocesser. This preprocesser does three things:

a. Converts as necessary all data from the units in which they are stored to inches, pounds, seconds, and radians, which are used throughout the subsequent performance calculations.

b. Selects prestored soil strengths and visibility distances according to run specifications, which are supplied as part of the scenario data (see below).

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* In the example, the area within any areal terrain patch is represented by an integral number of rectangular cells, 127x106 m. This representation allows results to be output on a normal computer printer in the form of 1:25,000 maps.

c. Calculates from the terrain measurements in the basic terrain data a small number of mathematically dependent terrain variables used in the computational modules.

Vehicle

The vehicle is specified in the vehicle data base in terms of its basic geometric, inertial, and mechanical characteristics. The complete vehicle characterization as used by the performance computation modules includes measures of dynamic response to ground roughness and obstacle impact, and the clearance and traction requirements of the vehicle while it is negotiating a parametric series of discrete obstacles.* The model structure permits use at these points of appropriate data derived either from experiments or from supporting stand-alone simulations used as preprocessers. One supporting two-dimensional ride and obstacle crossing dynamics module for obtaining requisite dynamics responses[6] and a second supporting module for computing obstacle crossing traction requirements and interferences[7] are available as elements of the AMM-75 model. Both derive some required information from the basic vehicle data base, and both, when used, constitute stand-alone vehicle data preprocessers.

There is also an integral vehicle data preprocesser which, like the terrain data preprocesser, has three functions:

a. To convert vehicle input data to uniform inches, pounds, seconds, and radians.

b. To calculate, from the input data, controlling soil performance parameters and other simpler dependent vehicle variables subsequently used by the computational modules, but usually not readily measured on a vehicle or available in its engineering specifications.

c. To compute the basic steady-state traction versus speed characteristics of the vehicle power train, from engine and power train characteristics.

As in the case of dynamics responses and obstacle capabilities, the last item, the steady-state tractive force-speed relation, may be input directly from proving ground data, when available and desired.

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* A simpler obstacle-crossing model was integral to the AMC-71 areal module.

Details of the vehicle input data required for operation of the areal, linear feature, on-road, and obstacle negotiation modules are given in Table 4. The two-dimensional ride and obstacle impact dynamics simulation requires special, detailed spring and damping data and mass properties not included in Table 4, but indicated in Reference 6.

Driver

The driver attributes used in the model characterize the driver in terms of his limiting tolerance to shock and vibration and his ability to perceive and react to visual stimuli affecting his behavior as a vehicle controller. While these attributes are identified in Figure 2 and Table 1 as part of the data base, in AMC-71 they are built into the program. AMM-75 provides for their specific identification and user control so that the effects of various levels of driver motivation, associated with combat or resupply missions, for example, can be considered.

Scenario

Several optional features are available to the user of AMM-75 (weather, presumed driver motivation, operational variations in tire inflation) which allow him to match the model predictions to features or assumptions of the full operational scenario for which he requires the predictions. Model instructions which select and control these options are referred to as scenario inputs.

The scenario options for AMC-71 are limited to the specification of season which, when seasonal differences in soil strength constitute a part of the terrain data, allows selection of the soil strength according to the variations in soil moisture with seasonal rainfall. AMM-75 expands the scenario options to include specifications of:

a. Weather, which affects soil slipperiness and driving visibility, (including dry snow over frozen ground and associated conditions).

b. Several levels of operational influences on driver tolerances to ride vibrations and shock, and on driver strategy in negotiating vegetation and using brakes.

c. Reasonable play of tire pressure variations to suit the mode of operation--on-road, cross-country, and in sand.

In addition, the model can now be used under a simple scenario command to make predictions in relation to a traverse (given directional terrain data specifically along the traverse) as well as to make omnidirectional predictions for an area.

Stand-Alone Simulation Modules

As indicated above, the model is implemented by a series of independent modules. The terrain and vehicle preprocessers, already described, form two of these. Two further major stand-alone simulation modules will now be briefed.

Dynamics module

The areal module examines as possible vehicle speed limits in a given terrain situation two limits which are functions of vehicle dynamic responses: speed as limited by the driver's tolerance to his vibrational environment when the vehicle is operating over continuously rough ground, and speed as limited by the driver's tolerance to impact received while the vehicle is crossing discrete obstacles. It is assumed that the driver will adjust his speed to ensure that his tolerance levels will not be exceeded.

The ride dynamics module of AMM-75[6] computes accelerations and motions at the driver's station (and other locations, if desired) while the vehicle is operating at any given speed over any given terrain profile. The profile may be continuously, randomly rough, may consist solely of a single discrete obstacle, or may be anything between. From the computed motions, associated with driver modeling and specified tolerance criteria, simple relations are developed for a given vehicle between relevant terrain measurements and maximum tolerable speed. The terrain measurement to which ride speed is related is the root mean square (rms) elevation of the ground profile (with terrain slopes and long-wavelength components removed). The terrain descriptors for obstacles are obstacle height and obstacle spacing.

The terrain parameters involved, rms elevation and obstacle height and spacing, are factors quantified in each patch description, and rms elevation is specified for each road segment. Preprocessing of the vehicle data in the ride dynamics module provides an expedient means of predicting dynamics-based speed in the patch and road segment modules via a simple, rapid table-lookup process.

The currently implemented ride dyanmics module is a digital simulation that treats vehicle motions in the center-line plane only (two dimensions). It is a generalized model that will handle any rigid-frame vehicle on tracks and/or tires, with any suspension. Tires are modeled using a segmented wheel representation,[8] and a variation of this representation is used to introduce first-order coupling of the road wheels on a tracked vehicle by its tracks. The simulation requires detailed vehicle data that are not used in the speed prediction modules and not shown in Table 4. The complete listing of vehicle input data used is given in Reference 6.

Driver model and tolerance criteria. It has been shown empirically that, in the continuous roughness situation, driver tolerance is a function of the vibrational power being absorbed by the body.[9] The same work showed that the tolerance limit for representative young American males is approximately 6 watts of continuously absorbed power, and the research resulted in a relatively simple model for power absorption by the body. The body power absorption model, based upon shaping filters applied to the decomposed acceleration spectrum at the driver's station, is an integral part of the AMM-75 two-dimensional dynamics simulation.

In AMC-71, only the 6 watt criterion was used to determine a given vehicle's speed as limited by rms roughness. More recent measurements in the field have shown that with sufficient motivation young military drivers will tolerate up to 15 watts for periods of many minutes. Accordingly, AMM-75 will accept as vehicle data a series of ride speed versus rms elevation relations, each corresponding to a different absorbed power level, and will use these to select ride-speed limits according to the operationally related level called for by the scenario. The ride dynamics module will, of course, produce the required additional data, but some increased running time is involved.

The criterion limiting the speed of a vehicle crossing a single discrete obstacle, or a series of closely, regularly spaced obstacles, is a peak acceleration at the driver's seat of 2.5 g passing a 30-Hz filter. Data relating the 2.5-g speed limit to obstacle height and spacing can be developed in the ride dynamics module by inputting appropriate profiles.

AMM-75 requires two obstacle impact relations: the first, speed versus obstacle height for a single obstacle (spacing very great); and the second, speed versus regular obstacle spacing for that single obstacle height (from the single obstacle relation) which limits vehicle speed to a maximum of 15 mph (24 kpm). For obstacles spaced at greater than two vehicle lengths, the single-obstacle speed versus obstacle height relation is used. For closer spacings, the least speed allowable by either relation is selected.

Obstacle-crossing module

A new module is provided in AMM-75 to determine interferences and traction requirements when vehicles are crossing the kind of minor ditches and mounds characterized as part of the areal terrain.[7] It is used as a stand-alone preprocessor module to the areal module of AMM-75.

The new obstacle-crossing module simulates the inclination and position, interferences, and traction requirements of a two-dimensional (center-line plane) vehicle crossing a single obstacle of any profile configuration or any arbitrary sequence of such obstacles. The module determines a series of static equilibrium positions of the vehicle as it progresses across the obstacle profile. Extent of interference is determined by comparison of the obstacle profile and the displaced vehicle bottom profile. Traction demand at each position is determined by the forces on driven running gear elements, tangential to the obstacle surface, required to maintain the vehicle's static position. Pitch compliance of suspension elements and of frame articulation (as at pitch joints, trailer hitches, etc) is accounted for.

In AMC-71, the determination of vehicle obstacle negotiation in an areal terrain unit was performed repeatedly within the areal module for each terrain unit as it occurred. This proved time-consuming and was unnecessary for two terrain units with the same obstacles. The AMC-71 obstacle routine made simplified tests for interference and traction requirements at a limited number of critical stages in the process of obstacle regotiation (for instance, front-end interference approach angle at initial obstacle contact, belly interference across the top of a mound, and traction required on the upslope side.) The routine assumed a rigid frame vehicle and a 2-axle or rigid track running gear with no suspension compliance. The AMC-71 modeling approach requires that the designer of the routine foresee all possible cases of interference for all types of vehicles. When this critical check technique is to be applied to suspended multi-axle vehicles, or to pitch-articulated vehicles, the number of tests to be made becomes very large and too much reliance is placed on the model designer's intuition. The chance of mistakes is great.

In response to these objections and with the desire to allow AMM-75 to treat properly a greater variety of more realistic vehicle designs, including articulated vehicles, softly-sprung vehicles, and vehicles with large variations in weight distribution from one running gear unit to another, the more detailed equilibrium calculation approach was adopted for interference and traction. In this technique, the vehicle, mathematically, is moved across the obstacle in fixed steps. At each step the vehicle's equilibrium elevation and attitude are calculated by minimizing the potential energy of height and suspension. Currently, the module is operational for wheeled vehicles on obstacles for which relatively small pitch angles can be assumed. This allows each equilibrium position to be found by the solution of linear equations.

In order to assure that all possible locations where interference can occur are at least approximated, the step size across the obstacle must be small compared to the size of the obstacle and vehicle. This forces the new model to consume considerable time to check each obstacle-vehicle combination. To minimize total computing time, the obstacle module is run out of the main stream of the AMM-75 processing modules. This is feasible because in AMM-75, as in AMC-71, obstacle cross sections characterized as part of the areal terrain (as distinct from major obstacles which are treated separately as linear features) are considered symmetrical and are defined by only three parameters: height (or depth), approach angle, and width.

The new model, run as a preprocesser module, produces a table of minimum clearances (or maximum interferences) and average and maximum force required to cross a representative sample of obstacles defined by combinations of obstacle dimensions varied over the ranges appropriate for features included in the areal terrain description. This is done only once for each vehicle. Included in the AMM-75 areal module is a three-dimensional linear interpolation routine which, for any given set of obstacle parameters, approximates from the derived table the corresponding vehicle clearance (or interference) and associated traction requirements. Obviously, the more entries there are in the table, the more precise will be the determination.

Main Computational Modules

The highly iterative computations required to predict vehicle performance in each of the many terrain units needed to describe even limited geographic areas are carried out in the three main computational modules. Each of these involve only direct arithmetic algorithms which are rapidly processed in modern computers. In AMM-75, even the integrations required to compute acceleration and deceleration between obstacles within an areal patch are expressed in closed, algebraic form.

Terrain input data include a flag, which signifies to the model whether the data describe an areal patch, a linear feature segment, or a road segment. This flag calls up the appropriate computational module.

Areal terrain unit module

This module calculates the maximum speed a vehicle could achieve and maintain while crossing an areal terrain unit. The speed is limited by one or a combination of the following factors:

a. Traction available to overcome the combined resistances of soil, slope, obstacles, and vegetation.

b. Driver discomfort in negotiating rough terrain (ride comfort) and his tolerance to vegetation* and obstacle impacts.

c. Driver reluctance to proceed faster than the speed at which the vehicle could decelerate to a stop within the, possibly limited, visibility distance prevailing in the areal unit (braking-visility limit).

d. Maneuvering to avoid trees and/or obstacles.

e. Acceleration and deceleration between obstacles if they are to be overridden.

Figure 4 shows a general flow chart of how the calculations of the areal module in AMM-75 are organized.

After determination of some vehicle and terrain-dependent factors used repetitively in the patch computation (1),** the module is entered with the relation between vehicle steady-state speed and theoretical tractive force and with the minimum soil strength that the vehicle requires to maintain headway on level, weak soils. These data are provided by the vehicle data preprocesser. Soil and slope resistances (2) and braking force limits (4) are computed, and the basic tractive force-speed relation is modified to account for soil-limited traction, soil and slope resistances, and resulting tire or track slip. Forces required to override prevailing tree stems are calculated for eight cases (3): first, overriding only the smallest stems, then overriding the next largest class of stems as well, etc., until in the eighth case all stems are being overridden.

Stem override resistances are combined with the modified tractive force-speed relation to predict nine speeds as limited by basic resistances (5). (The ninth speed corresponds to avoiding all tree stems.)

Maximum braking force and recognition distance are combined to compute a visibility-limited speed (6). Resistance and visibility-limited speeds are compared to the speed limited by tire loading (7), if

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* Checked as part of the areal terrain unit module.

** Numbers in parenthesis correspond to numbers in Figure 4.

applicable, and to the speed limit imposed by driver tolerance to vehicle motions resulting from ground roughness (8). The least of these for each tree override-and-avoid option becomes the maximum speed possible between obstacles by that option, except for degradation due to maneuvering (9).

Obstacle avoidance and/or the tree avoidance implied by limited stem override requires the vehicle to maneuver (or may be impossible). Using speed reduction factors (derived in 1) associated with avoiding all obstacles (if possible) and avoiding the appropriate classes of tree stems, a series of nine possible speeds (including zero, or NOGO) is computed (10).

A similar set of nine speed predictions is made for the vehicle maneuvering to avoid tree stems only (10). These are further modified by several obstacle crossing considerations.

Possible NOGO interference between the vehicle and the obstacle is checked (12). If obstacle crossing proves to be NOGO, all associated vegetation override and avoid options are also NOGO. If there are no critical interferences, the increase in traction required to negotiate the obstacle is determined (12).

Next, obstacle approach speed and the speed at which the vehicle will depart the obstacle, as a result of the momentarily added resistance encountered, are computed (13). Obstacle approach speed is taken as the lesser of the speed between obstacles, reduced for maneuver required by each stem override and avoid option, and the speed limited by the driver to control his crossing impact (11). Speeds off the obstacle are computed on the basis solely of the soil- and slope-modified tractive force-speed relation (22), i.e. before the tractive force speed relation is modified to account for vegetation override forces, the traction increment required for obstacle negotiation, or any kinetic energy available as a result of the associated obstacle approach speed (13).

Final average speed in the patch for each of the nine tree stem override and avoid options, while the vehicle is overriding patch obstacles, is computed from the speed profile resulting, in general, from considering the vehicle to accelerate from the assigned speed off the obstacle to the allowable speed between obstacles (or to a lesser speed if obstacle spacing is insufficient), to brake to the allowable obstacle approach speed, and to cross the obstacle per se at the computed crossing speed.

Following a final check to ensure that traction and kinetic energy are sufficient for single-tree overrides called for (and possible re-setting of speeds for some options to NOGO) a single maximum in-patch speed (for the direction of travel being considered relative to the in-unit slope) is selected from among the nine available values associated with obstacle avoidance and the nine for the obstacle-override cases. If all 18 options are NOGO, the patch is NOGO for the direction of travel. If several speeds are given, selection is made by one of two logics according to scenario input instructions.

In AMC-71 the driver was assumed to be both omniscient and somewhat mad. Accordingly, the maximum speed possible by any of the 18 strategies was selected as the final speed prediction for the terrain unit (and slope direction). Field tests have shown, however, that a real driver does not often behave in this ideal manner when driving among trees. Rather, he will take heroic measures to reach some reasonable minimum speed, but will not continue such efforts when those measures involve knocking down trees that he judges it imprudent to attack, even though by doing so he could go still faster. In AMM-75, either assignment of maximum speed may be made: the absolute maximum which addresses the vehicle's ultimate potential, or a lesser value which in effect models actual driver behavior more closely.

In AMM-75, if the scenario data specify a traverse prediction, the in-unit speed and other predictions are complete at this point, and the model stores those results specified by the user and goes on to consider the next terrain unit (or next vehicle, condition, etc). When a full areal prediction is called for, the entire computation is repeated three times: once for the vehicle operating up the in-unit slope, once across the slope, and once down the slope. Desired data are stored from each such run prior to the next, and at the conclusion of the third run, the three speeds are averaged. Averaging is done on the assumption that one-third of the distance* will be traveled in each direction, resulting in an omnidirectional mean.

The areal module of AMM-75, as compared with that of AMC-71, is significantly improved in several other respects.

a. AMC-71 assumes all running gears of a vehicle to be powered, geometrically identical, and equally loaded. AMM-75 can simulate vehicles and vehicle combinations having various configurations of powered, braked, and towed wheels and tracks, variously loaded. This is done by calculating the tractive effort and motion resistance of the vehicle running gear one element at a time and summing for the whole configuration. A separate value of excess vehicle cone index (VCI) is calculated for each running gear and then relations presented in References 1 and 10 are used to find traction and resistance coefficients for that running gear. The load (possibly modified for slope or buoyancy as specified by the terrain unit) and the running gear VCI's are then used to calculate overall maximum tractive effort and resistance. This allows the modeling of vehicles such as half-tracks; towed, powered, or braked trailers; articulated vehicles; and vehicles with gross variations in load distributions and running-gear geometry.

b. AMM-75 contains equations that allow simulations of travel across slippery soils, muskeg, and shallow dry snow in addition to the fine- and coarse-grained soils covered in AMC-71. Slipperiness effects are included whenever the scenario calls for rain or standing water and soil surfaces are flooded or locally very wet. Separate relations are used for fat clay soils, which are impervious to water, and for other more

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* $V_{av} = \dfrac{3}{\dfrac{1}{V_{up}} + \dfrac{1}{V_{across}} + \dfrac{1}{V_{down}}}$, i.e. mathematically the harmonic average of the three speeds.

pervious fine-grained soils. Where soil is relatively soft, slipperiness is not a factor. When the surface is very hard, the slipperiness factor becomes constant, indicating a "skating" condition. Muskeg performance relations included are those published in Reference 11. Shallow snow is defined as snow covering frozen ground at a depth less than the characteristic length of the tire or less than one third of the characteristic length of the track. To calculate the drawbar-pull and resistance coefficients for shallow snow, the model uses snow effective cohesion, internal friction, and specific weight. Traction is calculated by means of the familiar Coulomb relation, and motion resistance is obtained by means of two empirical functions (based upon limited tests in shallow dry snow over the years 1955 through 1972), one for tracked vehicles, one for wheeled. In both relations the fundamental prediction term involves the ratio of nominal running gear contact length to snow depth after compression of the snow to a specific gravity of 0.4. Drawbar-pull or net traction available is taken as the excess of traction over motion resistance.

c. The net tractive performance of wheeled vehicles in soils and dry snow is significantly influenced by tire inflation pressure, load and resulting tire deflection, and to a lesser extent by the fitting of slip-limiting or locking differentials. The effects of these factors are modeled in the revised soil submodel in AMM-75. A new speed limit is also introduced to ensure that the speed reduction which must accompany operations at reduced tire inflations is accounted for. Separate inflation versus speed-limit relations are used for bias-ply and radial tire construction.

d. In AMC-71 resistance encountered during obstacle crossing in an area is averaged over the entire patch area. In AMM-75, the full value of this resistance is introduced at the obstacles only, giving rise to possible deceleration and acceleration at obstacles in the same general manner as does the driver's slowing to reduce obstacle crossing impact to the tolerable level.

e. The relation between vehicle speed and tractive effort available at that speed is used throughout the module. In AMC-71 this relation is kept as a table, which necessitates frequent searches and interpolations. In AMM-75 the tractive force-speed is modeled as a series of quadratic equations, one for each gear or section of a gear range. The vehicle preprocesser initially fits the quadratics to the theoretical rimpull power train curve. The areal module then modifies the quadratics for traction limit, for slope, and for running-gear longitudinal slip. The availability of the tractive effort in quadratic form allows closed-form integration in the calculation of acceleration times and distances. This provides for a more precise and rapid calculation of average speed as a result of acceleration and deceleration between obstacles than was available in AMC-71.

f. The effects of rotating masses (gears, wheels, tracks, etc.), which must be rotationally accelerated as the vehicle pass per se is accelerated linearly, have been incorporated in AMM-75 computations of vehicle acceleration and deceleration performance. This is done by using values for the equivalent mass factor (apparent mass/actual mass) for the vehicle in each gear, in the vehicle power train data.

g. In AMM-75 final obstacle and vegetation-override GO/NOGO checks are made at the end of the speed computations for a terrain unit where the best estimate of approach speeds is available. This permits more rational assessment of kinetic energy availability to overcome any traction deficits. In AMC-71 these checks are made with basic soil- and slope-limited speeds, which are often reduced later in the computations by further speed-limiting considerations.

Linear feature crossing module

In context of AMC-71 and AMM-75, a linear feature is a distinct terrain element such as a stream, man-made drainge ditch, canal, escarpment, or a highway or railroad embankment, which is a potential barrier to vehicle movement normal to its characteristic length. By and large, most such features are represented by lines on a good 1:50,000 topographic map of an area.

Vehicle performance in crossing linear features requires somewhat different modeling from that used to deal with areal terrain because a vehicle does not necessarily negotiate a linear feature in the same manner that it does areal terrain. While crossing of smaller features is similar to the crossing of obstacles characterized as features of an areal patch, the linear feature obstacles themselves will generally be more severe. A model of the physical encounter must be able to deal properly with large changes in vehicle attitude, with load changes arising from this and from buoyancy effects, with complex obstacle cross sections, and complex changes in soil composition and strength across the section.

All of the above considerations apply also to modeling the crossing of larger linear features, plus the additional fact that complete crossing of a large feature need not be done on a single cross section. Successful negotiation often requires that the vehicle enter the feature at one point along its length, and remain "in" it (if it is a stream) or "on" it (if it is a road embankment) for some distance until a suitable exit point is found. Because linear features are frequently severe barriers, realistic predictions of crossing times must therefore include an assessment of alternatives to headlong crossing at a given site. These alternatives should include possible search distances to find suitable exit sites, and even to find a bridge or other gap in the barrier.

The linear feature crossing module[12] of AMM-75 is structured to address all of these special problems, albeit some on as yet relatively simple bases. The general flow of computations is shown in Figure 5.

The basic output of the module is a GO or NOGO determination for a given vehicle crossing a linear feature at a single, fully specified cross section characteristic of cross sections throughout some length of the feature. Such a nominally uniform length of the feature is called a linear feature segment, or a linear terrain unit. If the vehicle can cross, crossing time from bank top to bank top is computed. If the situation is NOGO, reasons are shown and an index of relative crossing difficulty is computed which can be used in a suitable output processer to assess delay times or to call for use of alternative crossing sites according to the user's full scenario. When area-wide predictions are required by the user (specified at run time), crossing is checked in both directions. For a traverse, crossing is checked only in the direction required.

Regardless of whether the cross section is GO or NOGO, data to permit consideration of alternative crossing sites are also developed for each linear feature segment. By consulting statistics for the area (the natural river meanders which depend on gross topography, and bridge spacings) and/or speed predictions for the area made by the areal terrain modules, two mean distances and associated travel times in the areal terrain (along the feature, but not "in" or "on" it) are assigned. One distance-time is given to the nearest suitable bridge (if applicable), and the other distance-time is to the nearest crossable section. Where crossing NOGO is the result of exiting traction and/or vehicle-bank interference problems, the nearest crossable section is characterized by an exit "window".

For a linear terrain unit wide enough and otherwise suitable for vehicle travel along its length, a second mean distance and travel time to the nearest exit window are also determined, based upon predicted vehicle speed "on" or "in" the linear segment.

The outputs, GO/NOGO, reasons for NOGO, index of crossing difficulty and times to cross or to find other crossing sites, are returned to the user with no further analysis. How they are used to calculate traverse times or average speeds depends on the total operational scenario of the user. The model does not postulate a complete scenario.

Basic two-way GO/NOGO determinations may be coded simply and overlaid on an areal terrain speed map to provide a more complete picture of the cross-country movement problem presented to a vehicle by a given geographic locale under given weather conditions. The complete output data are suitable for statistical aggregation to show the compatibility of a vehicle with the terrain and conditions, or for the support of vehicle mobility evaluations based upon various mission profiles and presumed levels of support.

Road module

The road module calculates the maximum speed a vehicle can be expected to attain along a nominally uniform stretch of road, termed a road unit. Travel on super highways, primary and secondary roads, and trails is distinguished by specifying a road type and a surface condition factor. From these, values of tractive and rolling resistance coefficients for wheeled and tracked vehicles on surfaced roads are determined by a table look-up. For trails, surface condition is specified in terms of cone index (CI) or rating cone index (RCI). Traction, motion resistance, and slip are computed using the soil submodel of the areal module, with scenario weather factors used in the same way as in making off-road predictions.

Relations for computing vehicle performance on smooth, hard pavements are taken from the literature.[13, 14]

The structure of the road module, while much simpler, parallels that of the areal module. Separate speeds are computed as limited by available traction and countervailing resistances (rolling, aerodynamic, grade, and curvature), by ride dynamics (absorbed power), by visibility and braking, by tire load, inflation and construction, and by road curvature per se (a feature not directly considered in the areal module). The least of these five speeds is assigned as the maximum for the road unit (for the assumed direction relative to the specified grade).

The basic curvature speed limits are derived from AASHO experience data for the four classes of roads[15] under dry conditions and are not vehicle dependent. These are appropriately reduced for reduced traction conditions, and vehicle dependent checks are made for tipping or sliding while the vehicle is in the curve.

At the end of a computation, data required by the user are stored. If the model is run in the traverse mode, the model returns to compute values for the next unit; if in the areal mode, it automatically computes performance for both the up-grade and down-grade situations and at the conclusion computes the bidirectional (harmonic) average speed.

Scenario options are similar to those for the areal module.

Output processing

At the conclusion of each computation of vehicle speed in a single areal terrain or road unit, or time to cross a linear segment, a list of up to 600 computed values is deliberately kept temporarily available in the derived data base. Included are all intermediate computed speeds and forces, descriptors of the power train curve as modified by soil strength and resulting slip, and numerous flags indicating special circumstances. Those values (and only those values) desired by the user for further processing, specified by him prior to a run, are stored in a user-designated file before a new terrain unit is considered.

Data saved for further output processing may range from single, final speed predictions, through information needed to diagnose vehicle-terrain compatibility, to figures needed for fuel consumption calculations or to introduce into traverse speed predictions the effects of acceleration and deceleration across terrain unit boundaries.

The basic in-unit speed predictions for a vehicle are the most fundamental output of the model. When these predictions are made for all areal terrain units in a given geographic area, they may be aggregated to calculate various average speeds in the terrain by weighting in-unit speeds according to the relative areal occupancy of associated terrain units or to the relative operational importance of the areas, for example.

The most straightforward and general portrayal of the basic speed results is a mobility map (Figure 6), which indicates the speeds of which the vehicle is capable (including zero, the NOGO condition) throughout the area under consideration. The sample map displays speeds in areal terrain patches only. Linear feature GO/NOGO characteristics can be superimposed to show where these constitute barriers, and a reasonably coded on-road speed map can also be overlaid. The mobility map is a suitable format for presentation of mobility data for many purposes--for example, as input to war gaming or other effectiveness analysis, or for operational planning. It is not directly suitable for applications of a parametric nature, such as assigning quantitative ratings to vehicle candidates for a given mission.

The development of a definitive parametric description of a vehicle's mobility is a task that has challenged vehicle researchers for many years. To date, no generally accepted definition has been forthcoming. However, substantial progess of a conceptual nature has been made during the past few years. The development of the AMM mobility model, which provides a mechanism to integrate the effects of diverse mobility impediments in accordance with their occurrence in the mission environment, constitutes a substantial contribution to this progress.

Because of the absence of a generally accepted parametric mobility description and the widely varying requirements and viewpoints of prospective model users, only one general-purpose output processer is considered to be a standard part of AMM-75. This routine provides a number of useful statistical interpretations of basic model output data for an area. Its principal product is a mobility profile (see Figure 7), which conveys a complete statistical description of a vehicle's mobility performance in all aspects save spatial distribution. The profile indicates the average speed the vehicle can sustain, as a function of the percent of the total area under consideration which it is able to avoid, assuming it avoids those areas posing the greatest impediment to its motion. For example, the intercept V_{90} = 13.5 mph at point A in Figure 7 denotes that the subject vehicle can average 13.5 mph (21.8 km/hr) in the area considered provided it can avoid the most difficult 10 percent of the terrain.

In addition to the mobility profile, the mobility statistics analysis also provides a set of diagnostic outputs to identify the specific mobility impediments limiting vehicle performance in each terrain unit. These diagnostic outputs in their simplest form can be usefully portrayed in histogram form, as in Figure 8, to provide a vivid depiction of the relative significance of the various primary impediments for the particular terrain-vehicle combination considered. The results presented in Figure 8, for example, indicate that maneuvering among obstacles (factor 8) and crossing obstacles (factor 10) are the dominant performance-limiting factors in the situation illustrated. From a design viewpoint, this finding suggests that improving the vehicle's suspension to reduce accelerations during obstacles crossing and increasing its power and hence its acceleration capability would produce an improvement in overall performance. On the other hand, had factors 6 and 9 been the dominant speed limiters, increased vehicle power only would have been suggested.

Although the interface has not been specifically developed, AMM-75 will also readily support the best-route selection model that is a part of AMC-71, should this be required. The route selection model determines the route a vehicle would take to minimize travel time across a terrain area between two given points.

To determine the route, the terrain area is overlaid with a rectangular grid, and the vehicle is constrained to travel only along straight lines between grid coordinates. Travel times along the allowable paths are predicted by AMC-71 (or AMM-75). The particular combination of such line segments over which the vehicle can negotiate the area in the shortest time is determined by dynamic programming techniques.[16-18] No claim is made that this mathematically defined least-time route is related deterministically to the route that a particular driver would select under operational conditions. It is hypothesized, however, that speed values thus computed for a specific vehicle between a number of random point pairs within an area represent a meaningful quantitative measure of the vehicle's mobility in the terrain under consideration.

Other special-purpose output processers are already operational: to compute traverse speed (including acceleration and deceleration across terrain unit boundries), to compute fuel consumption, and to produce speed maps on a high-speed computer printer, for example. In each case, the basic data developed in AMM-75 are essential, and the implementation relatively straightforward as computer programs go.

AMM-75, per se, is considered actually to end with basic performance predictions. These have been the crucial problem. Application routines, while interesting and often challenging, are best left to the user to tailor to his exact requirements of the moment.

Applications of Mobility Model

Intelligent application of the AMM-75 mobility model can contribute to every phase of the vehicle development process. The model can be particularly useful for:

a. Establishing mobility criteria to ensure a desired level of performance in a specified geographic area.

b. Determining and comparing the expected performance of various vehicle concepts in specified terrains.

c. Studying the effect of specific design changes on cross-country performance.

During the past two years to date, AMC-71 has been used with appropriate output analyses to develop terrain-specific mobility evaluations of a broad range of military vehicles in five principal geographic locales: two in temperate climates, two in dry desert areas, and one in a subtropical area largely in rice agriculture. These evaluations have affected decisions concerning the entire Army wheeled vehicle fleet and its high-mobility tactical truck components, the design of new main battle tanks, and the direction of self-propelled artillery and future Army scout vehicle developments.

These practical applications and the attendant opportunities to meet the vehicle user and his problems in real life and in real time, have been found useful in accelerating model development and validation. The most recently completed application, to the examination of high-mobility vehicles within actual full operational scenarios, for example, involved appropriately characterizing terrain in large new areas and major extensions in model ouput processing. Under these pressures, rapid, new computerized digital terrain mapping methods were implemented,[5] along with compatible output routines that make combined on- and off-road traverse performance predictions directly from relatively simple map inputs.

Model Running Time

The AMC-71 mobility model is currently operational at WES, TACOM, and Stevens Institute of Technology. AMM-75 is being implemented now. AMC-71 has also been made available to a number of other users. AMC-71 can be run on both time-sharing and batch-processing computer systems. Representative current computer running times to make predictions for a single vehicle in 1000 areal patches, once terrain and vehicle data are made available, are:

Areal predictions	2 min
Statistical consolidation	3 min

Figures for AMM-75 are expected to be of the same order.

The present supporting two-dimensional ride dynamics simulation, used in batch mode to simulate a normal military vehicle, runs at 10-20 times real time on a large third-generation digital computer. For a single vehicle, approximately four runs (at different speeds) over each of four 300-ft stretches of randomly rough terrain having rms elevation values from 0.5 to 3-in. are required to define the ride-speed curves used in AMC-71 and AMC-75. A like amount of computation is required to develop the obstacle crossing speed-limit relations as functions of obstacle height and spacing used in AMC-75.

The new AMM-75 obstacle negotiation model, as presently implemented (without refinements to minimize running time), requires 3 min to cross a single obstacle. Full exercise of the computer programmer's art will cut this in half, but even then the computer time to develop a 4 height x 4 width x 4 angle matirx will be of the order of 100 min. AMM-75 is deliberately structured so that this model need be run only once for a given vehicle, regardless of the number of areas the vehicle is subsequently checked against.

Further Developments

With the implementation and release of AMM-75, one major objective of the mobility elevation methodology development program will be substantially completed. Nonetheless, considerable directly related work will remain:

a. To validate the final AMM-75 algorithms and logic (the field validation program to date has dealt with AMC-71[19]) and make any necessary final adjustments.

b. To further upgrade the vehicle ride dynamics, obstacle negotiation, and linear feature crossing simulations.

c. To develop means to assess operationally reasonable time delays for NOGO situations.

d. To introduce variability of driver skill as a function of training.

e. To incorporate the model into the detailed vehicle design cycle through adapting it for use as an interactive computer-aided design and engineering tool.

f. To assist model users in their applications of AMM-75.

g. To manage the model once it is released; i.e. keep the full AMM-75 user community informed of all changes, from whatever quarter and of whatever magnitude, so that one, or two, or five years from now, all users will all have the same version at any given time.

With the successful demonstration by AMC-71 and AMM-75 of the potential benefits of deterministic engineering modeling of complex, terrain-dependent systems, emphasis is already rapidly shifting to new areas. Paramount among these are:

a. The development of vehicle-terrain-driver specific engineering modeling of combat vehicle agility performance.

b. The establishment of firm, supportive data interfaces between AMM-75 and higher order combat and logistics simulations.

c. The development of terrain and mission specific reliability modeling and its integration into the overall mobility evaluation methodology.

d. The application of the modeling approach and philosophy demonstrated in AMM-75 to other important military and commercial activities whose effectiveness is highly terrain dependent, such as combat engineering operations in support of mobility, pipeline construction and surface mining.

Concluding Remarks

AMM-75 is considered to be the cornerstone of a new unified engineering methodology for answering a broad range of mobility-related questions. AMM-75 is incomplete in some respects, imperfect in most. That is the nature of any simulation, a fact of which modern decision makers are aware. Nonetheless, used and interpreted with an appreciation of its inherent limitations, AMM-75 provides the vehicle development, procurement, and user communities with a set of analytical tools for obtaining quantitative engineering information to satisfy their needs in a systematic manner.

AMC-71 and AMM-75 have also proven to be the communication link long needed between users and researchers to guide further research and to establish common ground for the solution of vehicle designer and user problems. They are providing, across time and across organizations, objective, consistent communication among all elements responsible for improved Army mobility. Decisions growing in large part from the resulting new levels of technical understanding and communication will determine the major characteristics of the Army's vehicle fleet into the 1980's and beyond.

Acknowledgment

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References

1. Rula, A. A. and Nuttall, C. J., Jr., "An Analysis of Ground Mobility Models (ANAMOB)," Technical Report M-71-4, Jul 1971, U. S. Army Engineer Waterways Experiment Station, CE, Vicksburg, Miss.

2. "The AMC-71 Mobility Model," Technical Report No. 11789 (LL 143), Vols 1 and II, Jul 1973, U. S. Army Tank-Automotive Command, Warren, Mich.

3. Nuttall, C. J., Jr., Rula, A. A., and Dugoff, H. J., "Computer Model for Comprehensive Evaluation of Cross-Country Vehicle Mobility," Earthmoving Industry Conference, Paper No. 740426, presented at Society of Automotive Engineers, Apr 1974.

4. Nuttall, C. J., Jr., and Dugoff, H. J., "A Hierarchial Structure of Models for the Analysis of Land Mobility Systems," presented at Twelfth Annual U. S. Army Operations Research Symposium, Durham, N. C., Oct 1973.

5. Rula, A. A. and Nuttall, C. J., Jr., "Terrain Modeling to Support Mobility Evaluation," Proceedings of Fifth International Conference, The International Society for Terrain-Vehicle Systems, Inc., Detroit, Mich., Vol 4, Jun 1975.

6. Murphy, N. R. and Ahlvin, R. B., "Ride Dynamic Module for AMM-75 Ground Mobility Model," Proceeding of Fifth International Conference, The International Society for Terrain-Vehicle Systems, Inc., Detroit, Mich., Vol 4, Jun 1975.

7. Lessem, A. S., Jurkat, M. P., and Jacobson, R. W., "A Linear Model for Predicting the Obstacle Negotiation Characteristics for Suspended and/or Pitch Articulated Vehicles," (in preparation), Stevens Institute of Technology.

8. Lessem, A. S., "Dynamics of Wheeled Vehicles: A Mathematical Model for the Traversal of Rigid Obstacles by a Pneumatic Tire," Technical Report M-68-1, U. S. Army Waterways Experiment Station, CE, Vicksburg, Miss., May 1968.

9. Pradko, F., Lee, R. A., and Kaluza, V., "Theory of Human Vibration Response," Paper 66-WA-BHF-15, presented at ASME Meeting, Nov 1966.

10. Turnage, G. W., "Performance of Soils Under Tire Loads, Application of Test Results to Tire Selection for Off-Road Vehicles," Technical Report 3-666, Report 8, Sep 1972, U. S. Army Waterways Experiment Station, CE, Vicksburg, Miss.

11. Schreiner, B. G., "A Technique for Estimating the Performance of Tracked Vehicles in Muskeg," Journal of Terramechanics, Vol 4, No. 3, pp 23 to 29.

12. Sloss, D. A., Jr., "Hasty River and Dry Linear Feature Crossing Module of the U. S. Army Mobility Model (AMM-75)," Proceedings of Fifth International Conference, The International Society for Terrain-Vehicle Systems, Inc., Detroit, Mich., Vol 4, Jun 1975.

13. Smith, G., "Commercial Vehicles Performance and Fuel Economy," Society of Automotive Engineers, Special Publication 355, 1970.

14. Taborek, J. J., Mechanics of Vehicles, Penton Publication, 1957.

15. American Association of State Highway Officials (AASHO), "A Policy on Geometric Design of Rural Highways," 1965, The American General Offices, Washington, D. C., pp 152 to 159.

16. Janosi, Z. J. and Eilers, J. A., "Application of Dynamic Programming to Off-Road Mobility Problems," presented at Fourth International Conference of International Society of Terrain-Vehicle Systems, Stockholm, 24-28 April 1972.

17. Jurkat, M. P., "Data and Program Considerations for Path Selection in the AMC Mobility Model," Report SIT-DL-71-1564, Oct 1971, Stevens Institute of Technology, Hoboken, N. J.

18. Jurkat, M. P., "Automatic Path Selection for the AMC-71 Mobility Model," Report SIT-DL-73-1658, Sep 1973, Davidson Laboratory Stevens Institute of Technology, Hoboken, N. J.

19. Schreiner, B. G. and Willoughby, W. E., "Validation of the AMC-71 Mobility Model," Proceedings of Fifth International Conference, The International Society for Terrain-Vehicle Systems, Detroit, Mich., Vol 4, Jun 1975.

Table 1

Terrain, Vehicle, Driver Attributes Characterized in AMM-75 Mobility Model Data Base

Terrain	Vehicle	Driver

Surface composition
  Type
  Strength
Surface Geometry
  Slope/Altitude
  Discrete Obstacles
  Roughness
  Road Curvature/width/ Superelevation
Vegetation
  Stem size & spacing
  Visibility
Linear geometry
  Stream cross section
  Water velocity & depth
	Geometric Characteristics
	
	Inertial Characteristics
	
	Mechanical Characteristics
		Reaction times
		Recognition distance
		Acceleration and impact tolerances
		Minimum acceptable speeds

Table 2

Basic Content of Terrain Data Bank for Each Type of Patch or Segment

Terrain or Road Factor		Terrain Factor Family	Factor Complex
Off Road			
Slope		Surface Geometry	Areal (Patch)
Obstacle: Approach angle	Obstacle Geometry	Surface Geometry	Areal (Patch)
Obstacle: Vertical magnitude	Obstacle Geometry	Surface Geometry	Areal (Patch)
Obstacle: Length	Obstacle Geometry	Surface Geometry	Areal (Patch)
Obstacle: Width	Obstacle Geometry	Surface Geometry	Areal (Patch)
Obstacle: Spacing	Obstacle Geometry	Surface Geometry	Areal (Patch)
Obstacle: Spacing type	Obstacle Geometry	Surface Geometry	Areal (Patch)
Surface Roughness*		Surface Geometry	Areal (Patch)
Vegetation: Stem diameter	Vegetation Structure	Vegetation	Areal (Patch)
Vegetation: Stem spacing	Vegetation Structure	Vegetation	Areal (Patch)
Visibility*		Vegetation	Areal (Patch)
Surface Material: Type		Surface Composition	Areal (Patch); Linear (Segment)
Surface Material: Mass strength*		Surface Composition	Areal (Patch); Linear (Segment)
Water cover: Depth*		Surface Composition	Areal (Patch); Linear (Segment)
Water cover: Current*		Surface Composition	Areal (Patch); Linear (Segment)
Water cover: Width*		Surface Composition	Areal (Patch); Linear (Segment)
Left Approach Angle*		Surface Linear Geometry	Linear (Segment)
Right Approach Angle*		Surface Linear Geometry	Linear (Segment)
Differential Bank Height or differential vertical magnitude		Surface Linear Geometry	Linear (Segment)
Top width		Surface Linear Geometry	Linear (Segment)
Low bank height or least vertical magnitude		Surface Linear Geometry	Linear (Segment)
On Road			
Surface Material: Type		Surface Composition	Road (Segment)
Surface Material: Surface Strength and traction properties*		Surface Composition	Road (Segment)
Slope		Surface Geometry	Road (Segment)
Elevation		Surface Geometry	Road (Segment)
Surface Roughness*		Surface Geometry	Road (Segment)
Curvature		Surface Geometry	Road (Segment)
Super-elevation		Surface Geometry	Road (Segment)
Visibility*		Surface Geometry	Road (Segment)
Width		Surface Geometry	Road (Segment)

* Complete data for an area may include several values for any or all of these quantities representing seasonal variations. At run time appropriate value(s) are selected by the run specification.

Table 3

Terrain Data Required for AMC Mobility Model

Terrain or Road Factor	Range
Off Road	
Surface material	
Type, USCS/other	NA
Mass strength, CI or RCI	0->280
Slope, %	0->70
Obstacle	
Approach angle, deg	90-270
Vertical magnitude, cm	0->85
Length, m	0->150
Width, cm	0->1200
Spacing, m	0->60
Spacing, type	NA
Surface roughness, rms, cm	0-20
Stem diameter, cm (8 pairs)	0->25
Stem spacing, m (8 pairs)	0->100
Visibility, m	0->50
Water depth, m	0->5
Water velocity, mps	0->3.5
Water width, m	0->70
Top width, m	0->70
Left approach angle, deg	90-270
Right approach angle, deg	90-270
Differential bank height or differential vertical magnitude, m	0->4
Low bank height or least vertical magnitude, m	0->6
On Road	
Surface material	
Type, USCS/other	NA
Surface strength	
Trails, CI or RCI	0->280
Other, traction coefficients	0.01->0.80
Slope, %	0->70
Elevation, m	0->3000
Surface roughness, rms, in.	0->7.6
Curvature, deg	0-90
Width, m	1->60
Superelevation, %	0->10

Table 4

VEHICLE DATA FOR AMM-75 MOBILITY MODEL

1. Vehicle Identification

    Payload, Gross Combination weight (as characterized in data following)

2. Running Gear

2.1 Wheeled

Number of Axle Assemblies:

For each axle

Position (may be mixed with tracks)

Operating Load

Powered/Unpowered

Braked/Unbraked

Rim Type, Size

Tire Size

Tread

Construction

Rating

Rev./Mile

Nominal Diameter, OA

Width, OA

Section Height

Width

Inflation, Deflection: Sand

Cross Country

Highway

Number of Tires on Axle

Duals (Yes/No)

Tire Chains Fitted (Yes/No)

Central Tire Inflation (Yes/No)

Axle Ground Clearance

Axle Tread

Clearance Between Right-Left Tires

2.2 Tracked

Number of Track Pair Assemblies

(continued)

(Sheet 1 of 5)

For each pair

- Position (may be mixed with wheels)
- Operating Load
- Powered/Unpowered
- Braked/Unbraked
- Suspension Type
- Track Type (Flexible/Girderized)
  - Width
  - Pitch
  - Grouser Height
  - Thickness
  - Single Shoe Road Pad Area
  - Length on Ground
- Number of Road Wheels
- Road Wheel Diameter
- Hull Ground Clearance
- Track Tread
- Clearance Between Right-Left Tracks

3. Power Train
   - Tractive Force-Speed Curve (Optional)
   - Engine Identification
     - Maximum Gross HP, RPM
     - Maximum Gross Torque, RPM
     - Maximum Net HP, RPM
     - Maximum Net Torque, RPM
     - Torque-RPM Curve
   - Engine-to-Transmission Transfer Gears
     - Ratios, Efficiencies
   - Torque Converter (Yes/No)
     - Identification
     - Torque Ratio-Speed Ratio Curve

(continued)

(Sheet 2 of 5)

Input RPM-Speed Ratio Curve

Input Torque for Above

Converter Accessory Loss Curve

Lockup (Yes/No)

Transmission Identification

Gear Ratios, Efficiencies

Shift Times

Transmission-to-Final Drive Transfer Gears

Identification

Gear Ratios, Efficiencies

Final Drive Identification

Gear Ratio, Efficiency

Acceleration Mass Factors

Overall Gear Ratios, Factors

4. Vehicle Geometry

Overall Dimensions

Length (Combination)

Wheel Base (Prime Mover)

Width

Minimum Ground Clearance (except axles)

Angle of Approach

Departure

Pitch Joint/Fifth Wheel/Pintle (yes/no)

Distance from Front Axle/Road Wheel

Height Above Ground

Center of gravity

For Each Unit and Combination

Height Above Ground

Longitudinal, from Front Axle/Road Wheel

Lateral, from Vehicle CL

(continued)

Table 4 (continued)

Axle/Road Wheel Arrangement*

For each position

Axle Distance fron Front Axle/Road Wheel

Full Bump to Rebound Axle/Road Wheel Travel

Tandem Assembly (No, Dual, Triple)

Other Wheel Positions in Same Assembly

Bogie Axle Distance fron Front Axle/Road Wheel

Mean Spring Rate Between Stops (Two Sides)

Vehicle Bottom Clearnce Profile*

(Approximated by straight lines, specified by x-y coordinates of breakpoints, referenced to axes through Front Axle/Road Wheel Center, positive up and to the rear)

Number of x-y coordinate pairs

x-y coordinate pairs

Other

Height of Bumper/Push Point Above Ground

Height of Driver's Forward Line-of-Sight Above Ground

Maximum Depression of Driver's Forward Line-of-Sight

5. Water Characteristics

Fording Depth, Speed

Swamping Angle, Ingress, Egress

Floater (Yes/No)

Hull Type

Waterline Length

Beam

Draft to Hull Bottom

Minimum Freeboard

Propulsion System Type

Still Water Speed w/o Auxilary Propulsion

Still Water Speed with Auxiliary Propulsion

Width Required to Use Auxiliary Propulsion

Depth Required to Use Auxiliary Propulsion

Bouyancy versus Draft Curve

---

* Used in obstacle interference and traction module.

(continued)

Table 4 (concluded)

6. Highway Characteristics (Wheeled Vehicles Only)

   Aerodynamic Drag Coefficient

   Frontal Area

   Cornering Stiffness of Tires (at Highway Inflation and Load)

7. Mobility Assist Systems

   Winch Capacity; Speed

   Pushbar/Bumper Capacity

8. Ride and Obstacle Speed Limits (to one Ride Dynamics Module or Controlled Experiments)

   Number of Absorbed Power Levels

   Ride Speed Limit-RMS Curve for Each Absorbed Power Level

   Impact Speed Limit versus Obstacle Height Curve (Single Obstacles)

     Single Obstacle Height at 15 mph Limit (=HS)

   Impact Speed Limit versus Obstacle Spacing Curve (For Obstacle Height HS)

   NOTE: Requirements for additional data to use AMM-75 2-dimensional ride and obstacle impact simulation to develop above data are given in Reference 6.

9. Obstacle Interference/Clearance and Traction (from Obstacle Interference amd Traction Module)

   For Each of 3 or More Obstacle Heights with 3 or More Obstacle Widths with 3 or More Obstacle Approach Angles (27 or more):

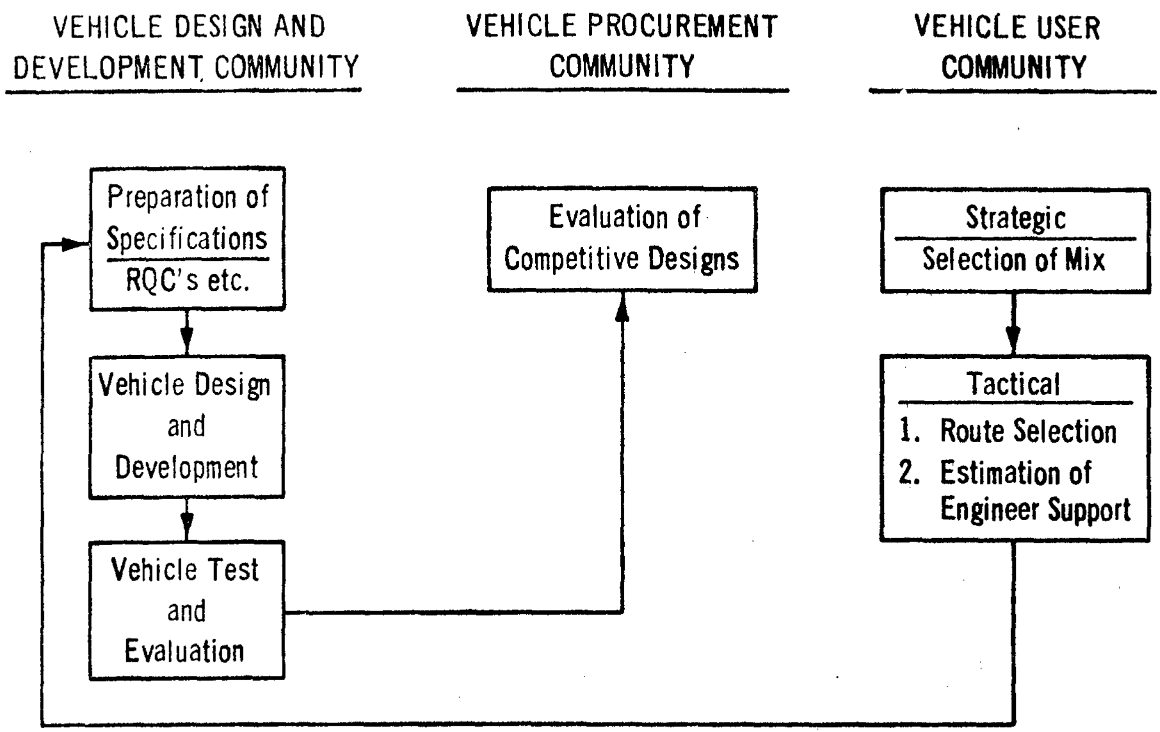
     Minimum Clearance During Crossing (Negative = Interference)

     Distance of Critical Clearance Point Behind Front Axle/Roadwheel

     Maximum Traction Required During Crossing

     Mean Traction Required During Crossing

(Sheet 5 of 5)



PROSPECTIVE USERS OF VEHICLE PERFORMANCE PREDICTION METHODOLOGY

Figure 1

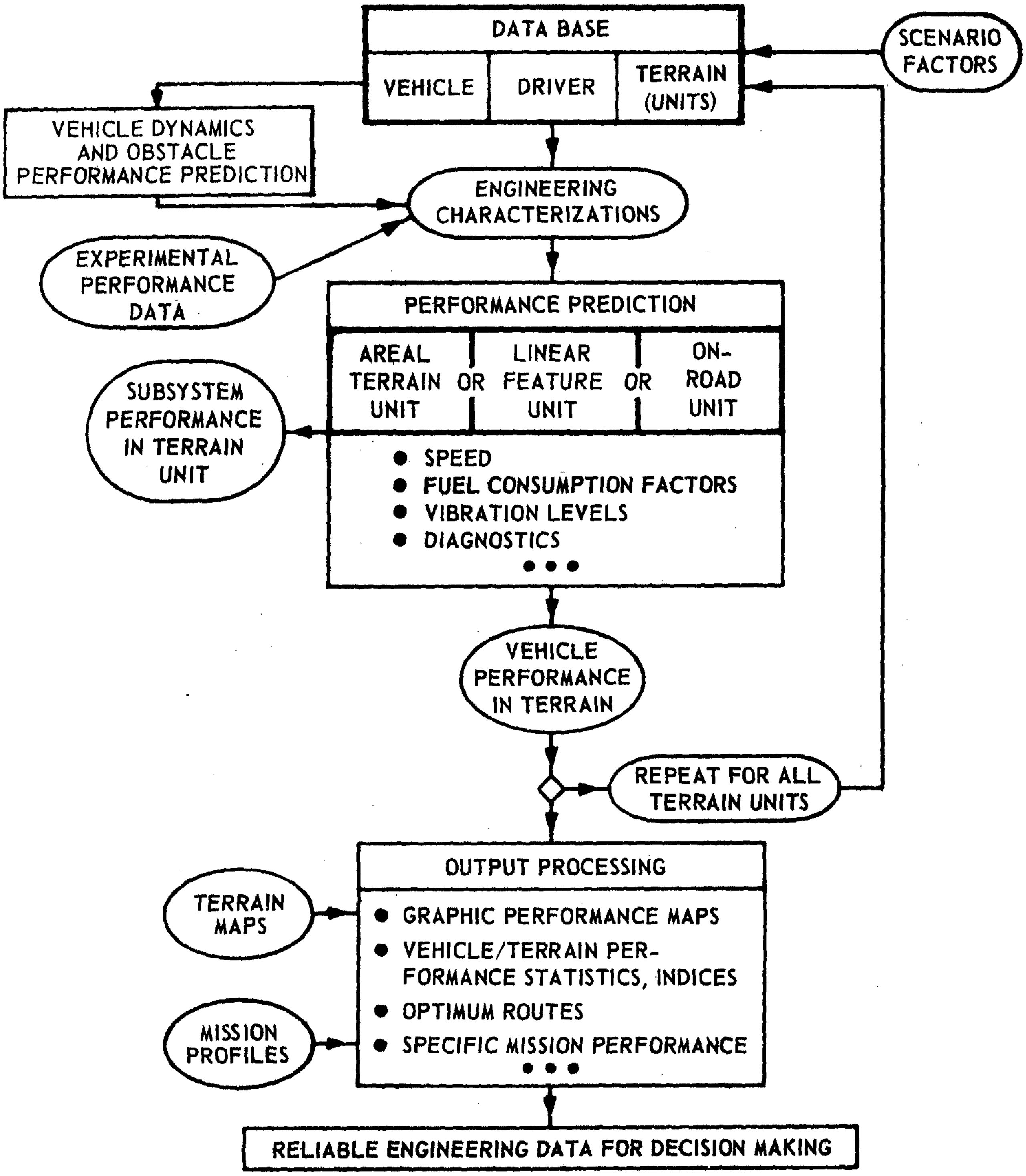
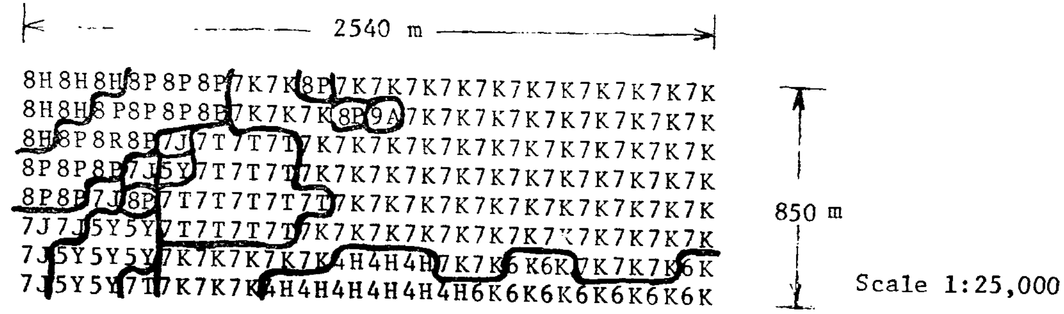


Figure 2. Gross structure of Army Mobility Model



LEGEND Patch Number		SOIL Type	SOIL CI or RCI (Wet-Wet)	Slope, %	OBSTACLES Angle, deg,	OBSTACLES Vertical magnitude, cm	OBSTACLES Width, cm	OBSTACLES Length, m	OBSTACLES Spacing, m	OBSTACLES Spacing type	Surface roughness, in. x 10 cm	STEM SPACING: Mean spacing (m) of stems having diameter >Y (cm) Y 0.2	3	6	10	14	18	22	25	Recognition distance, m (Jul-Sep)
....	...																			
(4H)	112	2	129	3	138	30	86	1	22	1	7	6	7	15	19	100	100	100	100	11
....	...																			
(5Y)	155	2	103	8	198	46	94	21	5	2	6	1	18	100	100	100	100	100	100	15
....	...																			
(6K)	167	2	111	4	188	33	103	20	24	2	7	6	6	6	7	7	7	7	7	10
....	...																			
(7J)	192	2	133	1	179	10	963	1	60	1	11	5	6	6	8	13	20	100	100	10
(7K)	193	1	114	7	197	46	104	30	56	2	12	5	6	6	20	100	100	100	100	11
....	...																			
(7T)	202	2	129	10	179	10	940	1	60	1	11	6	6	13	20	100	100	100	100	8
....	...																			
(8H)	216	2	130	4	204	30	112	22	32	2	8	6	7	7	7	7	7	7	7	32
....	...																			
(8P)	224	1	102	7	141	53	130	2	26	1	8	7	8	10	17	100	100	100	100	48
....	...																			
(9A)	235	2	125	6	108	43	71	18	29	2	1	5	5	6	7	18	21	100	100	7
....	...																			

Figure 3. Sample terrain factor complex map

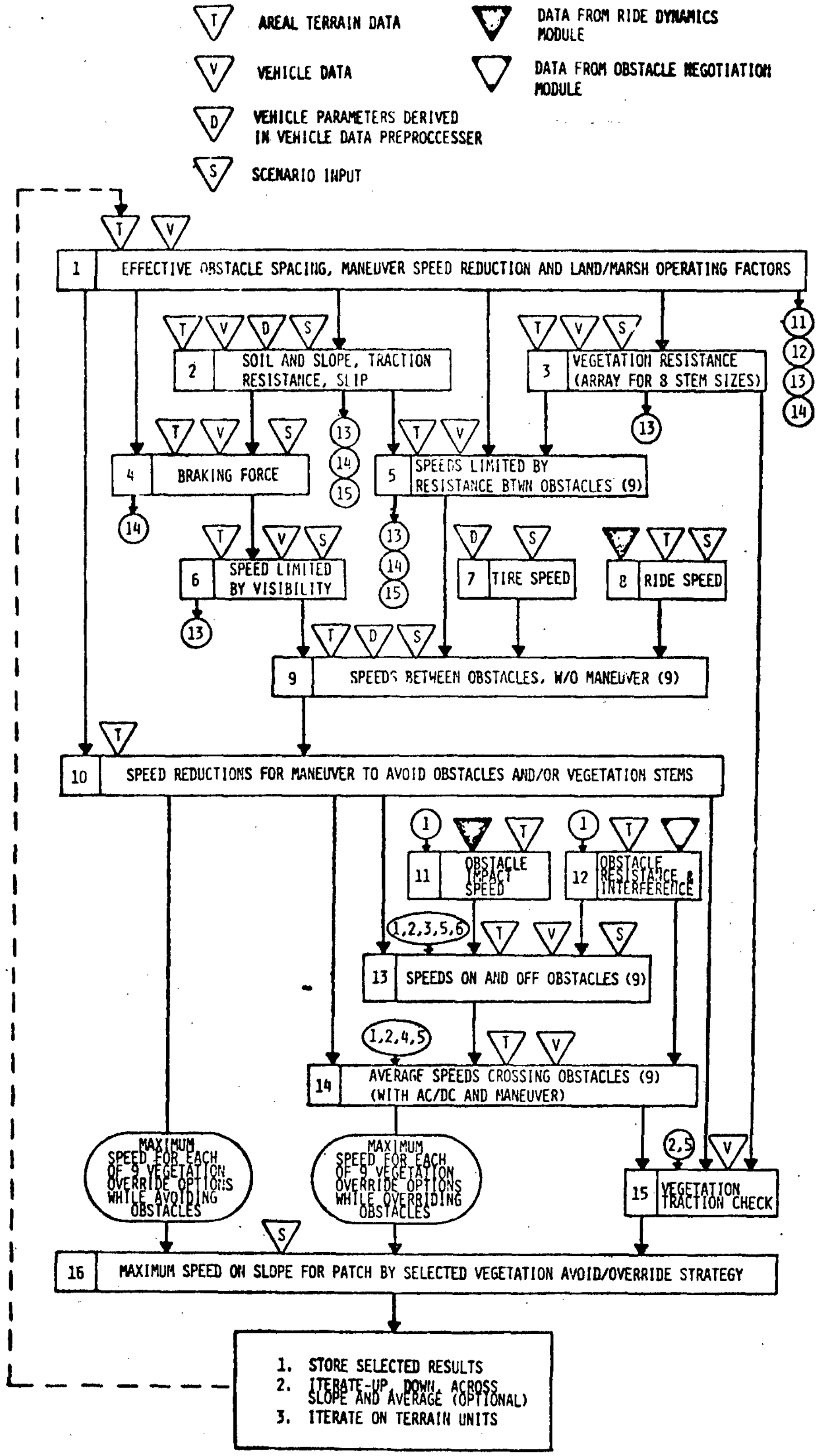


FIGURE 4. GENERAL FLOW OF AMM-75 AREAL MODULE

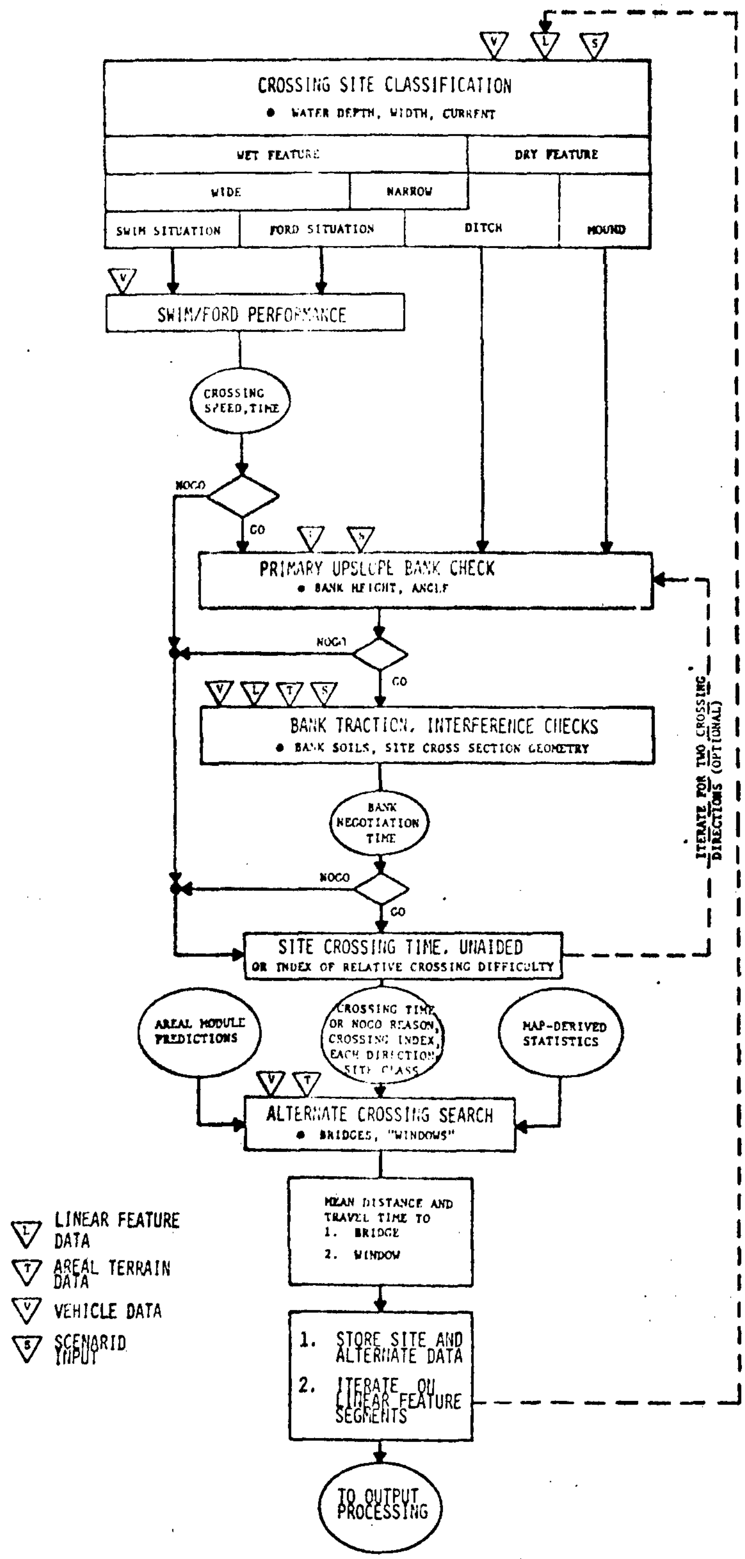


FIGURE 5. GENERAL FLOW OF AMM-75 LINEAR FEATURE CROSSING MODULE

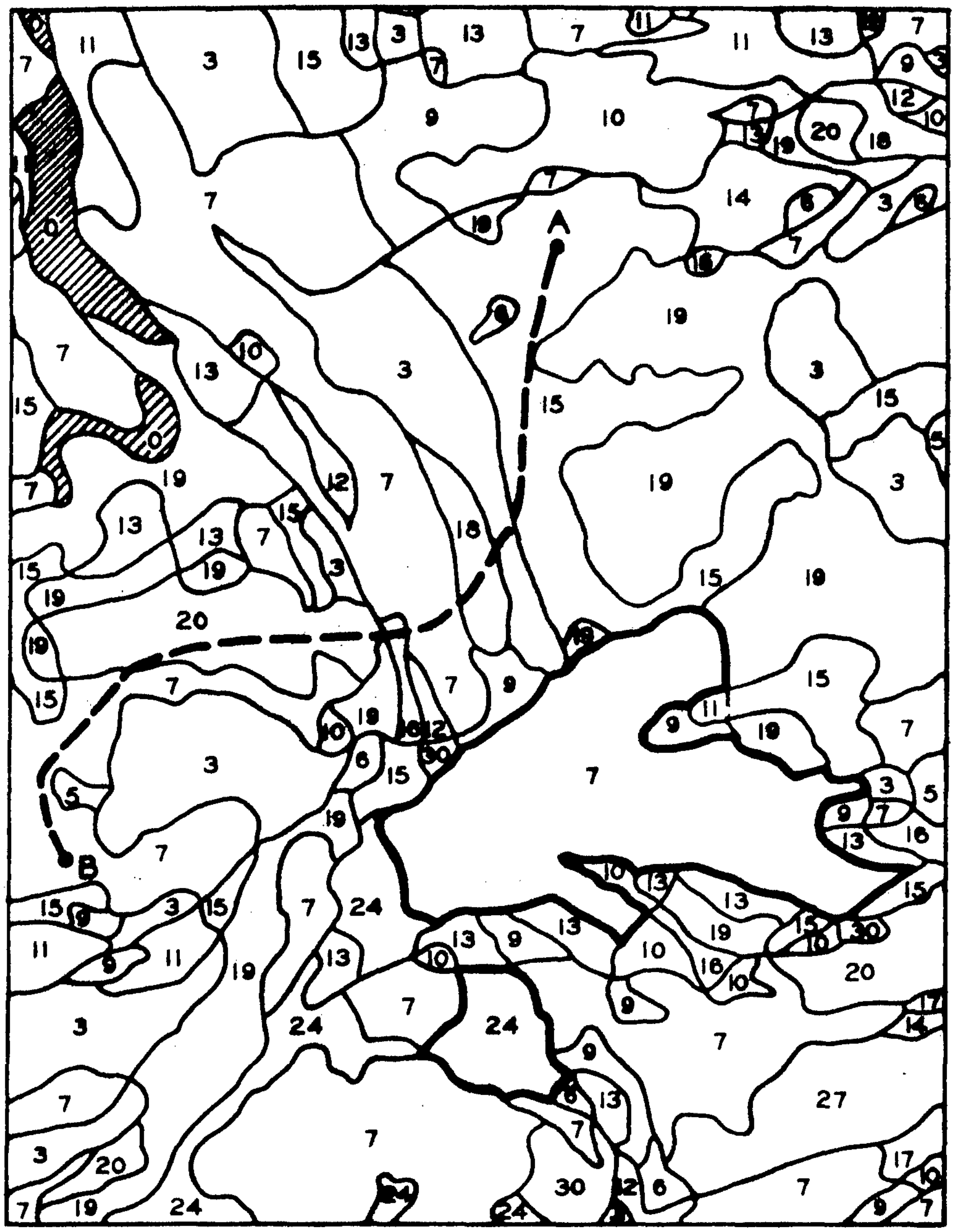
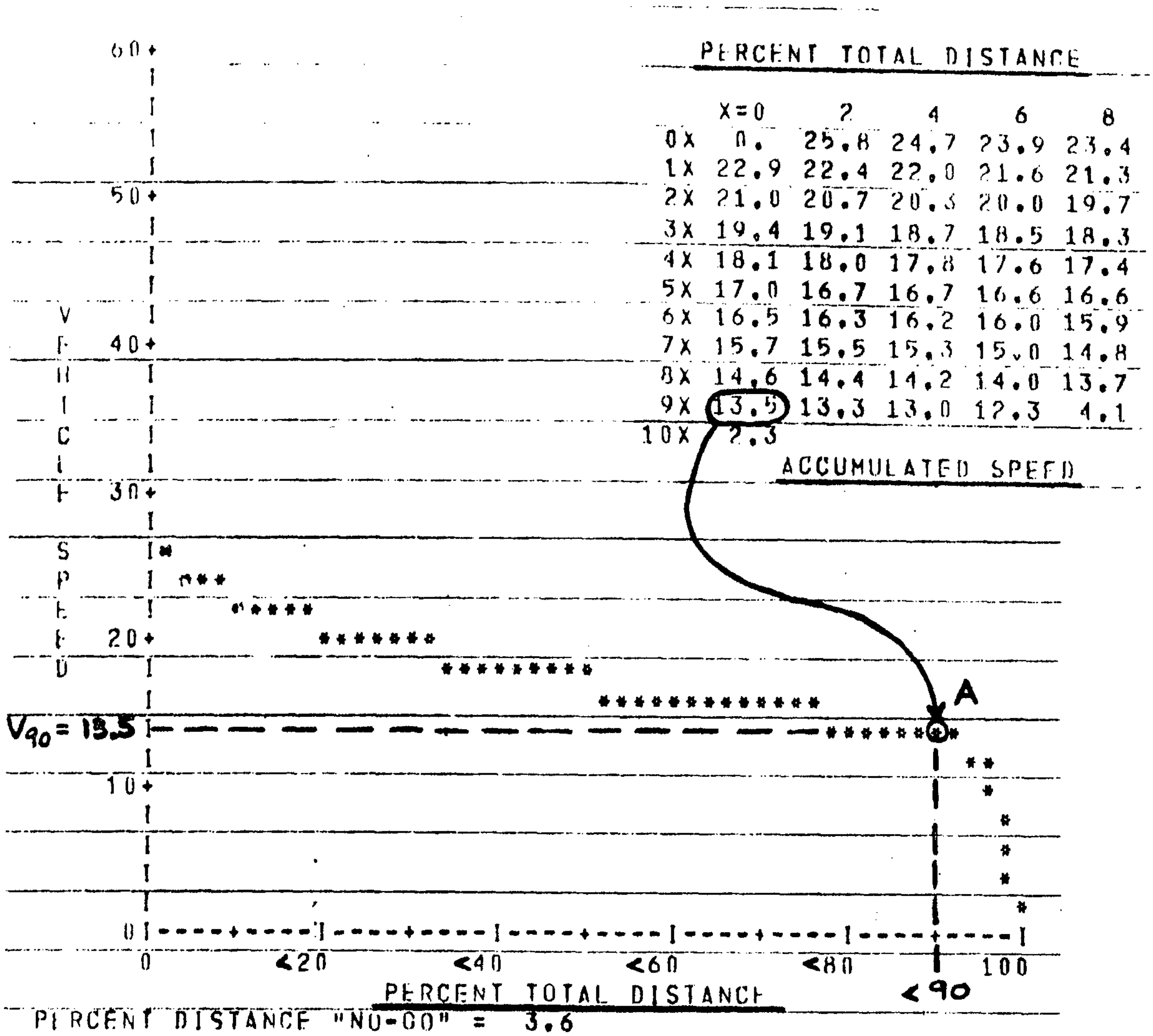


Figure 6. Mobility map of off-road performance of 2-1/2-ton truck speeds in mph



	X=0	2	4	6	8
0X	0.	25.8	24.7	23.9	23.4
1X	22.9	22.4	22.0	21.6	21.3
2X	21.0	20.7	20.3	20.0	19.7
3X	19.4	19.1	18.7	18.5	18.3
4X	18.1	18.0	17.8	17.6	17.4
5X	17.0	16.7	16.7	16.6	16.6
6X	16.5	16.3	16.2	16.0	15.9
7X	15.7	15.5	15.3	15.0	14.8
8X	14.6	14.4	14.2	14.0	13.7
9X	13.5	13.3	13.0	12.3	4.1
10X	2.3				

Figure 7. Mobility profile of off-road performance of 5-ton truck in desert terrain

```
    100+
       I    FACTOR LIMITING SPEED                          % AREA    AVERAGE
       I                                                   LIMITED    SPEED
       I
       I  ( 1)  INSUFFICIENT SOIL STREGNTH                  0.          -
     90+  ( 2)  INSUFFICIENT TRACTION                       0.          -
       I  ( 3)  OBSTACLE INTERFERENCE                       3.6       NO-GO
       I  ( 4)  COMBINATION OF TERRAIN FACTORS              0.          -
       I  ( 5)  ROUGHNESS (RIDE) SPEED LIMIT                0.          -
     80+  ( 6)  SOIL/SLOPE RESISTANCES                      8.7        10.9
 P     I  ( 7)  VISIBILITY LIMIT                            0.          -
 E     I  ( 8)  MANEUVER PROBLEM                           47.9        16.5
 R     I  ( 9)  VEGETATION RESISTANCES                      9.4        10.8
 C   70+  (10)  AC/DC BETWEEN OBSTACLES                    23.9         7.9
 E     I  (11)  EXTERNAL (URBAN) SPEED LIMIT                6.5        15.5
 N     I
 T     I
     60+
 T     I
 O     I
 T     I
 A   50+
 L     I                                          XXXXX
       I                                          XXXXX
 D     I                                          XXXXX
 I   40+                                          XXXXX
 S     I                                          XXXXX
 T     I                                          XXXXX
 A     I                                          XXXXX
 N   30+                                          XXXXX
 C     I                                          XXXXX
 E     I                                          XXXXX     XXXXX
       I                                          XXXXX     XXXXX
     20+                                          XXXXX     XXXXX
       I                                          XXXXX     XXXXX
       I                                          XXXXX     XXXXX
       I                                          XXXXX     XXXXX
     10+                              XXXXX       XXXXXXXXXXXXXXX
       I                              XXXXX       XXXXXXXXXXXXXXXXXXXX
       I            XXXXX             XXXXX       XXXXXXXXXXXXXXXXXXXX
       I            XXXXX             XXXXX       XXXXXXXXXXXXXXXXXXXX
      0+------------ 0.1 ------------10.9------16.5-10.8- 7.9-15.5
          (1)  (2)  (3)  (4)  (5)  (6)  (7)  (8)  (9)  (10) (11)

                        FACTOR LIMITING SPEED
```

Figure 8. Diagnostic output of AMM-75. Distribution of performance-limiting mobility impediments to 5-ton truck in arid terrain

MEASUREMENT OF TERRAIN SURFACES

Presented By

W. RAASCH
FRG

Surface Roughness as a Disturbance Variable for Running Vehicles

1. Terrain as Roadway

Track-laying vehicles are running during their essential missions mostly in terrains which are not spezial prepared for this reason. In spite of this the vehicles should not be rather disturbed in performing their missions.

There are two proporties of terrain which cause this disturbance

- the surface roughness
  this roughness and the vehicles velocity bring out a permanent disturbance to the vehicle

- the soil parameters which means soil friction, soil cohesion, moisture, density and plasticity

I, now will espacially discuss problems of the terrain's roughness.

The terrain's roughness limits the mobility of vehicle which have a nonidial running gear by the following criterias

For this reason the applying people of the vehicle have to make some decissions which are not questions of a technical optimization.

So it is necessary to describe clearly missions in peace of the vehicle, especially concerning the training grounds. You have also to give a response to the question of the modification of the terrain in aspect of war influences, it means synthetic ground roughness as a result of projectile impacts.

If all of these questions are answerd quantitatively there is no problem in constructing an optimized running gear of a tank.

At present we are engaged in contributing energy in these problems by measuring the roughness of mission grounds and trying to get generalized results.

2. Measuring Methods

To resolve this problem you need a suitable measuring method.

Usual methods are

- the measurement of discreet single points with theodolite
- the direct scanning of surface
- the indirect measurement of surface

The measurment with theodolite does'nt come into question because of wasting a lot of time.

For the direct scanning of surface there are a lot of methods. IABG has developed two different methods and also used which I will shortly discuss as examples of such methods.

The first method, here called IABG I, uses a sort of pair of compasses with constant length of legs of compasses. With this the surface is feeled out and the surface profile is transformed in tho two angles β and γ.

The inclination of the reference platform against vertical is measured by an inclination measuring equipment it means a sort of pendulum. The measurement range of one step is 2.5 Meters.

applied often for wheeled vehicles. In the last time such an equipment is developed and used by the Battelle company (Figure 5) in FRG.

You get here the height signal of the profile out of the measurement of the vertical axle's acceleration. This signal must be double integrated to get the height signal. Therefore the measuring bandwidth is limeted. There are also some problems to get the real horizontal displacement and not the development of the profile.

On the other hand the measuring velocity is rather high.

The methods described till now are never the less not suitable to get the demanded amount of surface profile datas.
We think the only suitable method is the photogrammetric one.

By taking a set of air surveys as shown in figure 6 you can measured out the surface profile in the laboratory.

We have been testing this method and found out that it is possible to get a solution of 0.1 Meter which will be just enough.

Figure 7 shows a comparison of the decribed method and figured out once more the profit of the photogrammetric

measurement. It is possible to take air surveys of large terrains in a rather short time and measure out the terrain profile later on and only such positions that are interesting.
Further more there is a very good appointment to terrain or to maps.

When displacing the reference point O the second leg is fixed at point P and for the absolute coordinates of this point does not change the new coordinate of the reference point are measured.

The reference platform (figure 2) is fixed on a little handcar together with the magnetic taperecorder which records the measured datas. These are the three angles α, β, γ and a reference signal for the conditions: measurement and displacement.

Figure 3 shows a view of the equipment without tape recorder.

The second method that was handled by IABG, here called IABG II works by a simular principle. Figure 4 shows the scheme. The surface of terrain is feeled out with a wheel. This wheel is connected versus a wire with the originator of the signal of the angle of pitch and the length of the wire -this means the distance between measuring- wheel and originator-head. For the pitch signal is originated as a sine-and a cosine-function and this signal is multiplied analogous with the value of wire length you get directly the height and length coordinates of profile. It is possible to measure also the horizontal position with a third originator.
I think there are a lot of simular methods and the described two shall stand for them .

A third rather good known method is working with a measuring wheel. There exist a lot of systems which are

- the vehicle gets in its natural frequency

- some obstacles cannot be overcome

- the divers's and commander's sight is restrained

- the rolling resistance rise

On the other hand there is the demand to the vehicle - and this is the job of the running gear - to operate without the influence of the terrain's properties. That means the running gear of such a vehicle should work in that way, that the disturbance level is minimized. Out of this you can see, that it is very important to have decisive datas of the roughness of those terrains in which designed vehicles should operate.

This is a request which cannot be realized directly. So there must be found standard values of the surface roughness. This is done serveal times, but those standard values are not fully satisfying till now.

We think, that for getting such standard values you need a reasonable lot of roughness- measurment datas of all possible mission terrains, so that on one hand the requests to the running gear are high enough especially if you have to run with high speeds, on the other hand the request to the running gear should not be to high to get not technical results, which are to sophisticated, expensive and may be too unreliable.

3. Interpolation of Terrain-measurments

As you need a areal solution of the terrain to find out the standard statistical cocfficients of the measurement the effort of evaluation is extraordinary high. Therefore it is necessary to record areas of no essential roughness only with little denity. Never the less the accuracy of the wheel should be high.

It is reasonable to use suitable computation methods for this. We intend to use a digital-terrain-modell for this purpose. This modell based on laying a polynominal plane through the measured heightcoordinates by using the method of balancing calculation. (Figure 8) For the use it is important to describe the surface wholy by laying down suitable equations for the interpolation.

As Figure 9 point out this is possible also if the measured point are in a very different position. In the moment we try to find out whether this method gives a reduction of effort or not.

4. Examples of Photogrammetric Measurements

With the following two examples of photogrammetric surface measurement I intend to point out the large differences of terrain roughness used by tanks during mission.

The first example is a provingground in the surroun- of Munich at Schleißheim. This provingground is expecially used for testing the running gears of tanks.

Figure 10 shows one of the air surveys which are use for photogrammetric measurement. You can see here rather good the marks of the tracks.

We have evaluated a lot of straight tracks and a closed course which was passed over during measurement on running gears.

Figure 11 shows 4 parallel tracks in the distance of 1.5 Meter. We will find out here also the excitation of the rolling movement of the vehicle, on the other hand you can see the high coherence of the four tracks. You also see the rather high roughness of this terrain within -short wave lengths.

The second example shows a surface that is typical for the terrain where tank are operating during tactical missions. It is a part of the terrain where the manoeuver Reforger happens last year(Figure 12).

Also here you can see the marks of the tracks. We have evaluated these tracks.

Figure 13 now shows the contourline of a tank path and you see that the surface is rather smoth within short wave length which are essential for vehicle excitation. As an comparison there is also plotted the contourline of the circular course of Schleißheim. As the air survery points out also the roughness of terrain is rather small if you have agricultural used terrain, where as provingground seemed to have a much higher surface roughness. The following two pictures shows only a small part of both terrain with a higher solution (Figure 14 und 15) Figure 15 points out also another characteristic of the agricultural used terrain that are places where the tank passes a road or a drain.

In the following short movie you will also see that the agricultural terrains makes no problems for tank but the crossing a road or a drain.

The movie was taken during a lot of manoever-like missions with Leopard I at places which are realy tactical mission terrain and which are dispersed all over FRG.

5. Evaluation of Problems of soil mechanic by air survey

I will only mention this problem (figure 16) for we yet have no expirience in this. But as you surely know and as this air survey points out it is possible to find out the moisture and some other soil values with the method of false coulour. May be it will be possible to get quikly the datas you need to give a actual answer of the accessibility of a certain terrain.

We well make efforts to find this out.

6. Mathematical Evaluation of Terrain Roughness

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		Nr. B-TF 333	15

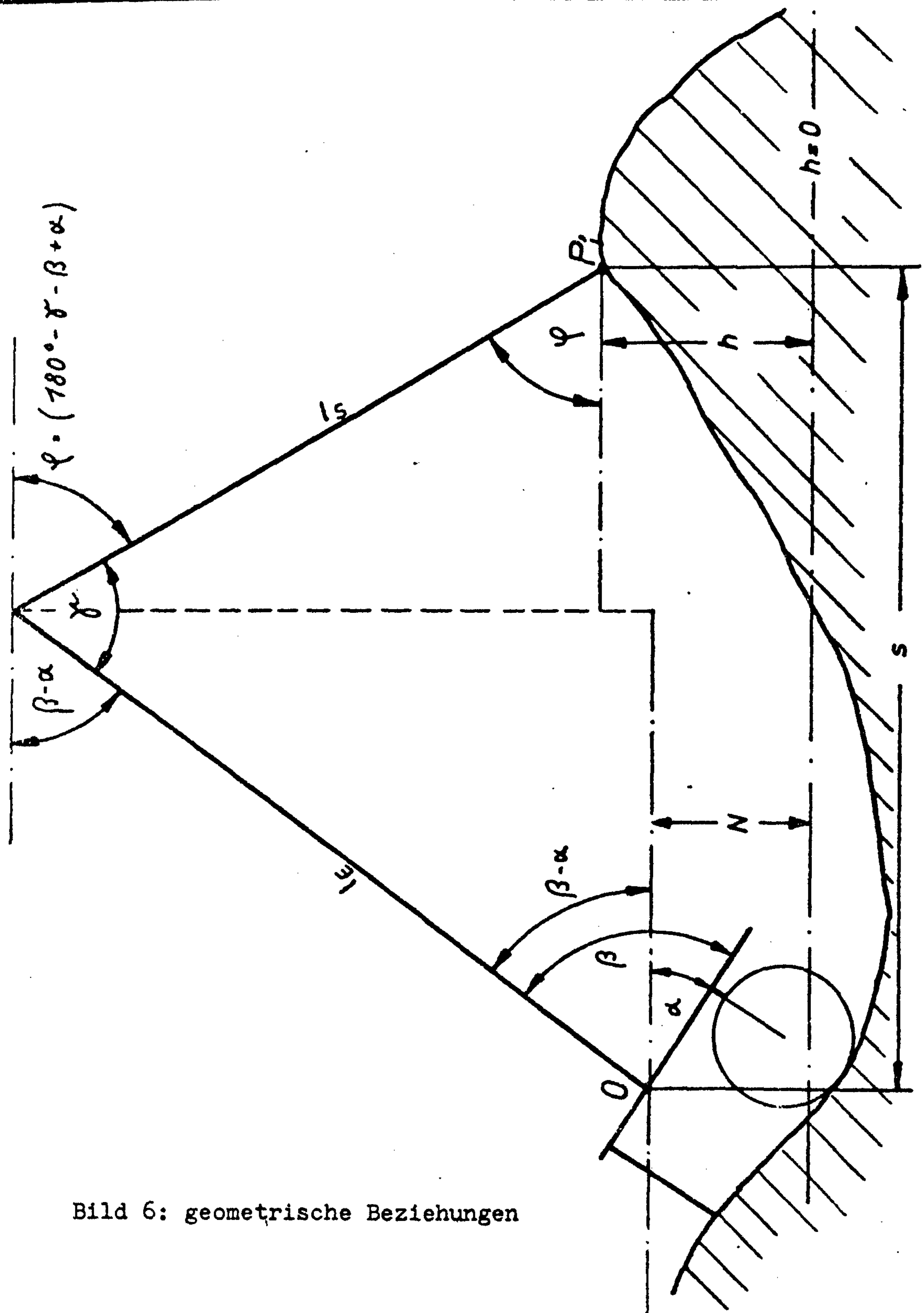

Bild 6: geometrische Beziehungen

Bearbeiter Mohr	Datum 21.05.73		

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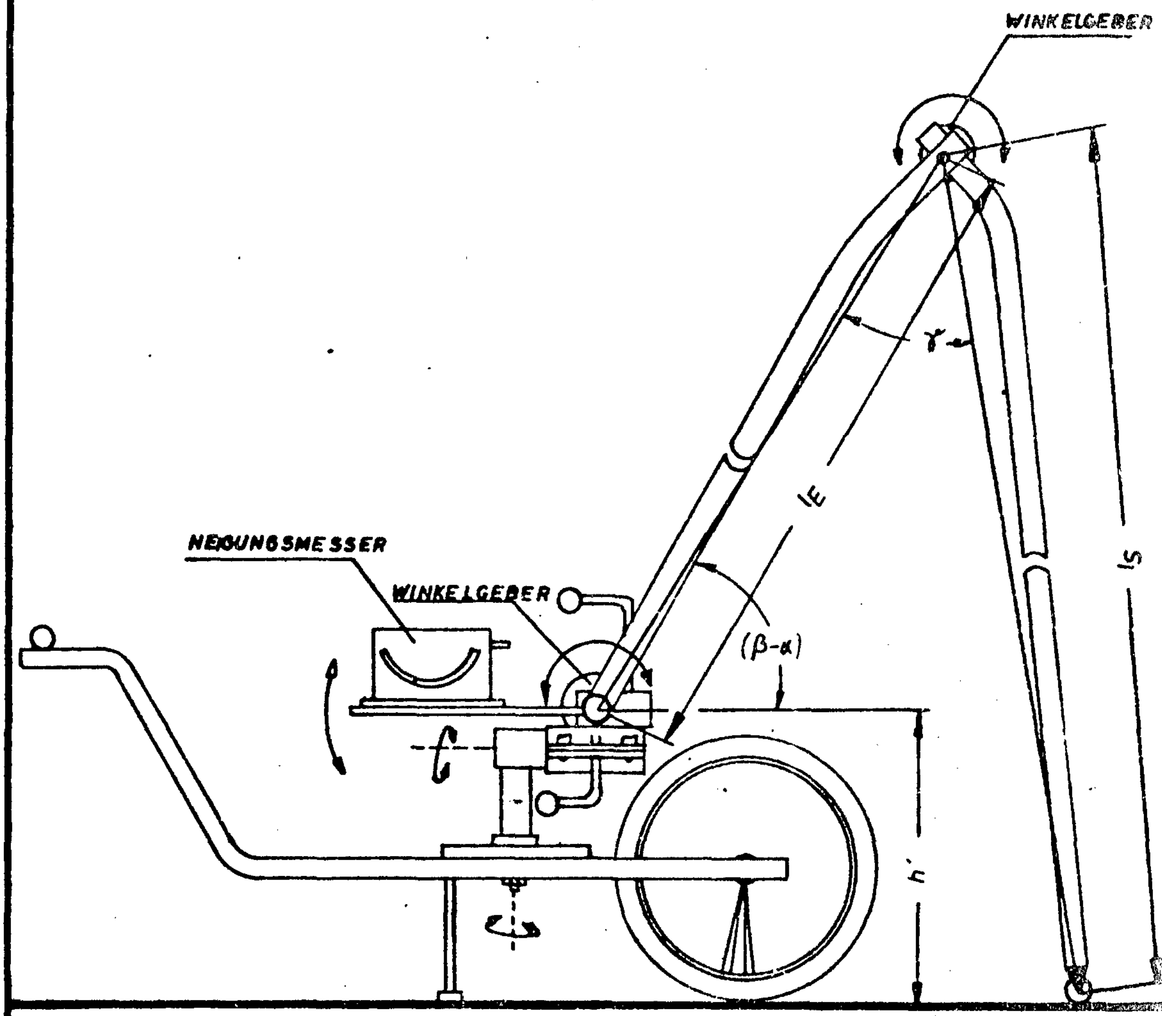
		Nr. B-TF 333	5

Bild 1 : Aufbau des Meßgerätes



α = **Neigungswinkel**

β = Elevationswinkel

γ = Spreizwinkel

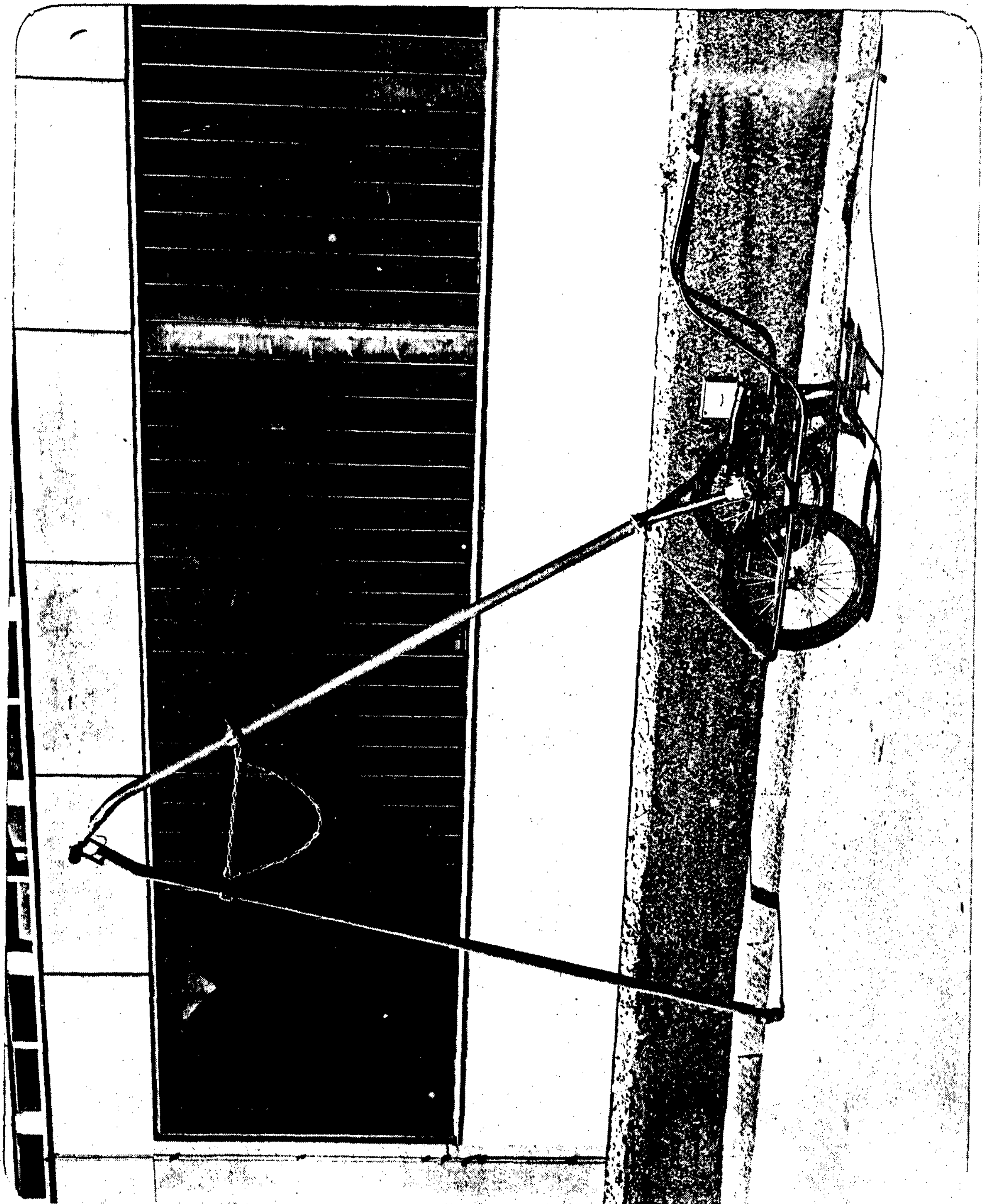
l_E = Elevationslänge = 2940 mm

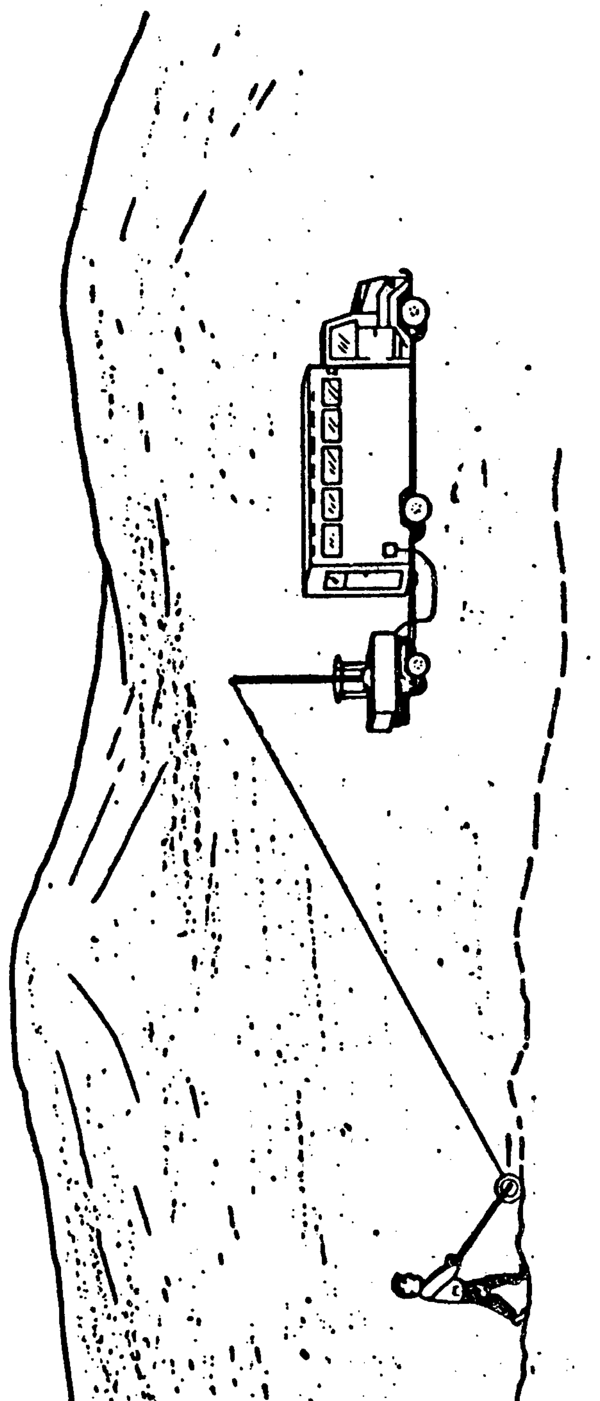
l_S = Spreizlänge = 2983 mm

h = 403 mm

20.92.1.1.

Bearbeiter	Datum		
Cronenberg	21.5.1973/st		

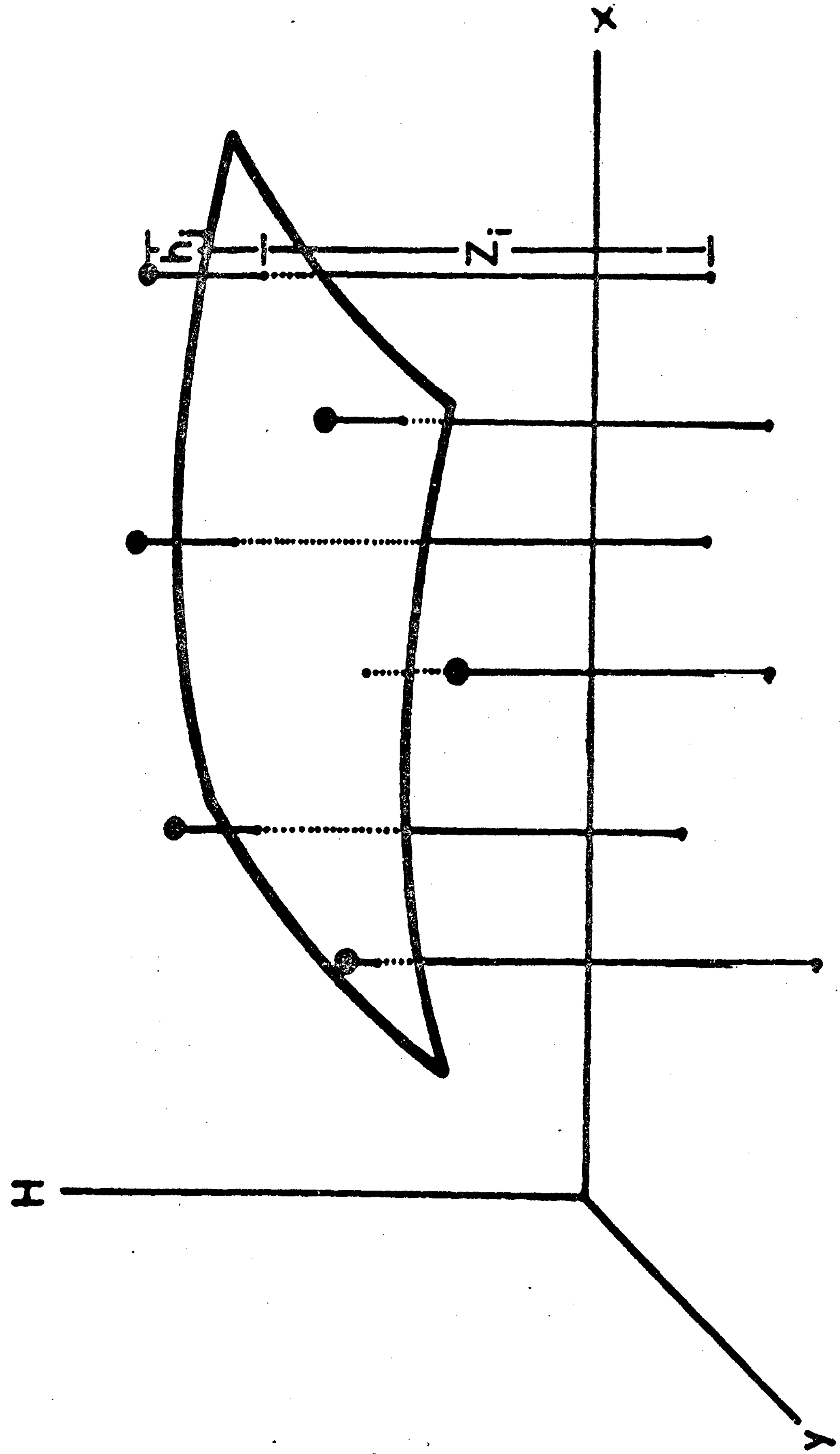


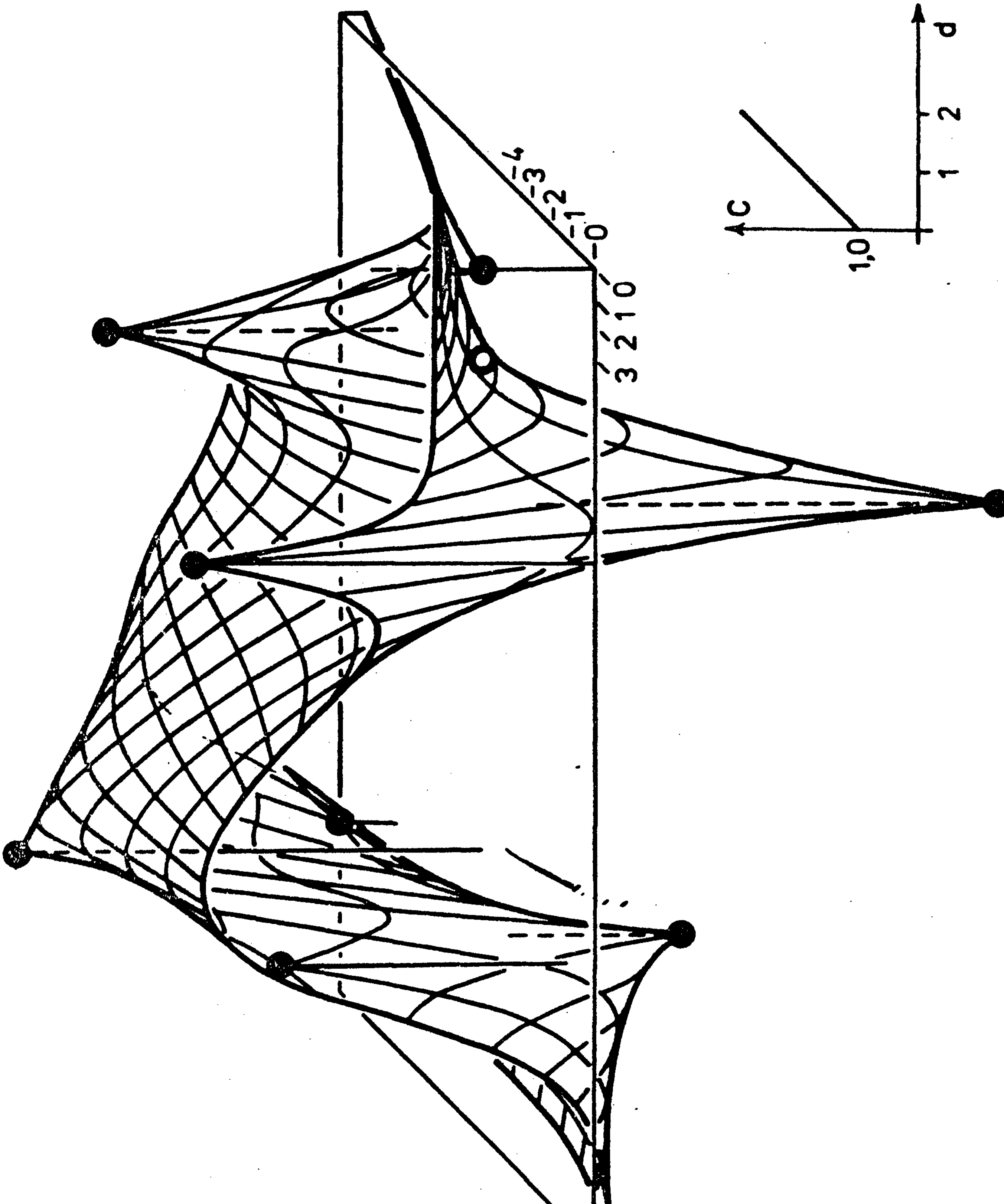


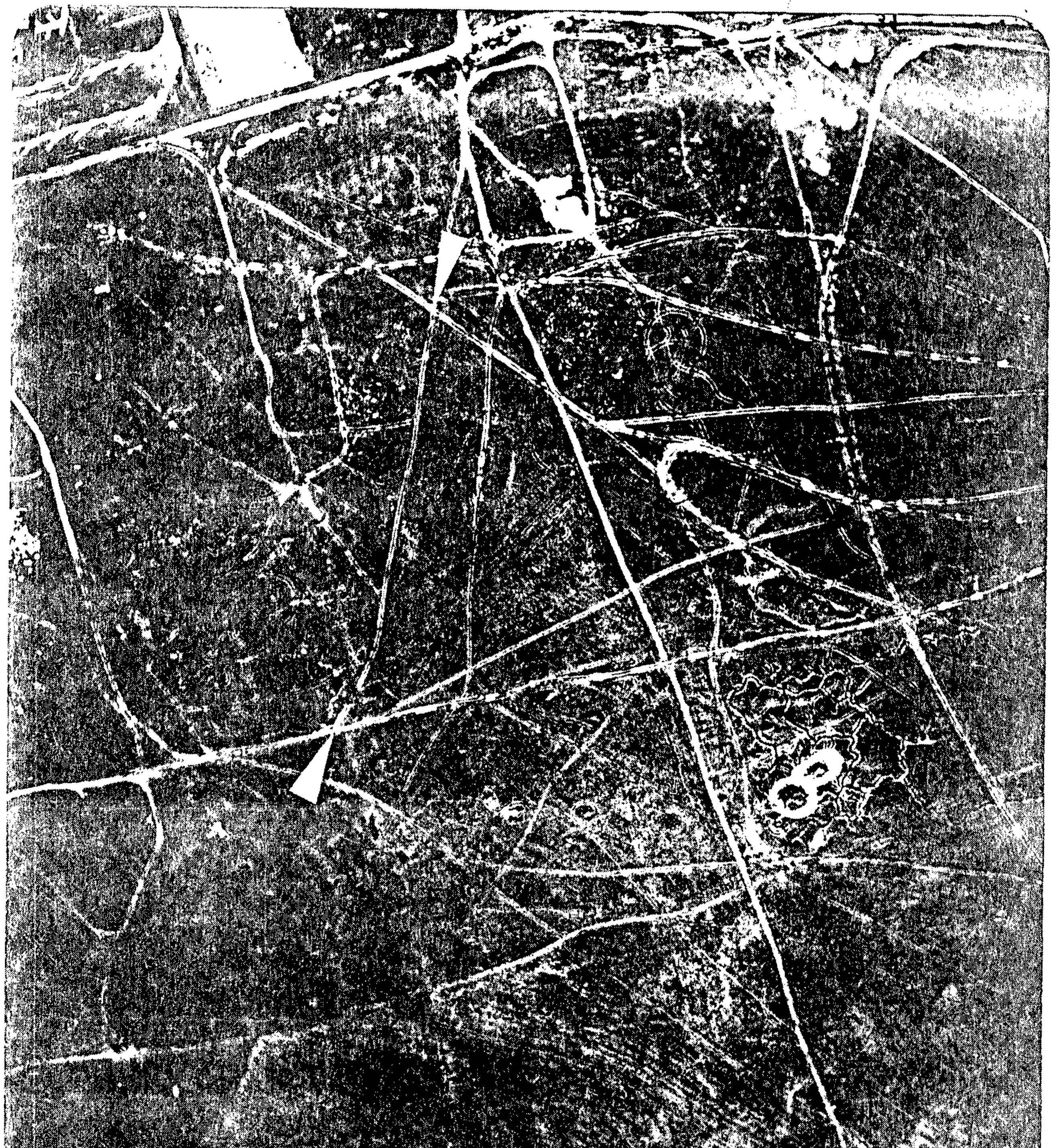
Versuchsanordnung bei der Fahrkurs- und Bodenprofilmessung

	Single Point Measurement	Direct Surface Pick Up			Not-direct Pick Up
	Theodolite	IABG I	IABG II	Measuring Wheel	Photogrammetric
Resolution	10^{-2} m	10^{-2} m	10^{-1} - 10^{-2} m	10^{-2} m	10^{-2} m
Profile precision	not good	good	good	not good	within the resolution good
Three dimensional Measusoment	yes	no	yes	no	yes
Orientation	only possible by laying down the course of measurement beforehand				very good by reestabilised air survey
Influencing of profile	no	no	a little	yes	no
Handling	very good	good	complicated	good	very good
Effort of evaluation	little	must be reestablished	little	measured value must be double integrated	rather high
Velocety of Measurement	little	500 - 1000 m/h	1000 - 2000 m/h	ca. 5000 m/h	ve.y high
Special property	Measured values only in listings	small	measurement output ground profile; terain must be passed over with truck	terrain must be passed over with truck	air survey, three dimensional measurement; depending on wheather

COMPARISON OF METHODS OF GROUND SURFACE MEASUREMENTS


X
h
Z
H
Y


ΔC
d
1,0
1
2
0 1 2 3
0 1 2 3 4



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TFM

Auftrag

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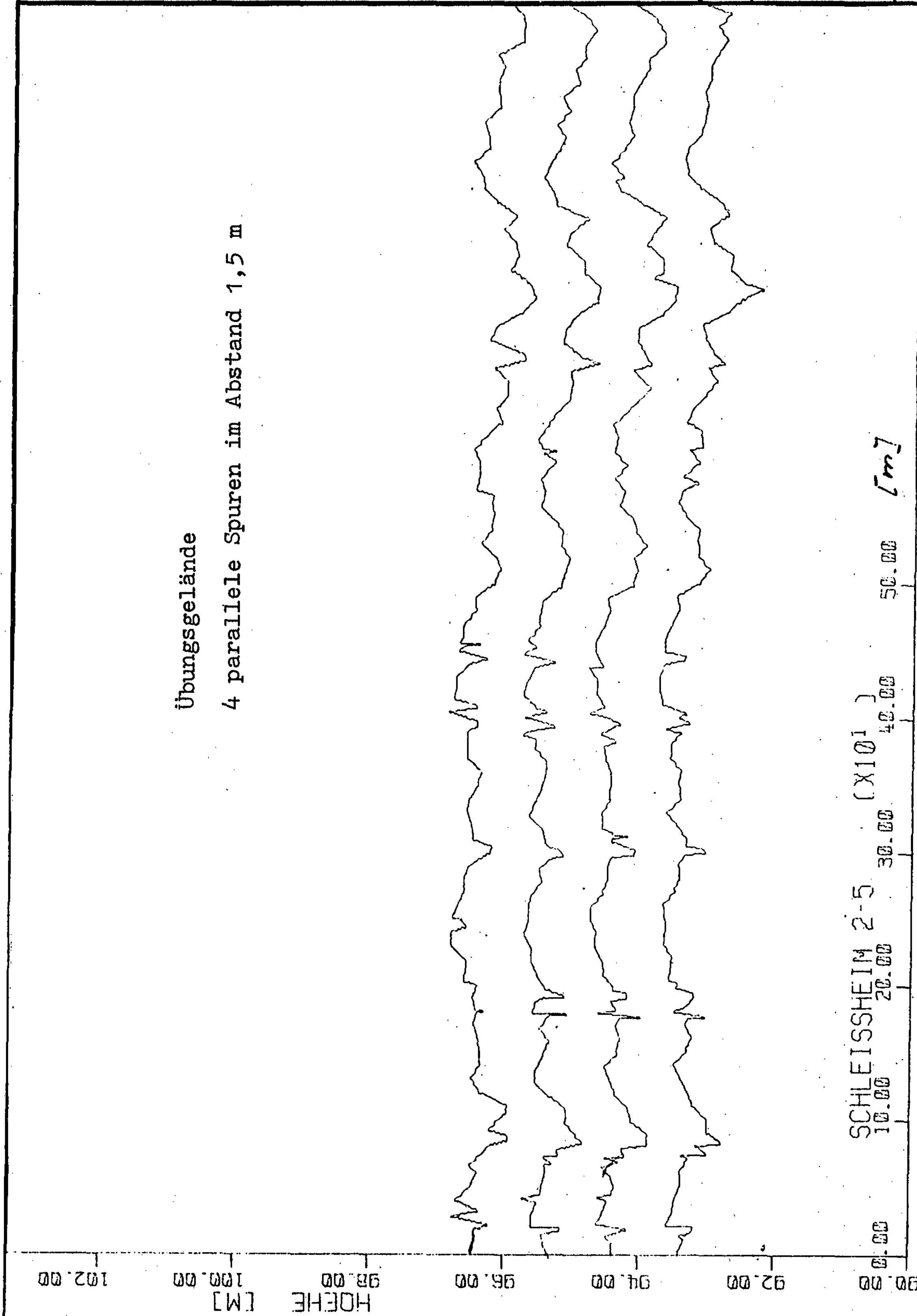
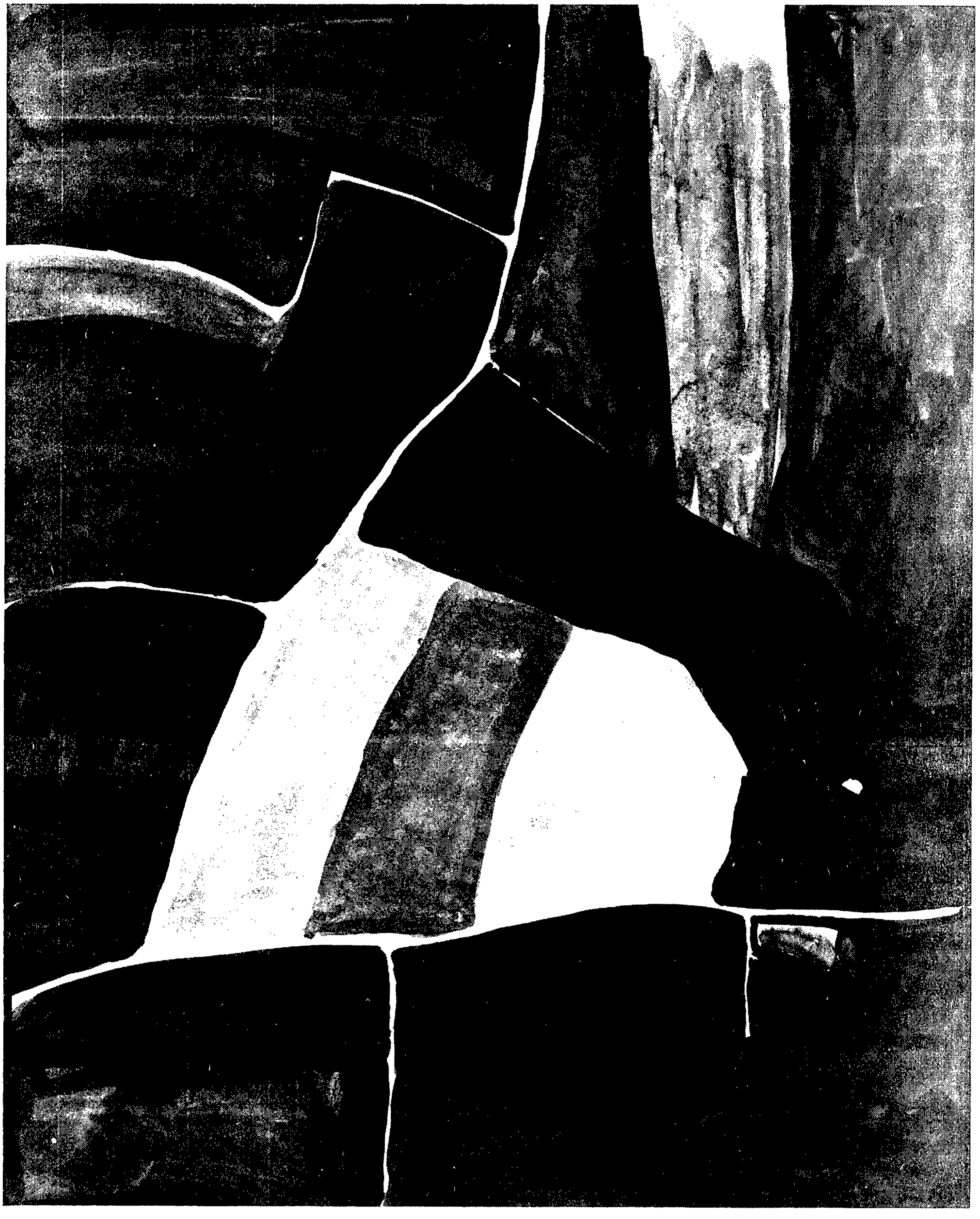


Bearbeiter

Datum

72



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Ottobrunn
TFM

Auftrag

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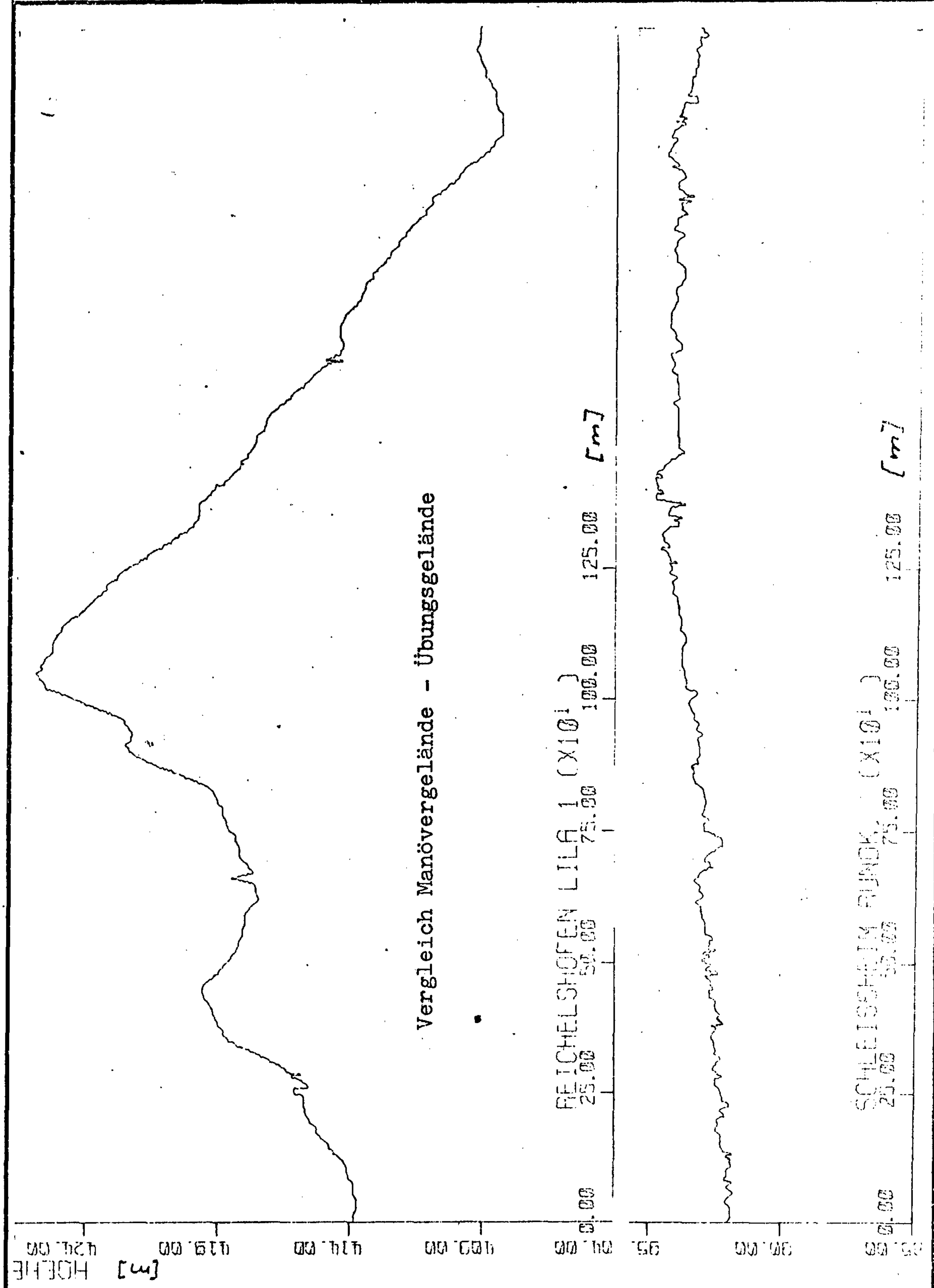

Bearbeiter

Datum

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IABG Ottobrunn TF		Auftrag
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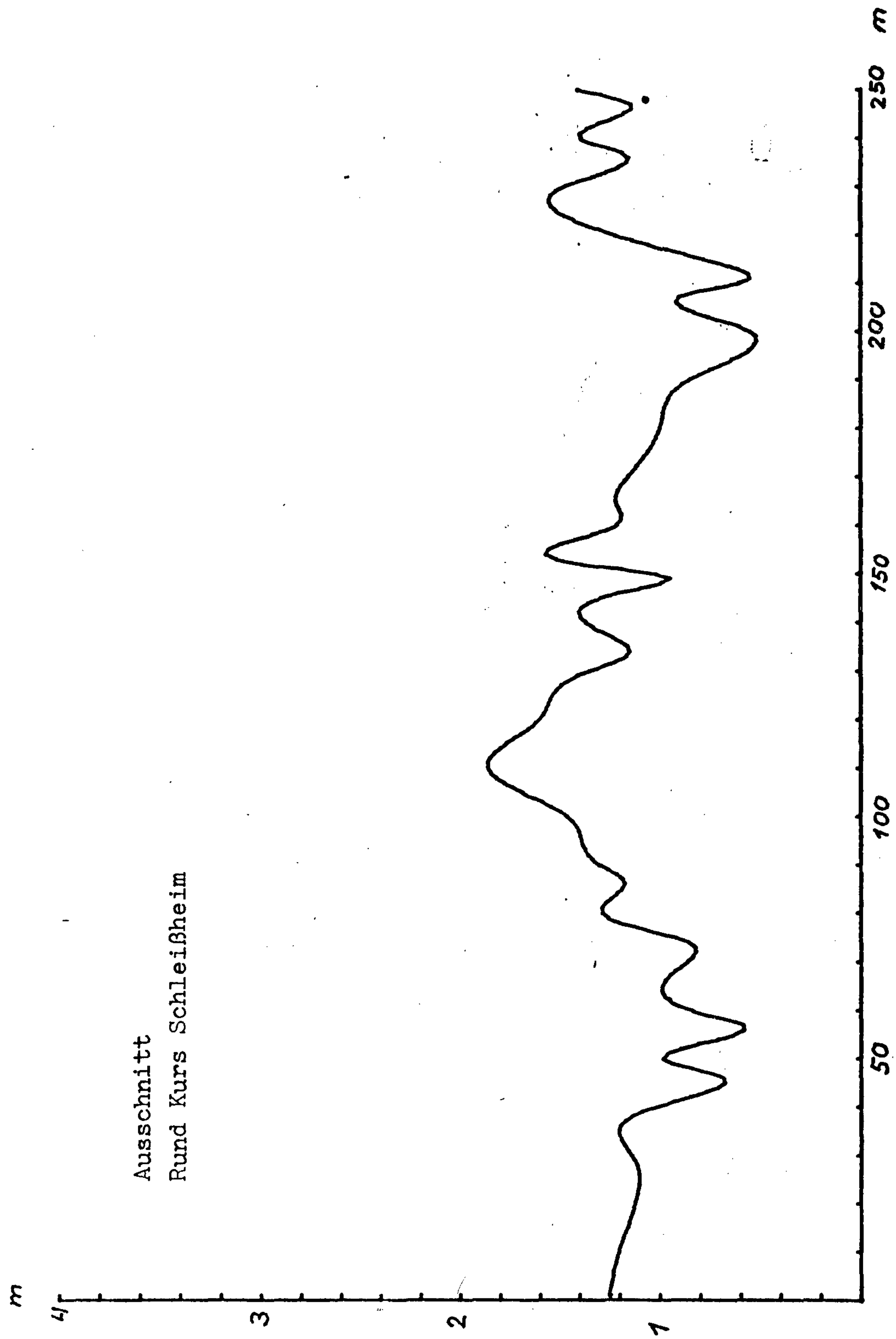

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TF

Auftrag

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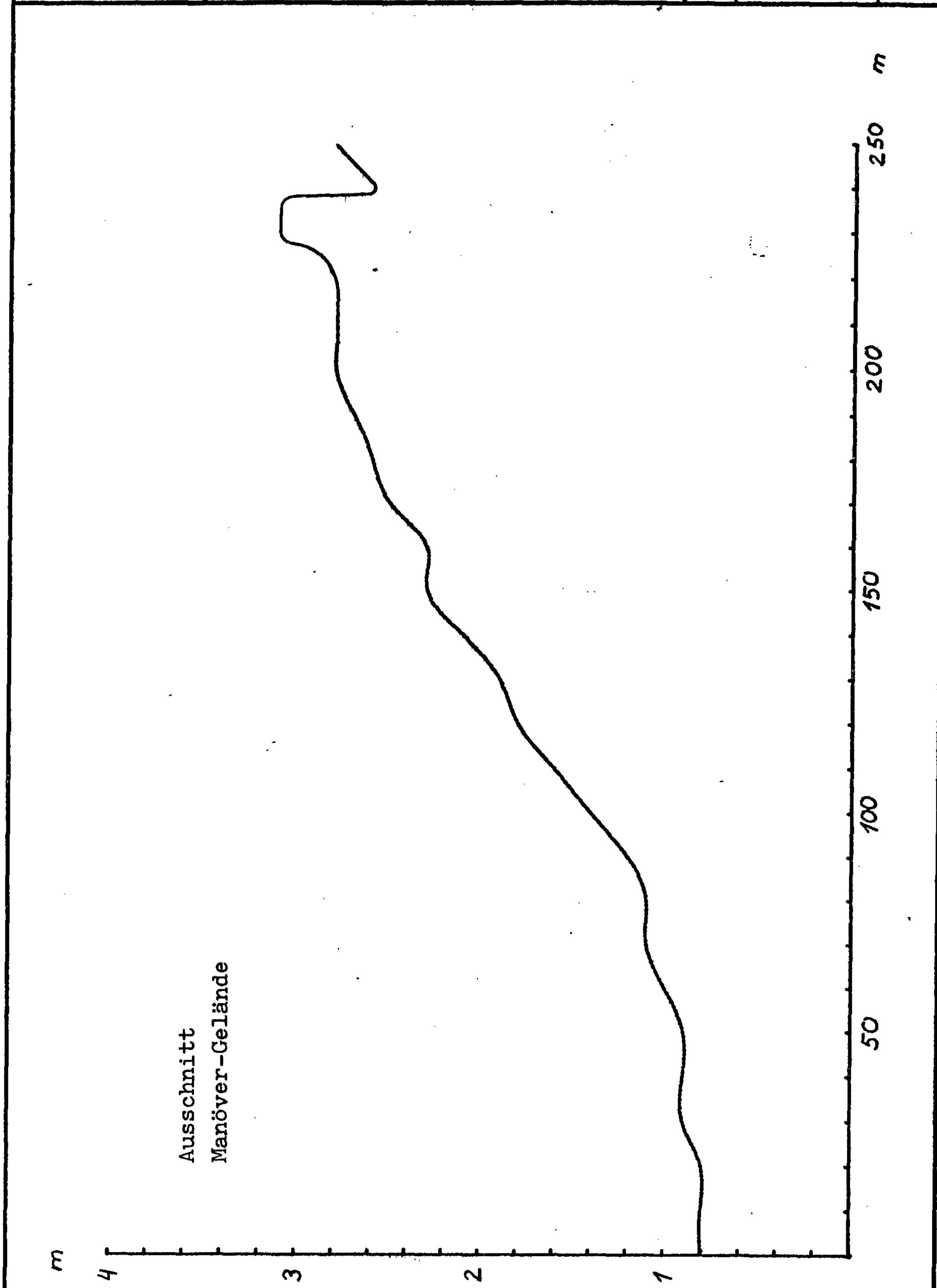


Bearbeiter	Datum 76		

20.92.1.1.

Measurement of Terrain Surface

The methods, just explained provide the micro terrain profiles to be used for digital or analogous tank simulation computations. But this description is not suited for a direct comparison of different profiles related to their roughness.

Therefore, in general, the power spectral density of the roughness is used. Figure 1 shows the power spectral density for some artificial terrains (roads, landing paths, proving ground "Belgisch Blockbahn") and for a cross country terrain (proving ground).

When making vehicle tests, we measured the ground roughness in the way described before.
Figure 2 shows some evaluations.

2. Vehicle related Roughness

The methods explained before, give only a geometric description of the terrain roughness. In many cases, this kind of data is sufficient. But as input data for computations we should have annother kind of roughness, the so alled "vehicle related roughness".

By the track, the soil is deformed plastically and elastically. This influence we cannot compute exactly enough. Therefore we tried to measure this roughness in a direct way. As measurement instrumentation we used the vehicle itself. By determining the vertical acceleration, the angle of piching or rolling it is possible to evaluate the power density of the roughness. For the evalution the frequency response of the vehicle used must be known. If this frequency response is unknown, the range of valuation will be limited. But it is possible, by comparison with other measurements to extrapolate the results.

Figure 3 shows the power spectral density of a proving ground evaluated from measured hull oszillations.

At present we only get the power spectral density of the terrain surface, not the amplitudes as a function of the distance. For our computer model we therefore are obliged, to get the terrain surface from a random generator with the power spectral density measured as explained before.

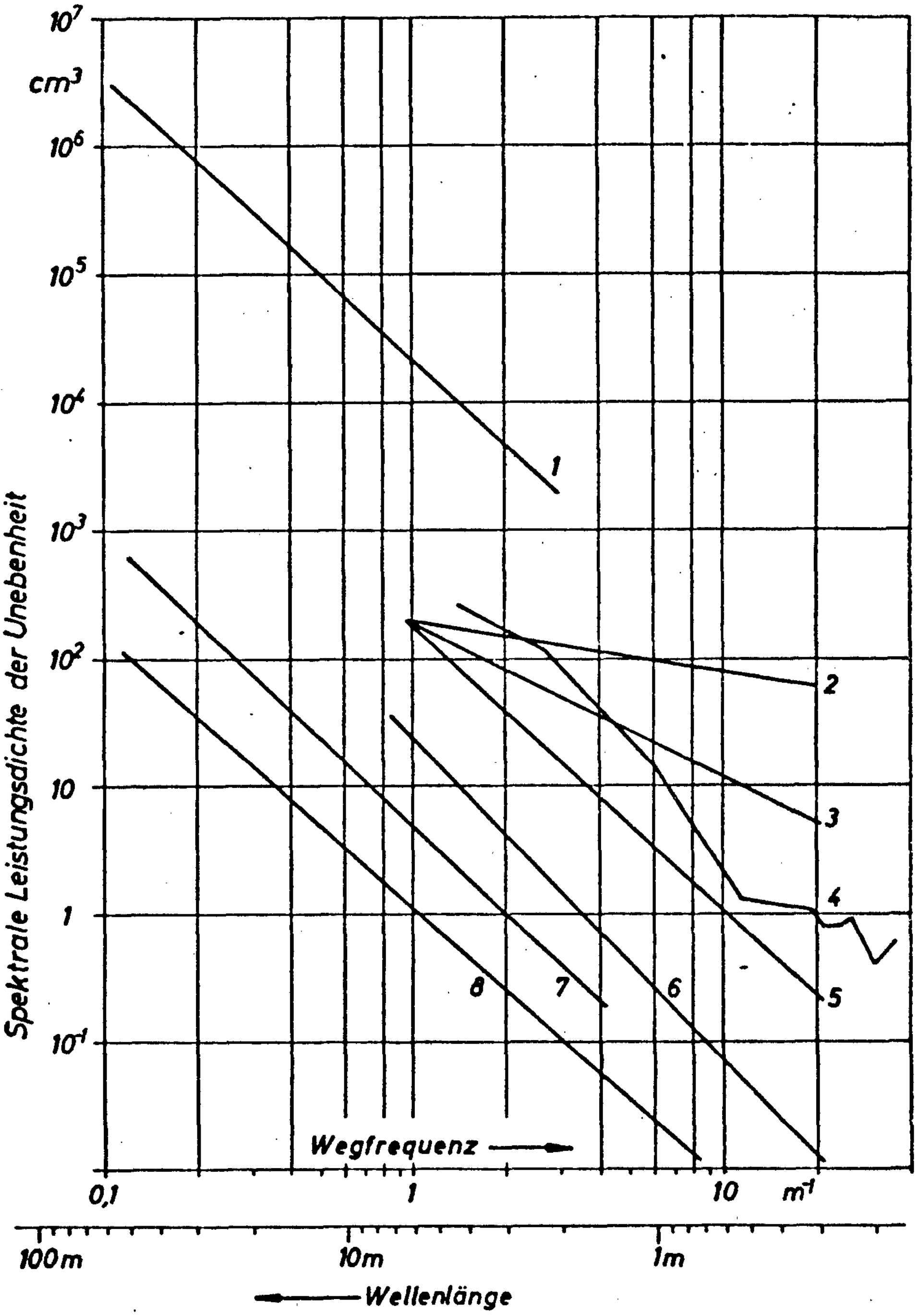


Bild 1 : Spektrale Leistungsdichten für verschiedene Gelände

1. Panzererprobungsgelände
2. Sturzacker
3. Furchenacker
4. Belgisch-Blockbahn
5. Feldweg
6. Landstraße
7. Schlechte Flz.-Landebahn
8. Gute Flz.-Landebahn

Bearbeiter	Datum		
Merklinghaus	25.11.1974/na		

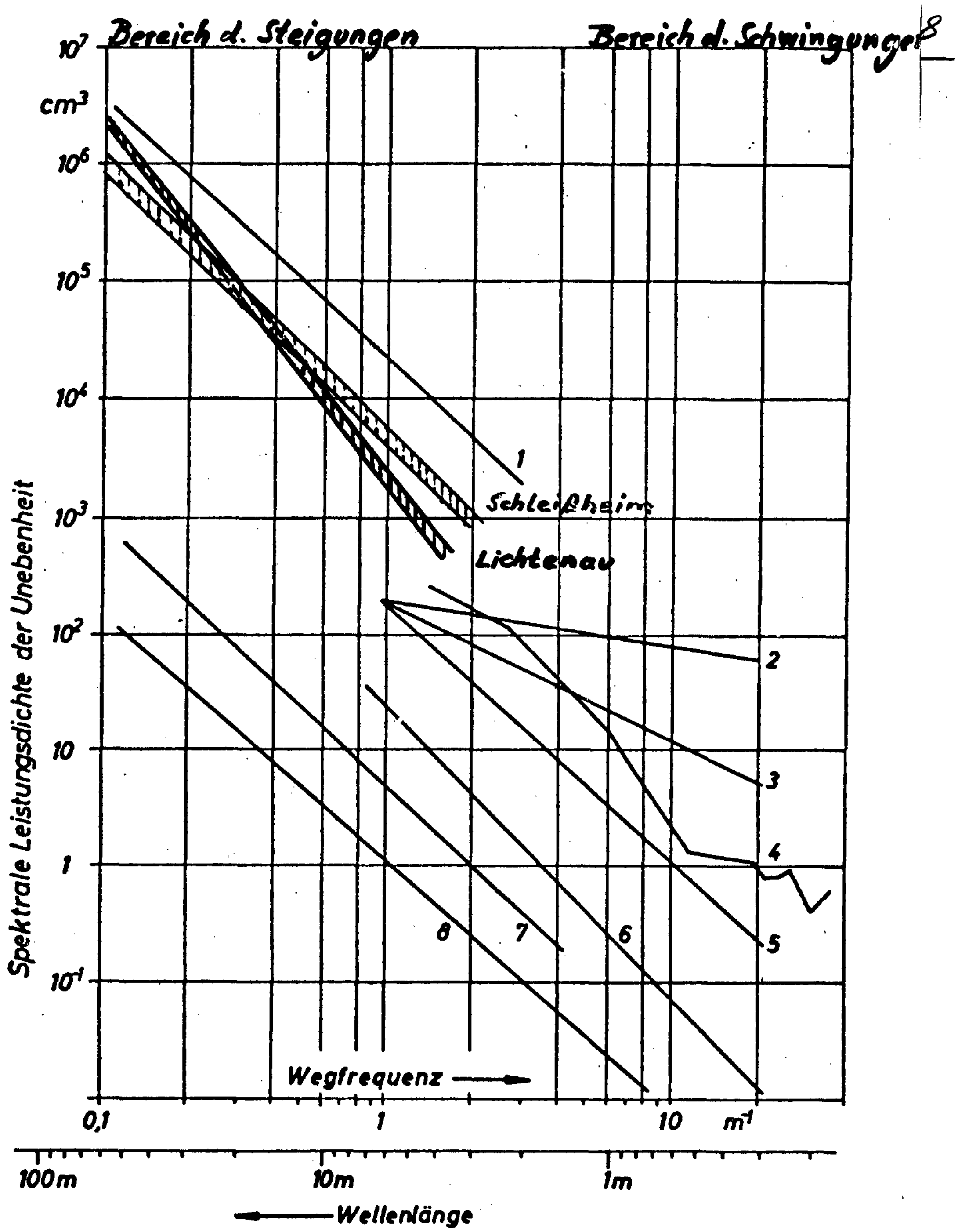
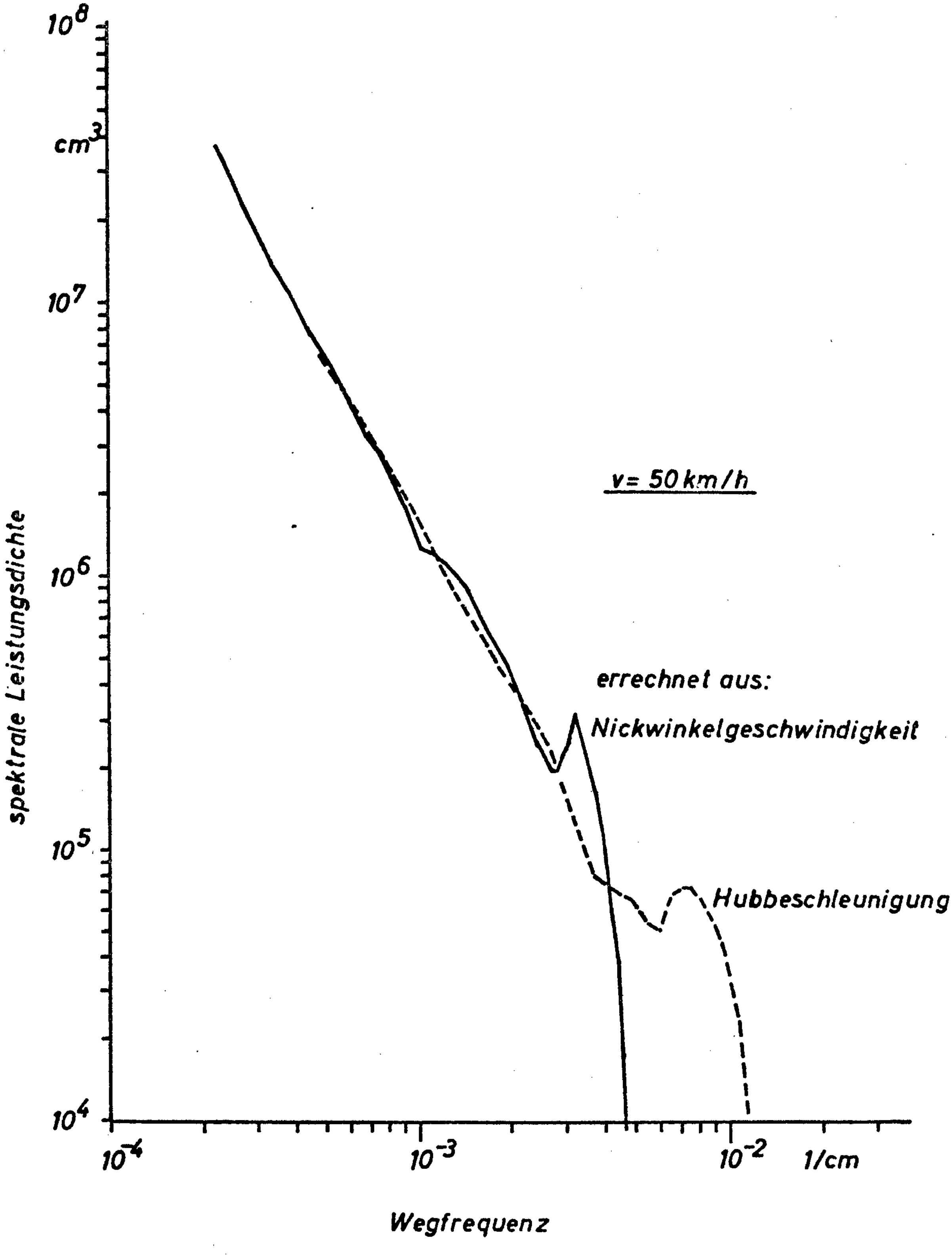


Bild 2: Spektrale Leistungsdichten für verschiedene Gelände

1. Panzererprobungsgelände
2. Sturzacker
3. Furchenacker
4. Belgisch-Blockbahn
5. Feldweg
6. Landstraße
7. Schlechte Flz.-Landebahn
8. Gute Flz.-Landebahn

Bearbeiter	Datum		
Merklinghaus	25.11.1974/na		



10^8
cm³
10^7
10^6
10^5
10^4
spektrale Leistungsdichte
v= 50 km/h
errechnet aus:
Nickwinkelgeschwindigkeit
Hubbeschleunigung
10^-4
10^-3
10^-2
1/cm
Wegfrequenz

Bild 3

CYBERNETIC APPLICATIONS TO VEHICLE CONTROL

Presented By

R. BECK
U.S.

SUMMARY

Two M-113 armored personnel carriers have been coupled with a controlled articulation joint incorporating force-feedback. This coupled vehicle has demonstrated an outstanding increase in off-road mobility.

By far the greatest improvements in performance, as compared to a single vehicle, have been in the water exiting, and the vertical obstacles and trench crossing capabilities. This is due to several factors: the greater momentum, compliance with the terrain, the ability to supplement with inter-vehicle forces those lacking in traction; all made possible because of the responsiveness of the hydraulic assist and the ease of control.

It is only in the area of maneuverability on relatively hard ground where the single vehicle, with its pivot steer capability, is superior to the coupled vehicle with its powered yaw articulation. But the coupled units' turning radius is still comparable to that of highway vehicles of the same overall length.

The coupled vehicle in its present form has great potential as an experimental test bed for further exploring its operational capabilities, for determining the influence of variations in the control system on its performance, and to gain information for the design of other applications.

To what degree the disadvantages of lesser steerability, maneuverability, the added complexity and weight, negate the demonstrated advantages, will have to be determined in further evaluations. The trade-off in performance reduction as a function of reducing the size and complexity of the hydraulic and control system need to be determined.

LIST OF FIGURES

INTRODUCTION

Vehicle articulation is not a new idea, it has been advocated by Bekker and Nuttall for about thirty years.

Studies with articulated vehicles have demonstrated that coupling and articulation produce significant gains in mobility and allow negotiation of many terrain features which are impassable to a single-frame vehicle. The improved performance can come either from the ability of the system to conform to the terrain or from the inter-vehicle assist available or from both.

Experiments with the Cobra (1)* (Figure 1), have demonstrated that a further improvement can be made by also controlling pitch articulation. However, exercising the additional control presents additional demands and information processing problems to the driver. Consequently, the emphasis of this study was to design an articulation system incorporating simplified controls and including force-feedback. The force-feedback control provides the driver with additional cues via his "sense of feel" to make his driving task easier, and to enhance the performance of the man-vehicle system.

Two production M-113 Armored Personnel Carriers were used as the basic vehicles for this study (2) (Figure 2). They were chosen for several reasons: they are readily available; they have an obviously apparent application; their use makes possible a direct comparison of performance with a single-unit counterpart; and using them also demonstrates the possibility of retrofitting a special purpose system on existing hardware.

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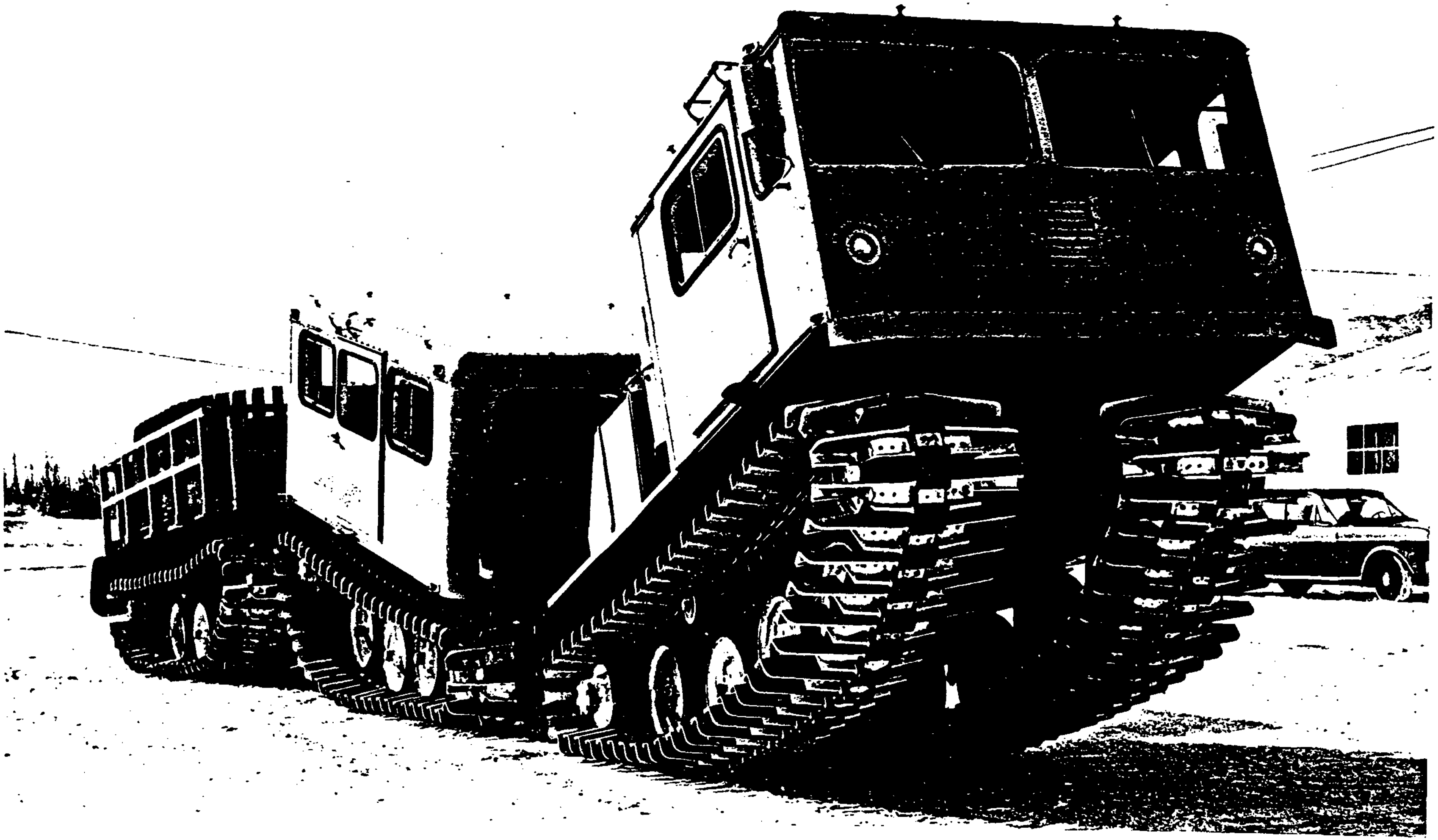
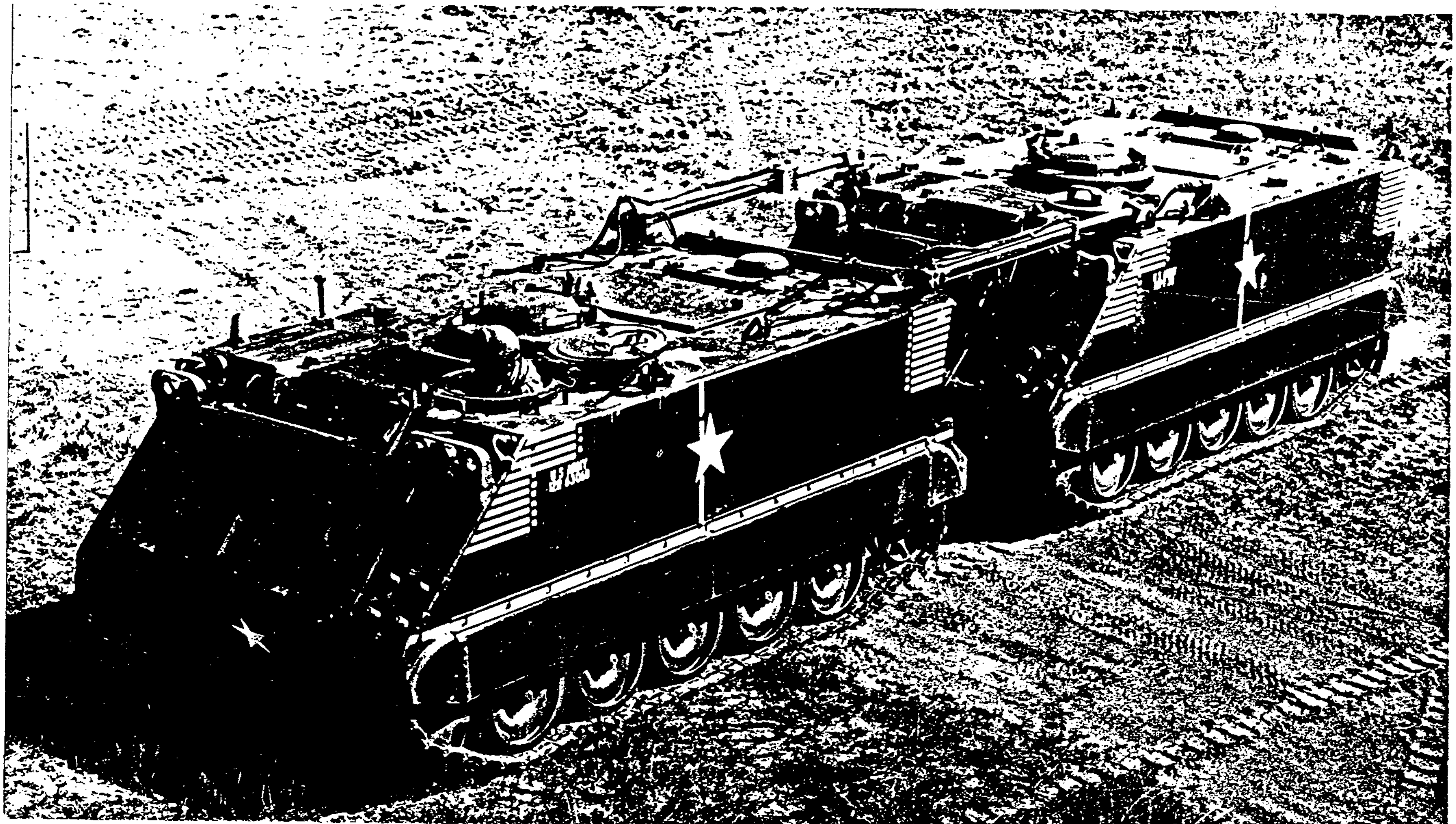
* Numbers in paranthesis denotes references cited at the end.



THE COBRA

FIGURE 1



THE COUPLED M-113's

FIGURE 2

The operational and performance specifications expected of the proposed system were (3):

1. Climb vertical steps up to five feet high.
2. Cross trenches up to ten feet wide.
3. Climb a sixty percent slope of fifteen feet in length with soil parameters: cohesion, C, equal to 1.0 psi and internal friction angle, ϕ, equals to 25°.
4. A turning radius of forty feet on hard ground.
5. Cross a two and one-half foot high obstacle at two and one-half miles per hour.
6. Operate over various adverse terrains.
7. Be able to be controlled from either front or rear unit.
8. Be able to enter into, cross and exit from inland waterways which represent a NO-GO situation for the single M-113.
9. Have force-feedback capability when negotiating vertical obstacles.

The principal objectives of this program were:

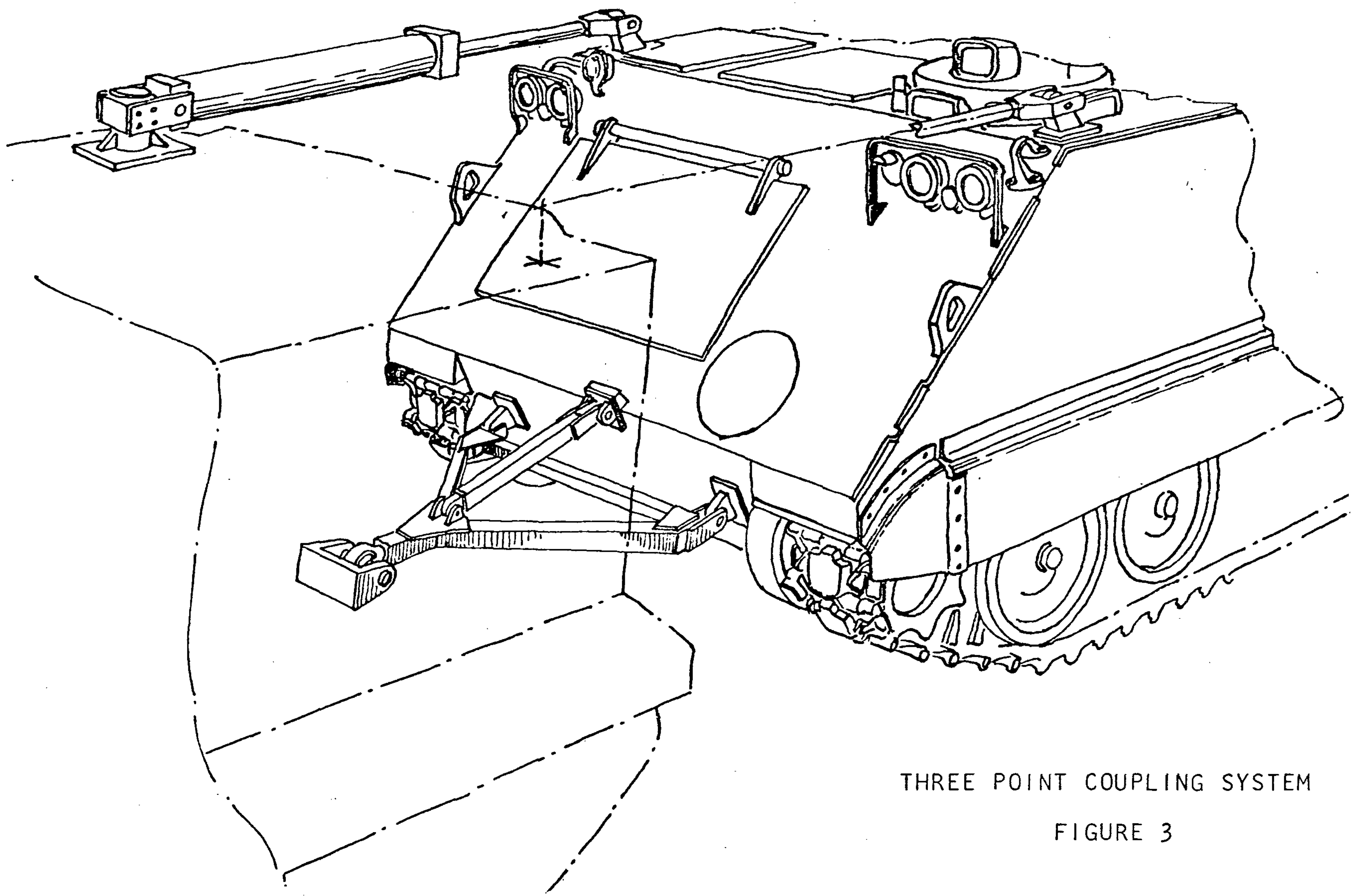
1. Compare the difference in mobility between the coupled units and a single vehicle (pro and con) especially in their obstacle and water crossing capabilities, and steerability vs. maneuverability.
2. Investigate the man-machine interaction and the potential of a controlled articulation joint between two identical vehicles employing force-feedback control.
3. Provide input data and a means of verifying the theoretical analysis and mathematical models of such systems.
4. Use the coupled units as a test bed in the study of articulated vehicle systems operating with or without force-feedback control.
5. Apply the theory established to the design of future vehicle systems employing articulation with or without force-feedback.

ARTICULATION CONTROL SYSTEM DESCRIPTION

The design basically consists of connecting the two vehicles by means of a ball joint allowing roll, pitch and yaw between the two units. The yaw and pitch motions are controlled by two hydraulic actuators. Roll motion is unconstrained. A "three point" coupling system (Figure 3) was designed because the rear ramp of the M-113 prevents the use of a conventional articulation joint, in which the joint and power cylinder are in one unit, and also because the larger moments thus realized help reduce the component size.

The articulation motion is generated by an electro-hydraulic servo system controlled by a single "joy-stick" lever located in the driver's compartment. Force-feedback to the "joy-stick" is provided for the pitch motion only. The engines and transmissions are synchronized by electro-mechanical servo systems. All the components are standard off-the-shelf commercial items with the exception of the drawbar and the actuator mounts and portions of the control and force-feedback system. The intergrity of each individual vehicle is not disturbed so that they can operate independently when uncoupled. The complete coupling or uncoupling process takes about 15 minutes.

The inter-vehicle connection basically consists of the two hydraulic actuators, the spherical ball joint and an umbilical cord. All rotation between the two vehicles takes place about the ball joint. The socket of the ball joint is mounted in the apex of an A-frame which is rigidly attached to the base of the rear vehicle, and the ball is connected to the forward vehicle (Figure 4). In order to counteract the nose-heaviness of the M-113, the ball joint was located as close as possible to the forward unit. This increases the moment arm between the pitch pivot point and the center of gravity of the rear vehicle, thereby increasing the lift of the forward vehicle during pitch articulation.



THREE POINT COUPLING SYSTEM

FIGURE 3

INTER-VEHICLE A-FRAME CONNECTOR

FIGURE 4

The two actuators mounted on the adjoining top corners of the vehicle provide the forces necessary for both pitch and yaw control. Pitch motion is generated by simultaneous contraction or extension (Figures 5 & 6); yaw motion by equally contracting one actuator and extending the other (Figure 7). The available pitch motion between units is 20 degrees up and 28 degrees down, yaw motion is 31 degrees in both directions.

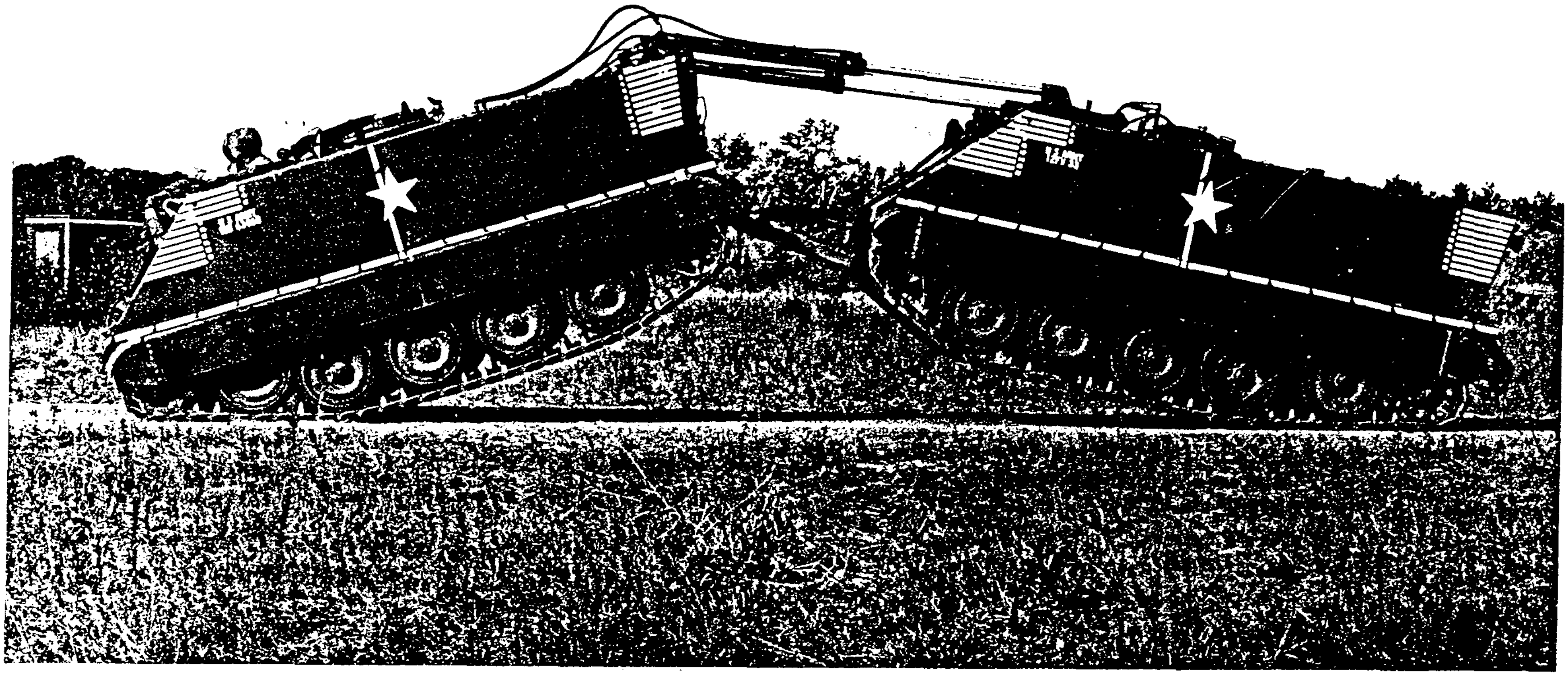
All hydraulic components and their controls are located in the forward vehicle. Each hydraulic actuator is powered by its own variable stroke piston pump. Both pumps are driven mechanically from the transfer case power take-off; pump speed is directly proportional to engine speed. The displacement and flow direction of each pump is controlled by an electronic servo control system.

Figure 8 is the block diagram of the electro-hydraulic control system, one for each pump/actuator combination. The driver controlled "joy-stick" produces a proportional command voltage, R(S), to the electronically controlled hydraulic system, resulting in hydraulic fluid flow, $Y(S)Q_p$, to the actuator $G_2(S)$. As the actuator piston is displaced Z(S) the positional feedback potentiometer voltage C(S) feeds back to reduce the signal E(S). When C(S) equals R(S), E(S) is zero, the pump yoke returns to zero stroke, Y(S) = 0, resulting in zero flow and the desired vehicle attitude is maintained.

Rearward movement of the control stick produces an identical command voltage to both the right and left control systems and causes both pumps simultaneously to displace fluid into the rod end of the actuator, producing a pitch up motion; similarly forward motion of the control stick produces pitch-down motion. Side to side motion of the control stick causes the pumps to deliver fluid to opposite ends of the actuators so that one extends and the other contracts. The control stick can also be moved in any oblique direction to produce simultaneous yaw and pitch motion.

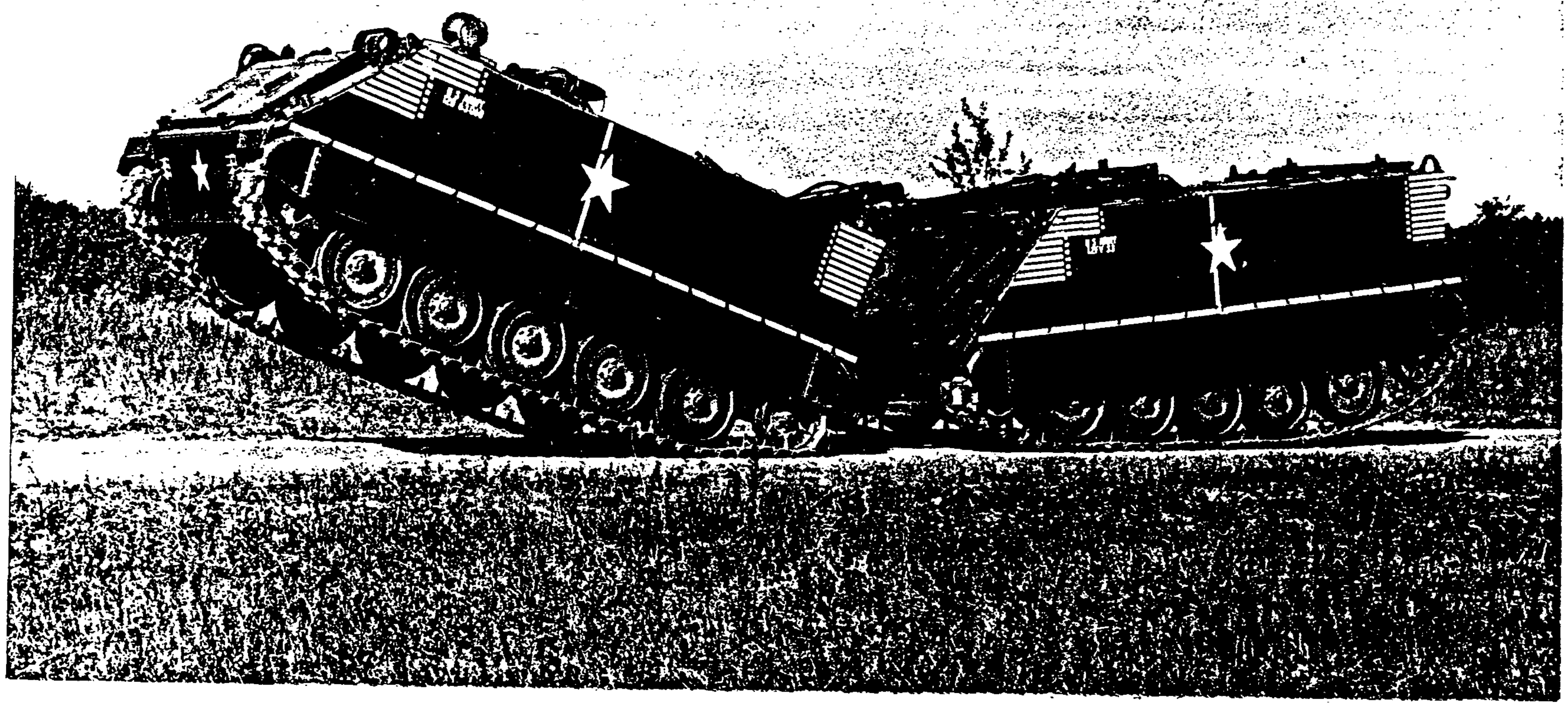
Figure 9 shows the layout of the control stick in the pitch motion plane.

Without force-feedback the control stick is mechanically self-centering by the action of springs K_1 and K_2 (connected to provide tension only). A



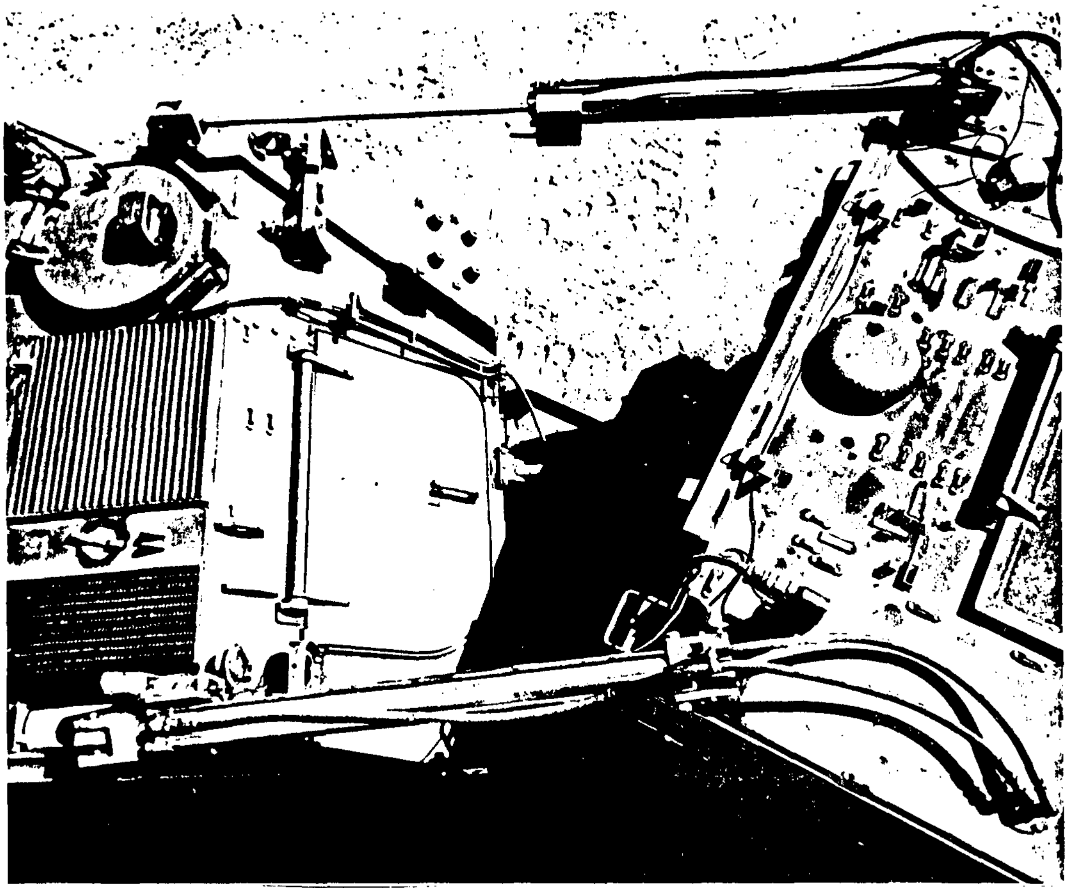
PITCH - DOWN ARTICULATION

FIGURE 6



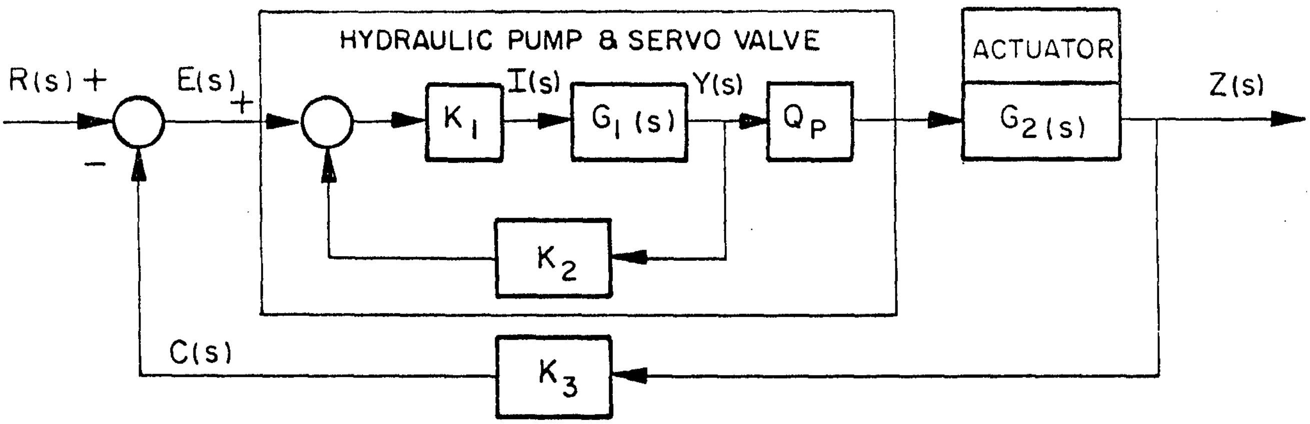
PITCH - UP ARTICULATION

FIGURE 5



YAW ARTICULATION

FIGURE 7



ELECTRO-HYDRAULIC CONTROL SYSTEM BLOCK DIAGRAM

FIGURE 8

where: $R(s)$ = Reference voltage signal from control stick potentiometer, volts

$Z(s)$ = Cylinder displacement, inches

$C(s)$ = Cylinder feedback voltage, $C(s) = K_3Z(s)$

$Y(s)$ = Pump Yoke displacement, % stroke

$I(s)$ = Amplifier current, ma

$E(s)$ = Error signal voltage, $E(s) = R(s) - C(s)$

K_1 = Amplifier gain, ma/volts

K_2 = Yoke potentiometer feedback gain, volts/% stroke

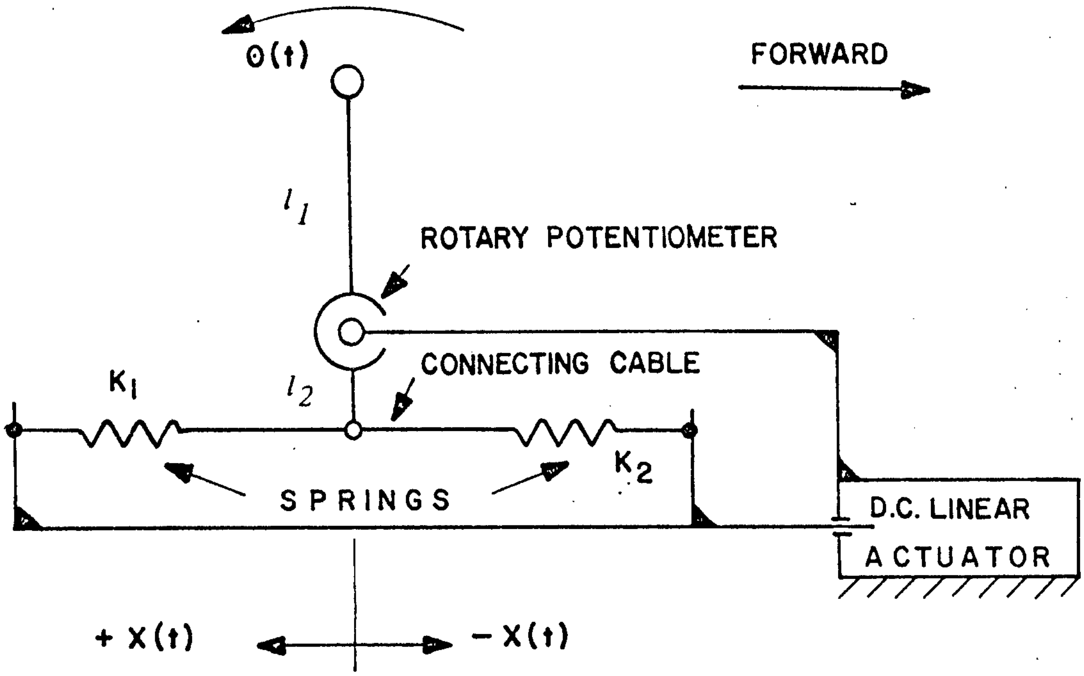
K_3 = Cylinder potentiometer feedback gain, volts/in

$G_1(s)$ = Transfer function of pump servo valve % stroke/ma

$G_2(s)$ = Transfer function of pump-cylinder/vehicle, in/GPM

Q_P = Pump flow gradient, GPM/% stroke (also a function of engine RPM)

s = Laplace operator



CONTROL STICK AND FORCE FEEDBACK ACTUATOR

FIGURE 9

zero position detent ensures that the vehicle will assume a zero pitch and yaw attitude when the control stick is released. When the control stick is moved to a + θ position, the forward unit will pitch up an amount corresponding to the voltage signal, R(S), generated by the rotary potentiometer. The driver feels only a small restoring force generated by the springs K_1 and K_2 proportional to the amount of deflection.

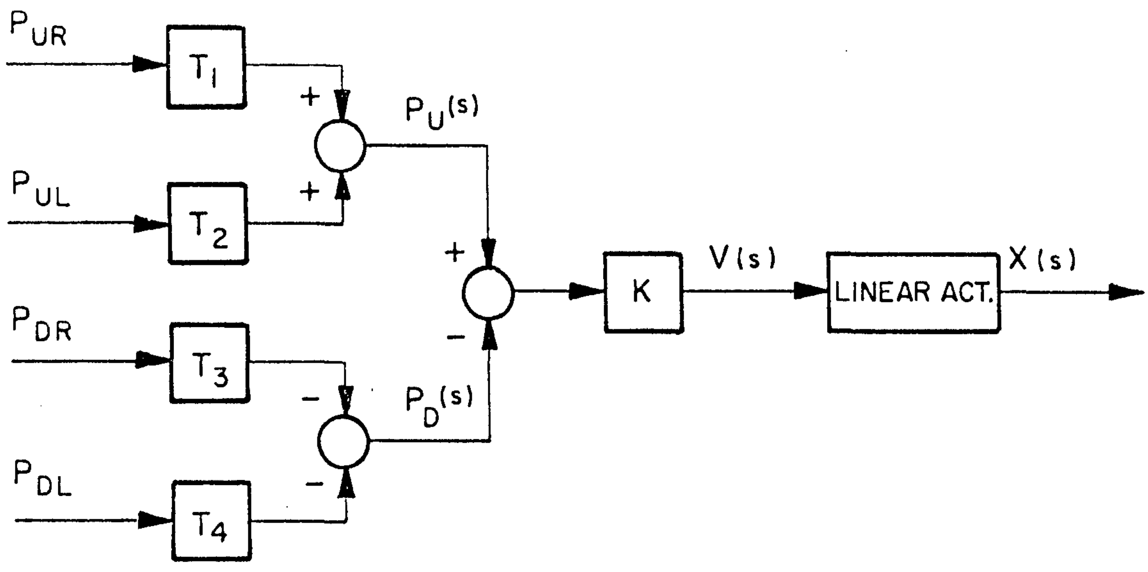
When the force-feedback system is activated, the linear actuator will move toward the left (as oriented in Figure 9) in response to a pressure in the pitch-up direction. As the springs, which are linked to the actuator, move in the +X(t) direction, the spring force K_1 increases and the driver will feel an increase in force:

$$\Delta F = K_1 \frac{\ell_2}{\ell_1} X(t).$$

Likewise, a pitch down force is generated in the opposite direction.

In order to maintain an inter-vehicle attitude, the driver has to oppose this force; if he releases the control stick it will move in the $-\theta$ direction, thus producing a command signal which will cause the hydraulic system to move the actuators until the inter-vehicle forces are zero. Thus, in a hands-off mode, the vehicle will always assume a "zero-pitch-force" attitude with the force-feedback system active; with the force-feedback inactive it will assume a "zero-pitch-angle" attitude.

The inter-vehicle forces are reflected by the pressure in the hydraulic system. Figure 10 is a simplified block diagram of the force-feedback servo system generating the control signal V(S) required to cause the linear actuator to displace $\pm X(S)$. The electrical outputs of the right and left system pressure transducers are added in the pitch-up or the pitch-down direction. This combined electrical output corresponds to $P_U(S)$ and $P_D(S)$ which represent the sum of the two individual pressures required for pitching up and down respectively. Simultaneous $P_U(S)$ and $P_D(S)$ corresponds to yaw motion and are cancelled electrically (note polarities). The electrical signal, $P_U(S)$ or $P_D(S)$, is then amplified and transmitted to the electro-mechanical linear actuator, thereby creating the positive or negative displacement X(S) respectively.



FORCE FEEDBACK BLOCK DIAGRAM

FIGURE 10

where:

P_{UR}, P_{UL}	=	Pressure generated when pitching up right and left cylinders respectively, psi
P_{DR}, P_{DL}	=	Pressure generated when pitching down, right and left cylinders respectively, psi
P_U	=	Combined pressure pitching up, volts
P_D	=	Combined pressure pitching down, volts
K	=	Amplifier gain, volts/volts
$T_{1,2,3,4}$	=	Pressure transducers, volts/psi
V(s)	=	Control signal voltage, volts
X(s)	=	Actuator displacement, in

POWER CONTROL

The engine and transmission of the rear unit are synchronized with those of the front unit*. This was accomplished by installation of position sensors (potentiometers) on the governor and transmission linkages and using them to produce a voltage signal to the DC linear servo actuators installed on the engine governor and transmission linkages of the rear unit. Starting interlocks and engine safety cut-offs are included in the engine and transmission controls.

PERFORMANCE

The performance evaluation of the coupled vehicles in comparison to a single M-113 covered the following aspects:

1. System capability and performance
2. Drawbar pull in sand
3. Drawbar pull in snow
4. Cross country ride evaluation
5. Negotiation of sand slopes
6. Maneuvering on land and in water
7. Water speed evaluation
8. Exiting from deep water
9. Negotiation of rigid step obstacles
10. Crossing vertical wall trenches

All tests were performed near Houghton, Michigan with the support of the U.S. Army Keweenaw Field Station. The vehicles were ballasted to their combat load of 23,000 lbs. each and trimmed to the proper center of gravity location.

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*In case the vehicle is driven from the rear, the driver control console is moved to the rear vehicle and the front unit is slaved to the rear.

1. System Capability and Performance

The control system can be selectively operated in several modes:

Mode 1. Positional Control only, without force-feedback.

Mode 2. Positional Control with the force-feedback system active.

Mode 3. By-pass valves between the two actuator ends are open to such a degree as to permit yaw control of the vehicle but allowing a certain amount of pitch freedom with the valve acting as a motion damper. In this mode, only partial inter-vehicle forces can be transmitted. As a variation of this mode the by-pass line can be completely opened, without damping restrictions, which results in complete freedom to pitch or yaw, but of course without any hydraulic control, the only control resulting from the use of the laterals.

The gearing of the pumps to the engine is such that maximum allowable pump speed is not exceeded at governed engine rpm. The best torque point and general range of operating rpm are at 65 and 75 percent governed engine speed respectively. Furthermore, the crossing of most obstacles does not require operation at full torque so that the engine may be operating at as low as 50% governed speed. As a consequence, the pumps are usually only delivering only about 50 - 75 of rated flow even at full stroke. Also, the nature of the control system is such that as the actuator movement produced feedback signal increases, the net input signal to the control system is reduced continuously until it is nulled at the equilibrium position. Therefore, the actuator only sees an average flow which is roughly one-half that of an open loop system (without the position feedback potentiometer). The difference in the transient response of the pump yoke, system pressure and the actuator displacement between an open loop and a closed loop system are shown in Figure 11 a-d. The open loop system (11a and 11c) is of academic interest only because it cannot relate vehicle position to control stick position.

At about 75% governed engine speed the times to reach full up and full down pitch from a level attitude are two and four seconds respectively; maximum yaw takes five seconds, either direction, from a straight ahead

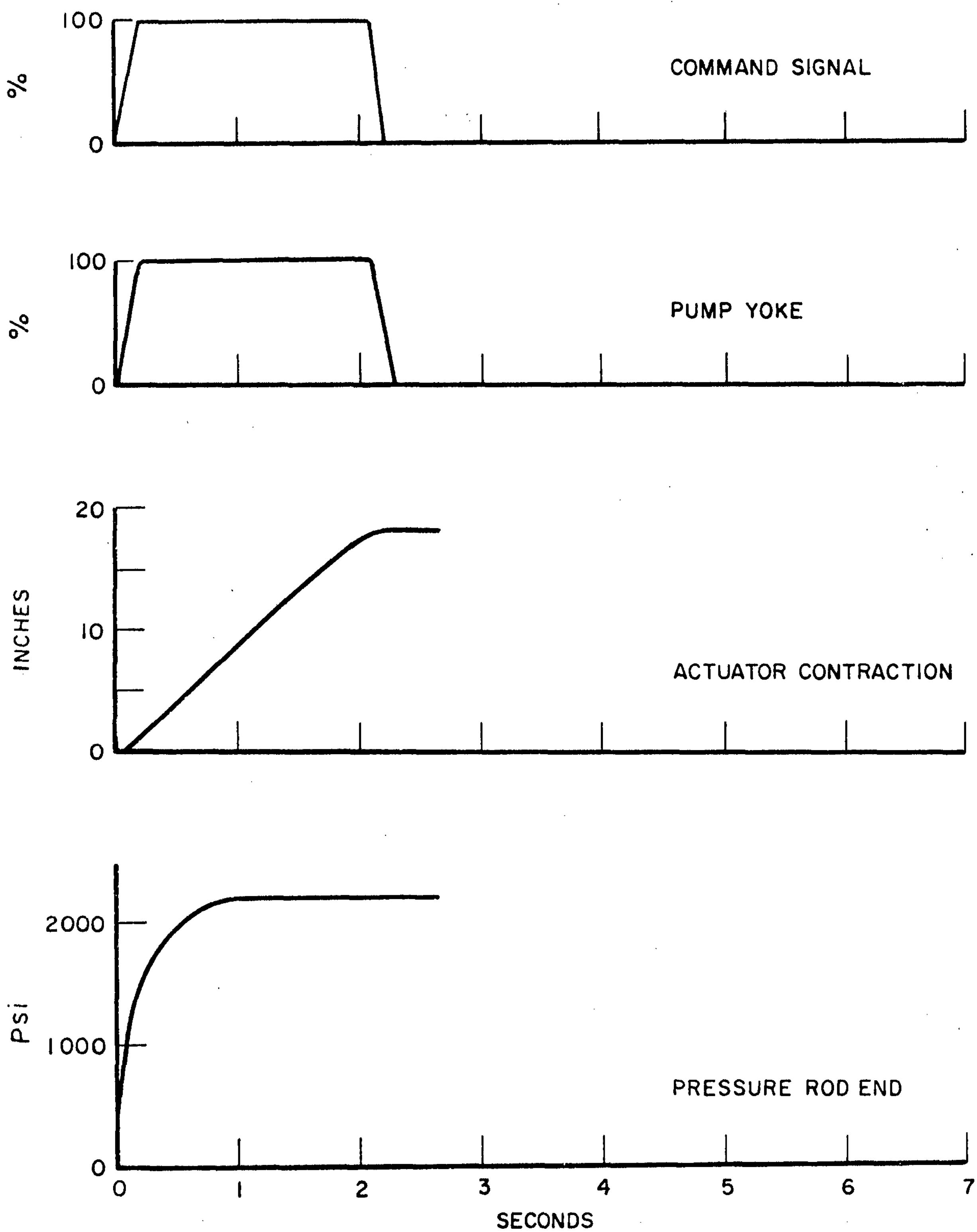


FIG. 11a. TRANSIENT RESPONSE OPEN LOOP
TEST 18 PITCH-UP 1500 ENG. RPM

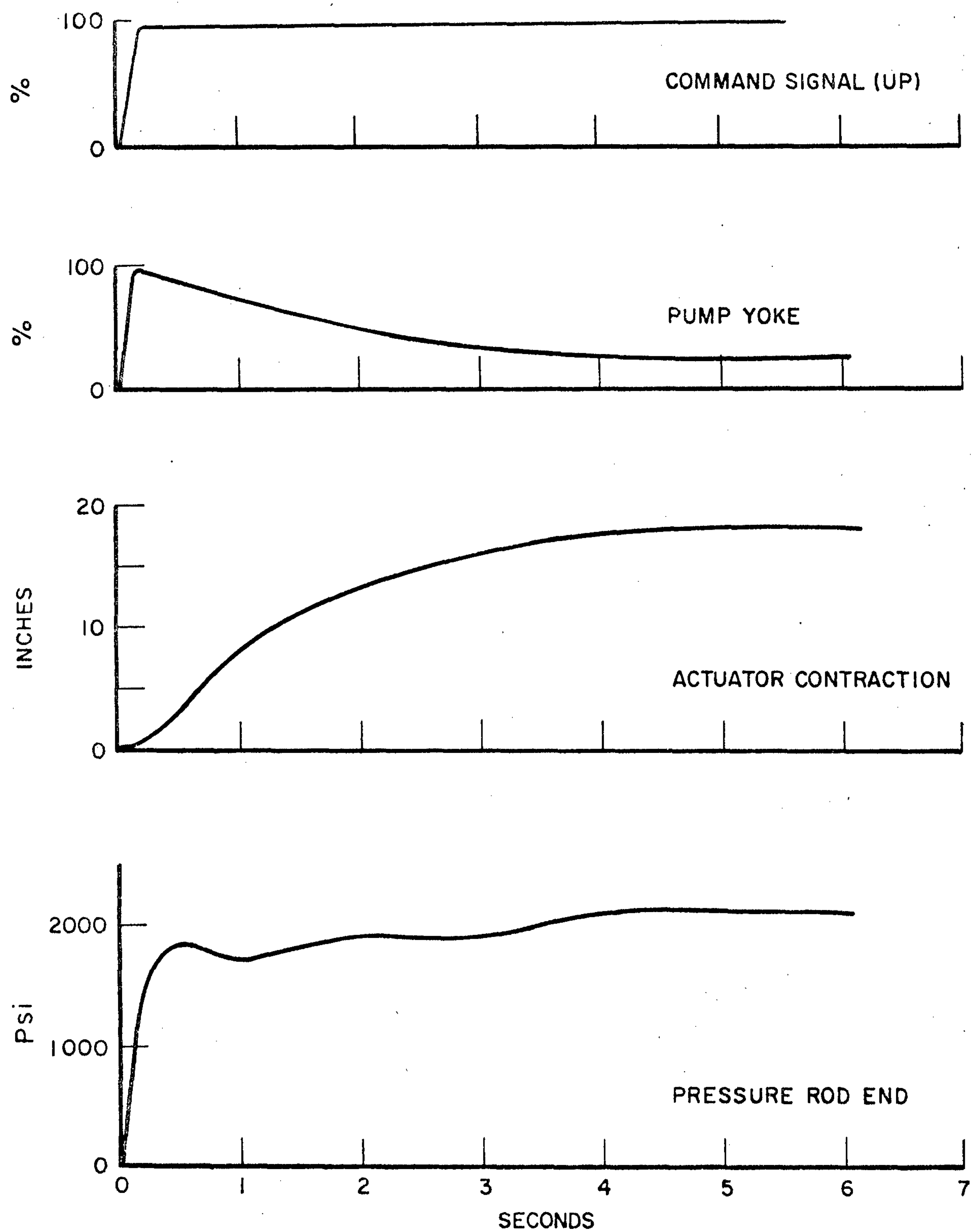


FIG. 11b. TRANSIENT RESPONSE CLOSED LOOP
TEST 16 PITCH-UP 1500 ENG. RPM

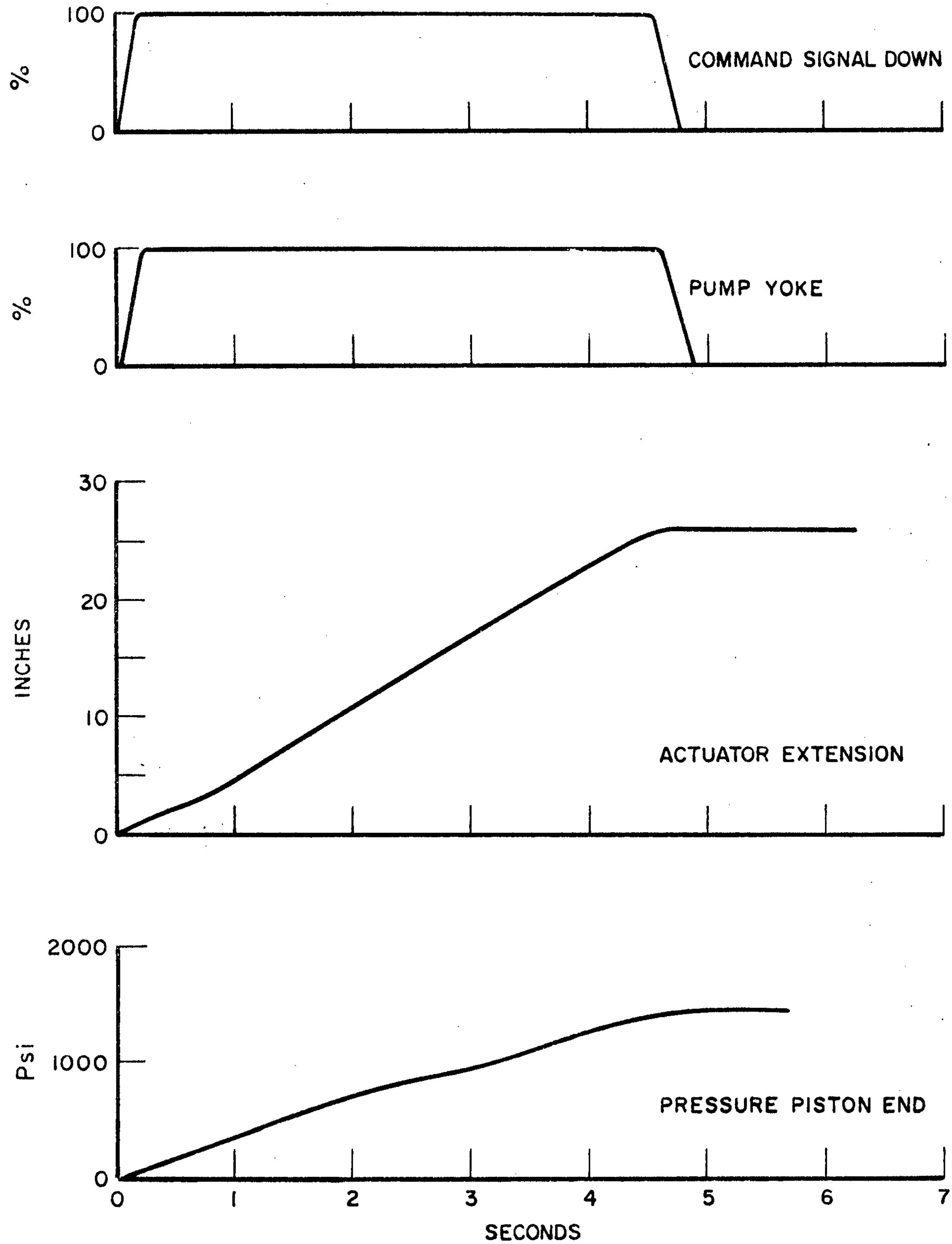


FIG. 11c. TRANSIENT RESPONSE OPEN LOOP
TEST 18 PITCH DOWN 1500 ENG. RPM

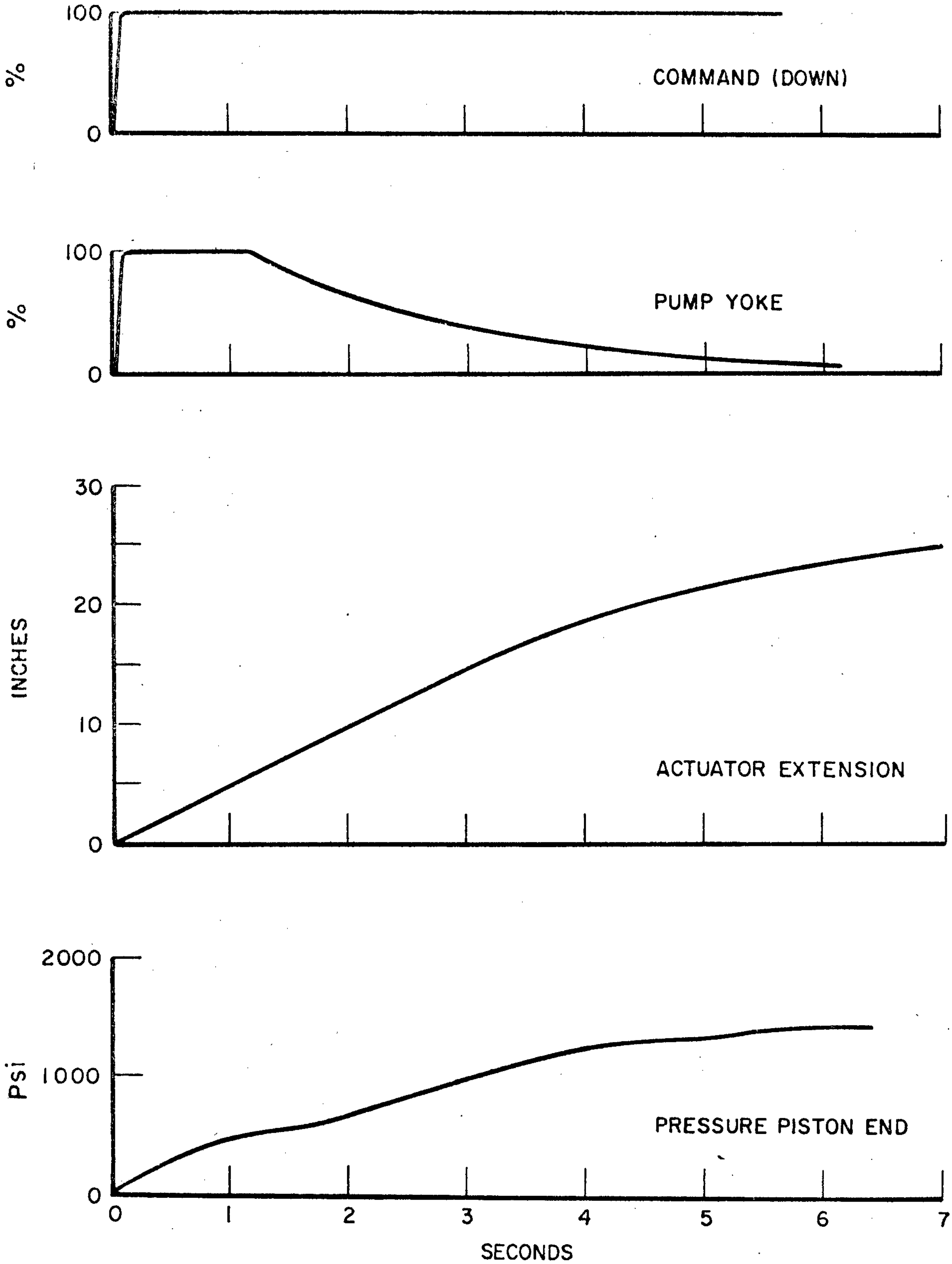


FIG. 11d. TRANSIENT RESPONSE CLOSED LOOP
TEST 16 PITCH-DOWN 1500 ENG. RPM

attitude while the vehicles are stationary on hard ground. The hydraulic pressure to maintain pitch-up attitude is 2,100 psi each actuator, to hold pitch-down is 1,500 psi., this difference is due to the differential piston area of the double ended actuators and to the assymmetry of the forces developed between the two attitudes. The differential piston area does not adversely affect the yaw articulation motion.

2. Drawbar Pull in Sand (4):

The tests were conducted in stamp sand, the tailings of copper mining operations found along the shore of Lake Superior near Gay, Michigan. This stamp sand has an angle of internal friction of 34 degrees and cohesion equal to 0.

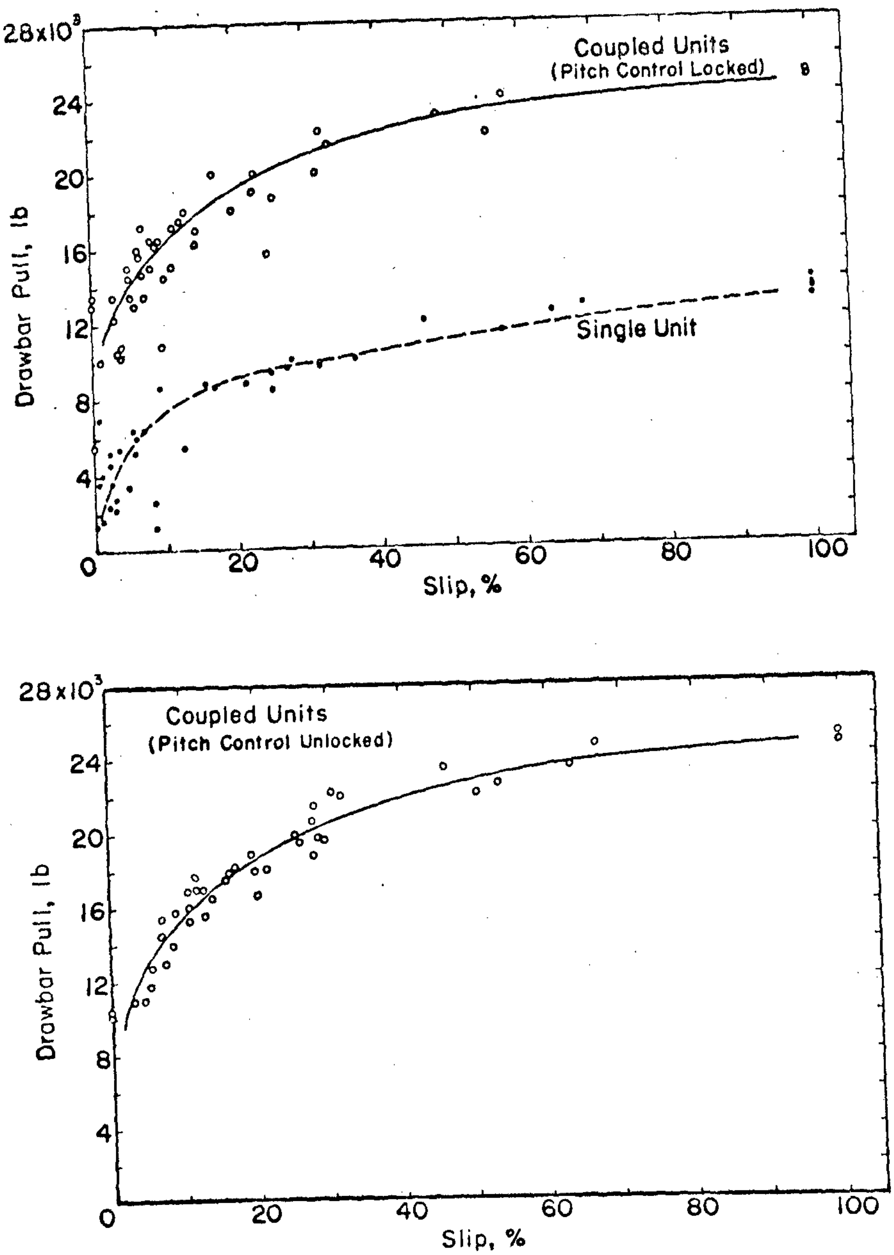
Figure 12 shows the drawbar pull in the active (Mode 1) and passive (Mode 3) conditions, in comparison with that of the single unit.

The coupled units exhibited approximately twice the drawbar pull of the single unit (with, of course, twice the weight). Drawbar pull with pitch control active is only marginally better than with it passive so that the real difference in operation will probably go unnoticed. All tests were run at 1,800 engine rpm, approximately the peak engine torque point. In all tests the forward unit was traction limited (high slip) and the rear unit torque limited (track stalled) (with both engines exactely rpm speed synchronized).

It appears therefore, that in a straightline drawbar pull test, there is no appreciable net gain in tractive effort due to the coupling arrangement.

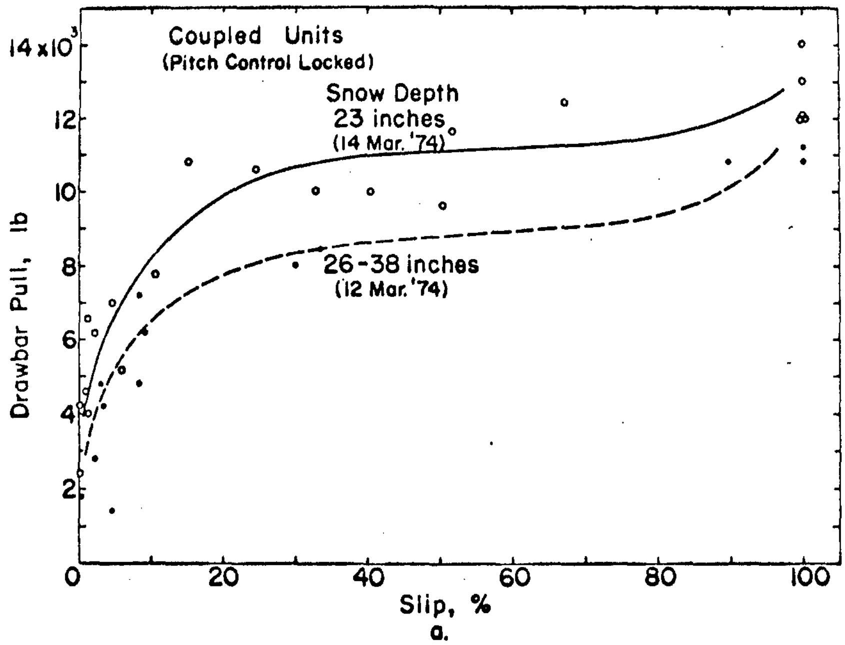
3. Drawbar Pull Tests in Snow (4):

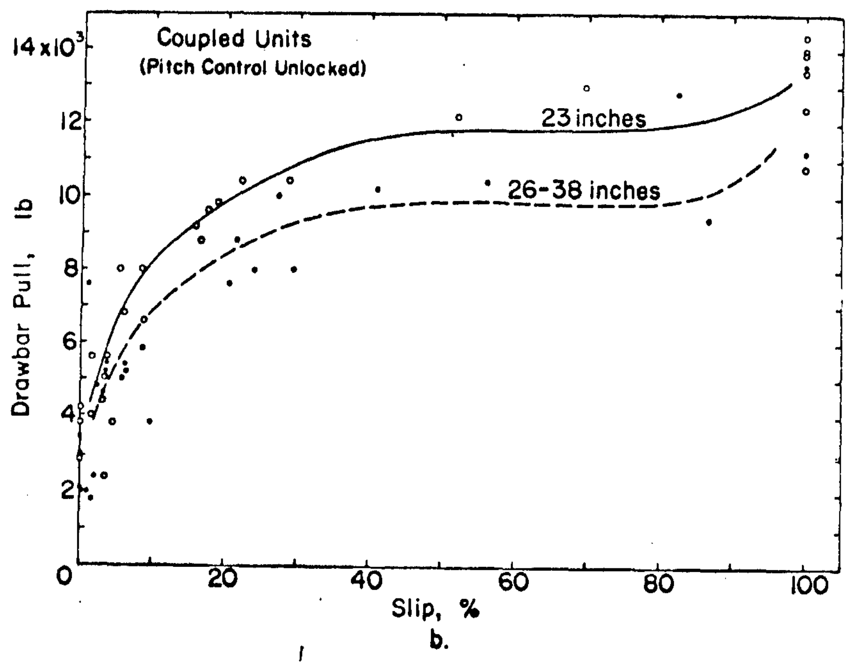
The snow tests were run near the U.S. Army Keweenaw Field Station in snow depths of 23 and 38 inches. As Figure 13 shows, the drawbar pull in the three control modes is again only so marginally different as not to be detectable in real operations. In contrast to the tests in sand, both units were traction limited.



DRAWBAR PULL-SLIP RESULTS IN STAMP SAND (4)

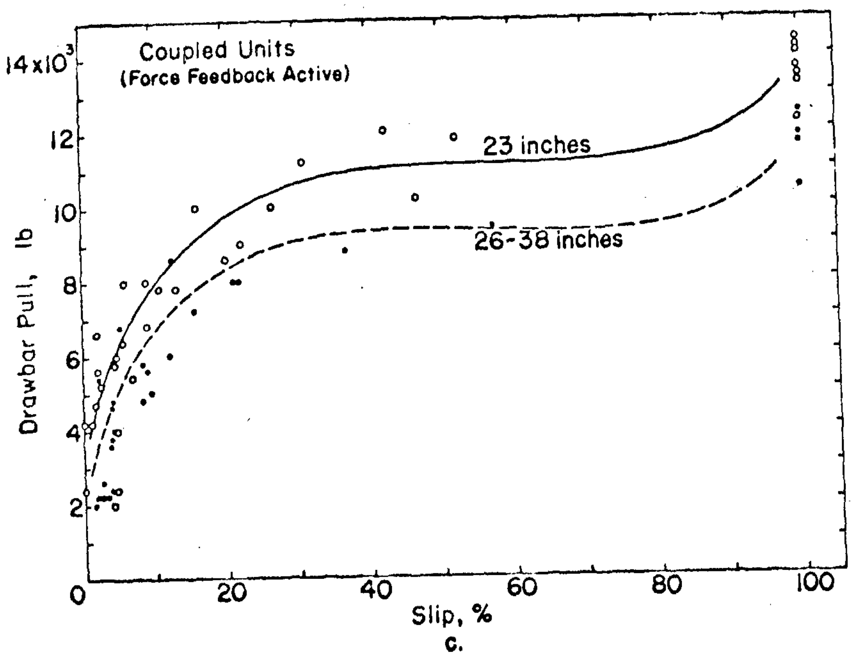
FIGURE 12

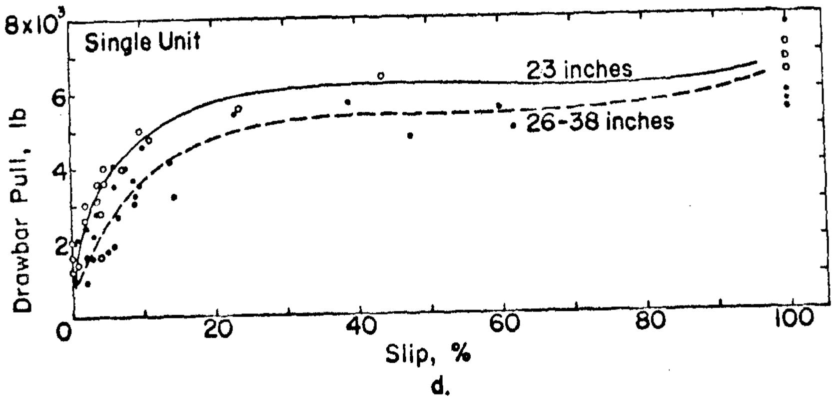




DRAWBAR PULL-SLIP RESULTS IN DEEP SNOW (4)

FIGURE 13





DRAWBAR PULL-SLIP RESULTS IN DEEP SNOW (4)

FIGURE 13

The properties of the snow measured in the track averaged $\phi = 25^{\circ}$, $C = 0.1$ psi and $k = 2$.

4. Cross Country Ride Evaluation (5):

A cross country ride and speed evaluation was conducted in the summer of 1973, during the debugging process. At the time of the tests the force feedback system was not operational, therefore, tests were only conducted in Modes 1 and 3.

Since power absorbed by the drivers body is the main speed limiting criterion on rough terrain, a comparison is made on the basis of an acceptable power absorption level of 6 watts in the vertical direction, and at the driver seat (see reference 6).

The summation of data shown in Figure 14 shows that the coupled units could be driven 50 percent faster in Mode 1, and twice as fast in Mode 3, over a course having an RMS roughness of about 2 inches. However, since the vehicle is less maneuverable in Mode 3, more extensive tests must be performed in order to evaluate the tradeoffs fully.

Later, general cross-country operation using force-feedback control showed that the vehicle pitch motion responds to external force inputs created by obstacles and vertical terrain-vehicle interaction. However, due to the slow (low frequency) response of the force-feedback servo system, it will do so only at relatively low speeds. On relatively large terrain features such as a 2 ft. high step, it will respond up to about 5 mph. For terrain features encountered in normal cross-county operation, the force feedback control system will follow the hydraulic servo system up to approximately 15 mph. At greater speeds the force-feedback system is non-responsive, and the operation reverts back essentially to Mode 1. However, since the driver can anticipate large features and override the force-feedback, the pitch motions of the coupled units are much less and the general ride, subjectively evaluated, is significantly better than that of the single vehicle for the same speed.

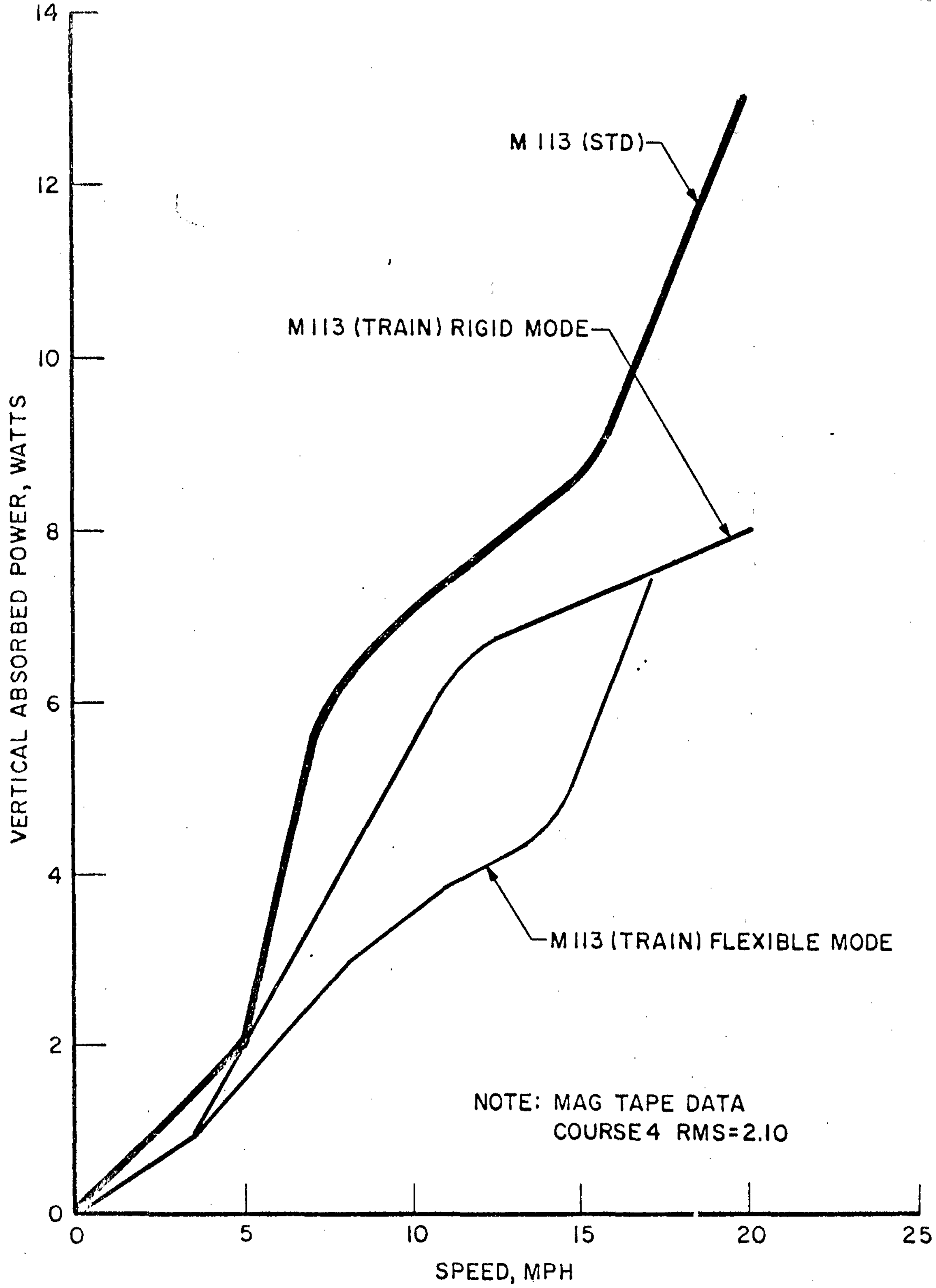


FIG. 14. ABSORBED POWER VERSUS SPEED
STD M113 & M113 (TRAIN)
HOUGHTON TESTS (SERIES I)

5. Performance on Sand Slopes:

The slope tests were conducted at Gay, Michigan in the same sand ($\phi = 34^{o}$, $C = 0$), used in the drawbar pull test. Tests were performed on short slopes of 60% and long slopes up to 53%. On short sand slopes, up to $1\frac{1}{2}$ single vehicle lengths, the coupled units will negotiate a 60% grade, because of its added momentum and because the unit on the level assists the unit on the slope. The importance of the ability to control the pitch motion smoothly and quickly enough not to lose momentum is very obvious during such a maneuver. By comparison the single unit was immobilized on this same slope.

On the longer slope both the coupled and the single units could negotiate a 45% grade. But on a 53% slope, the coupled units proceeded 56 feet before being immobilized whereas the single unit only 40 feet, the difference being that of one vehicle length. As in the drawbar tests, the forward unit is traction limited whereas the rear unit is torque limited. At the point of immobilization the front track spins and excavates a large amount of sand rearward creating an unsurmountable obstacle for the rear unit. It is conceivable that independent power application of the two units could be of advantage. Such a mode of operation, using two drivers, should be explored in the future.

It has also been found that in general cross-country operation the lack of the full braking capability may be a detriment. The laterals which are usually used for braking can only be used in the lead unit. They cannot be activated in the rear unit without an extra driver. Although engine braking is very effective in general operation, low speed braking has been a problem under certain conditions.

6. Maneuverability on Land and in Water:

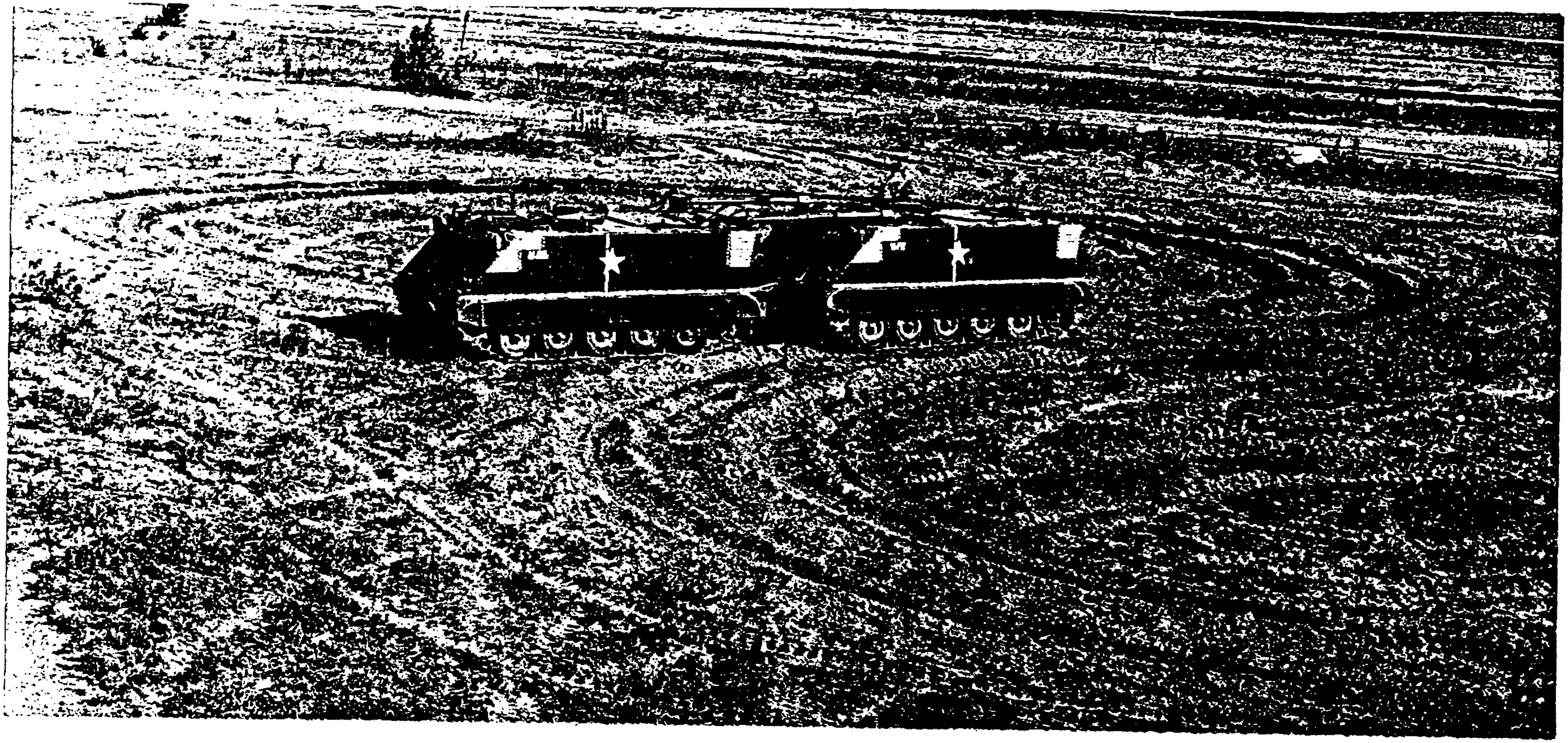
Since the coupled vehicles are force articulated in yaw, all four tracks are fully powered during a steering maneuver. This ability to provide full traction during a turn provides superior maneuverability in difficult terrain. This is particularly noticeable in snow. Tests performed in approximately 3 feet of snow showed repeatedly that the coupled vehicle could be maneuvered precisely under conditions where the single vehicle could barely extricate itself.

The steerability of the coupled vehicles is limited in the present configuration to a turning radius of 41 feet (Figure 15) but is relatively unaffected by conditions which will immobilize the single unit. The single vehicle, with track steer, can turn approximately in its own length, but only when soil conditions are permitting. The tradeoffs between steerability and maneuverability in difficult soil will have to be investigated in the future.

In the water maneuverability tests conducted in Quincy Pond, Mason, Michigan, both the coupled and single units were subjected to the modified Kempf "Z" maneuver previously developed by Stevens Institute (6). The results (7) shows that both the coupled and single unit have approximately equivalent overall control in water. A representative result shown in Figure 16 shows that the coupled units need almost negligibly more space than the single vehicle to complete a maneuver, but, although their straight ahead speed from t_o to t = 10 sec. is higher, the speed in a turn t_{10} to t = 45 sec. is slower. A complete full circle turning test shows that both the single and coupled units have about the same turning radius of approximately 40 ft. The coupled units are steered by yaw articulation and the single unit by track steer.

7. Water Speed Evaluation:

The coupled vehicle achieved a maximum water speed of 4 mph (7). This represents an increase of approximately 10 percent when compared with a single M-113 tested under the same conditions. The increase in speed was achieved by pitching the front unit up 8 degrees. In this condition, the rear unit is pitched down 13 degrees. Previously conducted and related model tests (8, 9) indicated that an increase in the order of 14 - 16% could be expected due to the increased length of the combination.



TURNING TESTS ON LAND

FIGURE 15

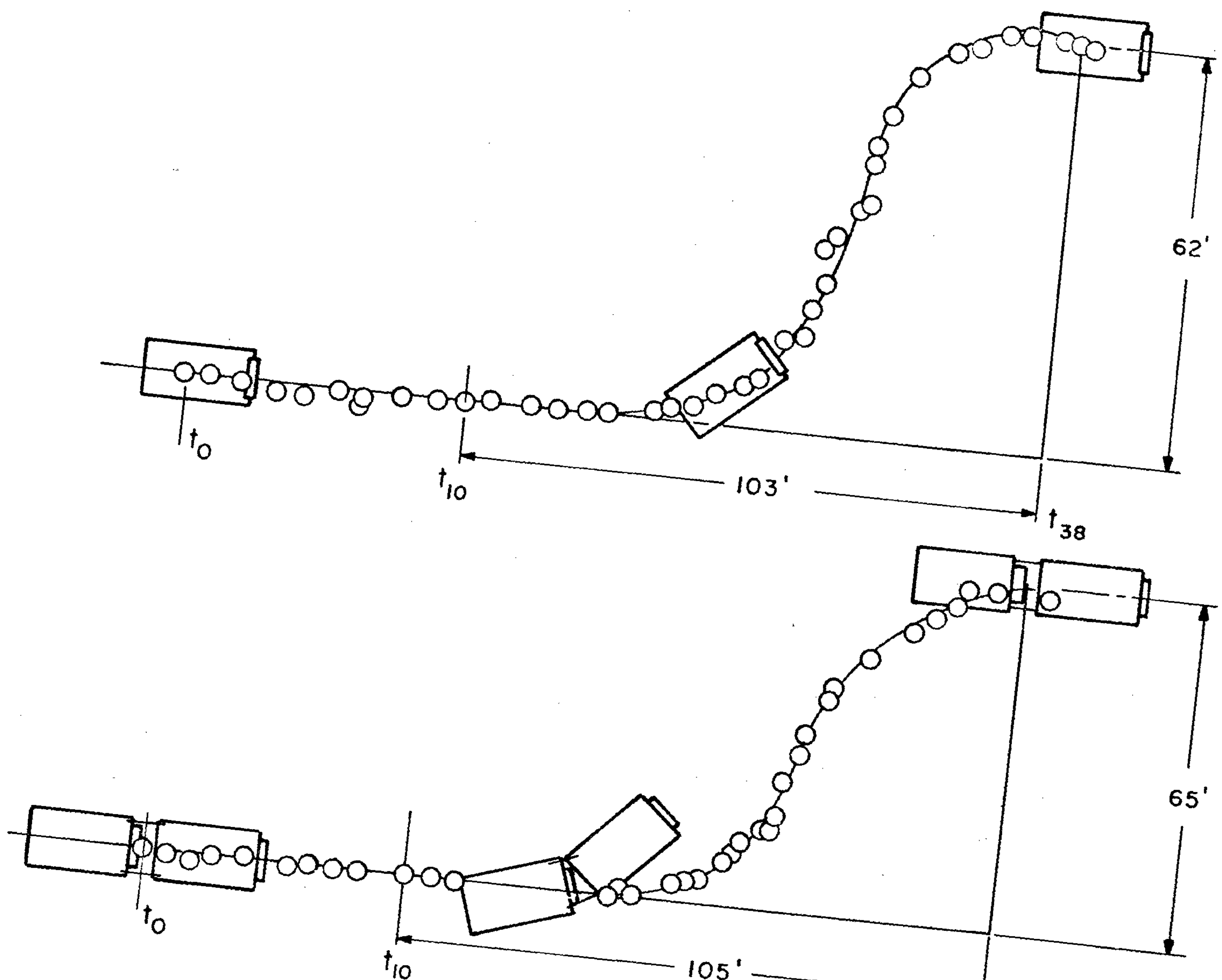


FIG. 16. PLOT OF KEMPF "Z" MANEUVER FOR SINGLE AND CYBERNETICALLY COUPLED M-113.

Observation of the water flow (Figure 17)* shows considerable turbulence between the vehicles. This turbulence is due to the separation distance between the vehicles necessary for adequate articulation, and because some of the water flow resulting from the front unit's track propulsion impinges on the front slope of the rear unit. The wave board of the rear unit was deployed but was too narrow to suppress this flow, but mudflaps attached to the fender extensions of the forward unit proved to be effective. It is also very likely that the wake fraction of the front track propulsion degrades the propulsive efficiency of the rear track. Whether or not independent control of power of the two vehicles could improve water speed need to be explored.

8. Exiting From Deep Water:

The exiting tests were also performed at the Quincy Pond test site. The approximate water depth in the test area is 25 feet. The banks are composed of fine sand and silt and are highly unstable. At three out of the four exit sites, the bank slopes below the water were equal to or slightly greater than the angle of repose of the soil. The soil internal angle of friction, was 32-36 degrees and the cohesion was 0-0.1 psi in the wet condition. The bank slope is maintained at these high angles by the apparent cohesion caused by the water. There are no vegetation or other soil stabilizing influences. Therefore, the soil conditions at the test site are considered more severe that those anticipated on an "average" river bank.

The coupled vehicle was able to exit over all of the banks whose profiles are shown in Figure 18. The single vehicle was able to exit at Site 1 but was immobilized at Site 2 and consequently Sites 3 and 4 were not attempted. Figure 19 shows the coupled units exiting at Site 4.

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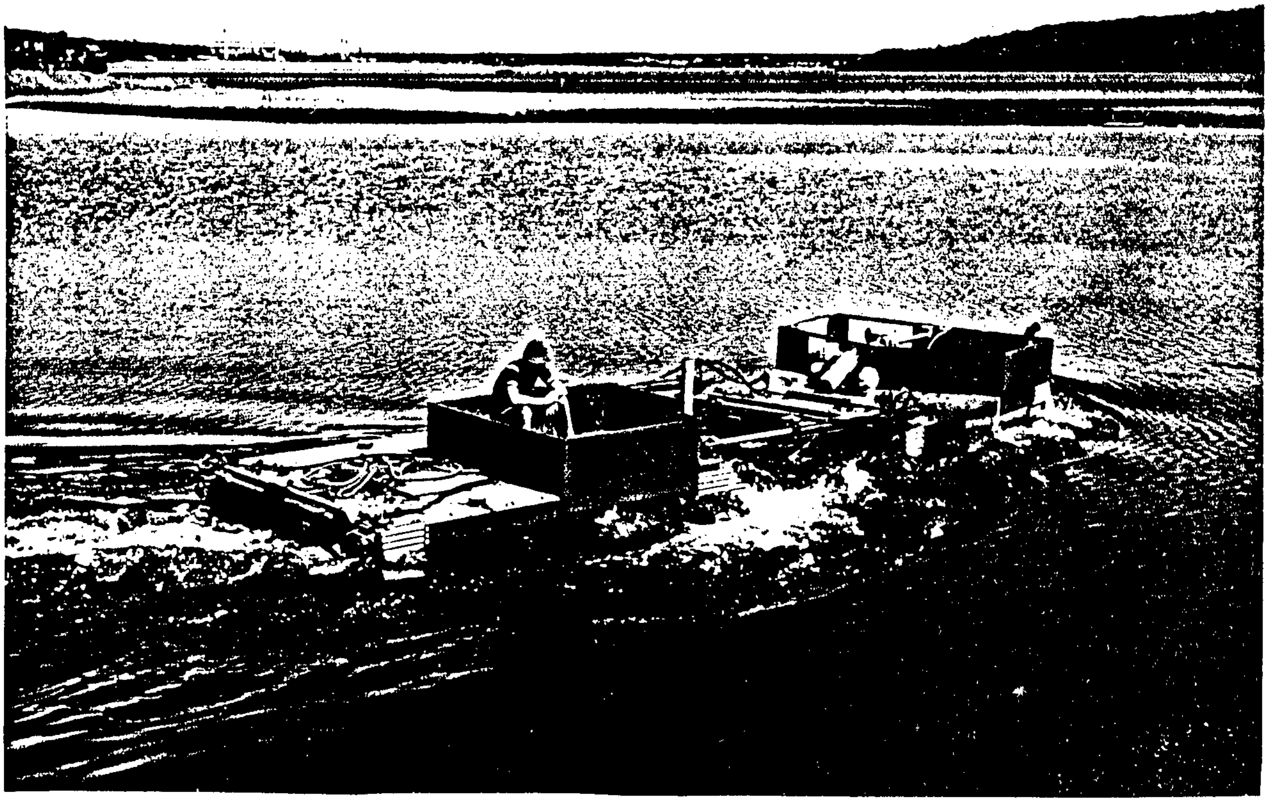
*The superstructures were added in anticipation of extreme attitudes during the pitching and the exiting tests.



COUPLED VEHICLES AFLOAT

FIGURE 17

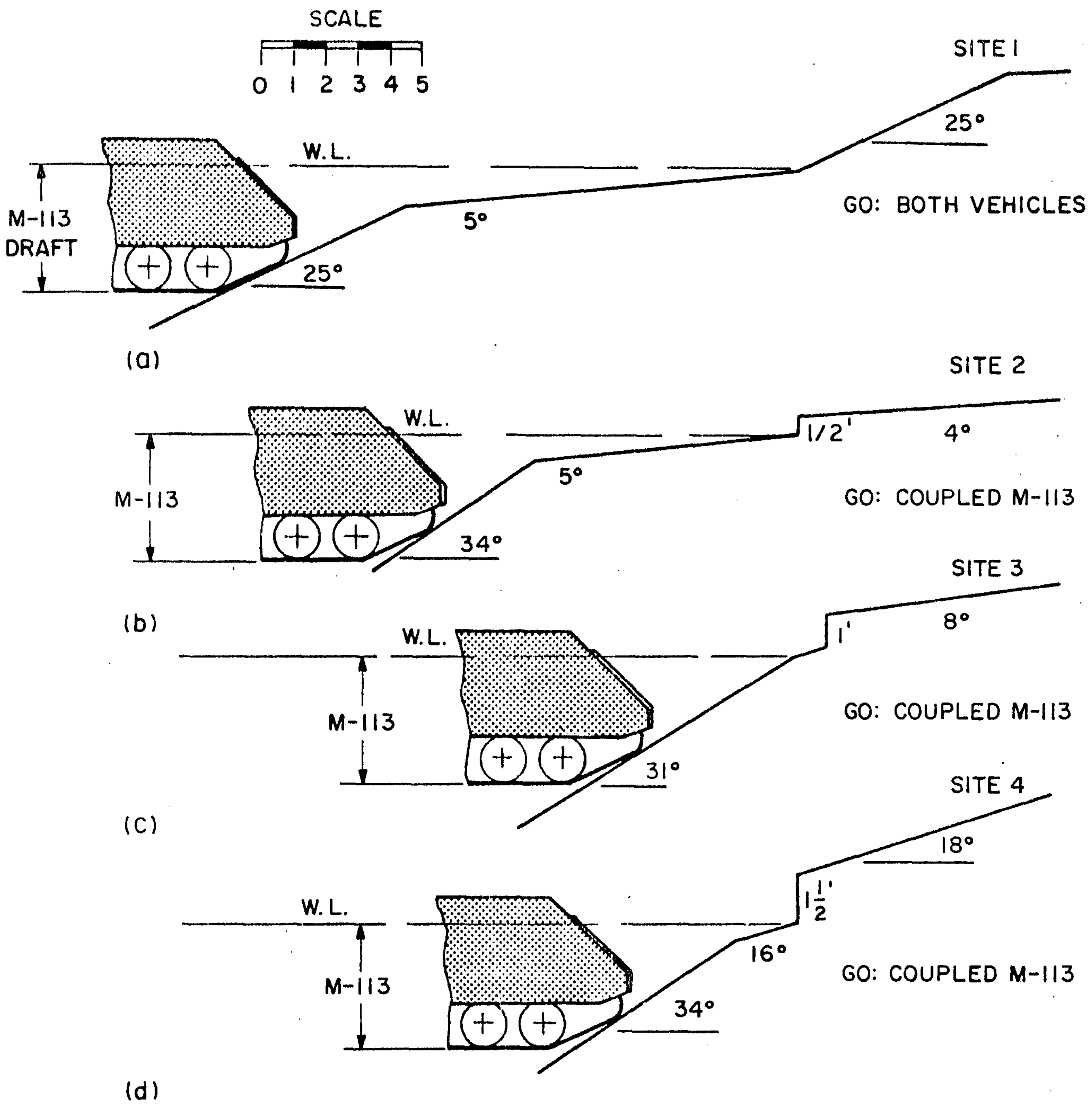
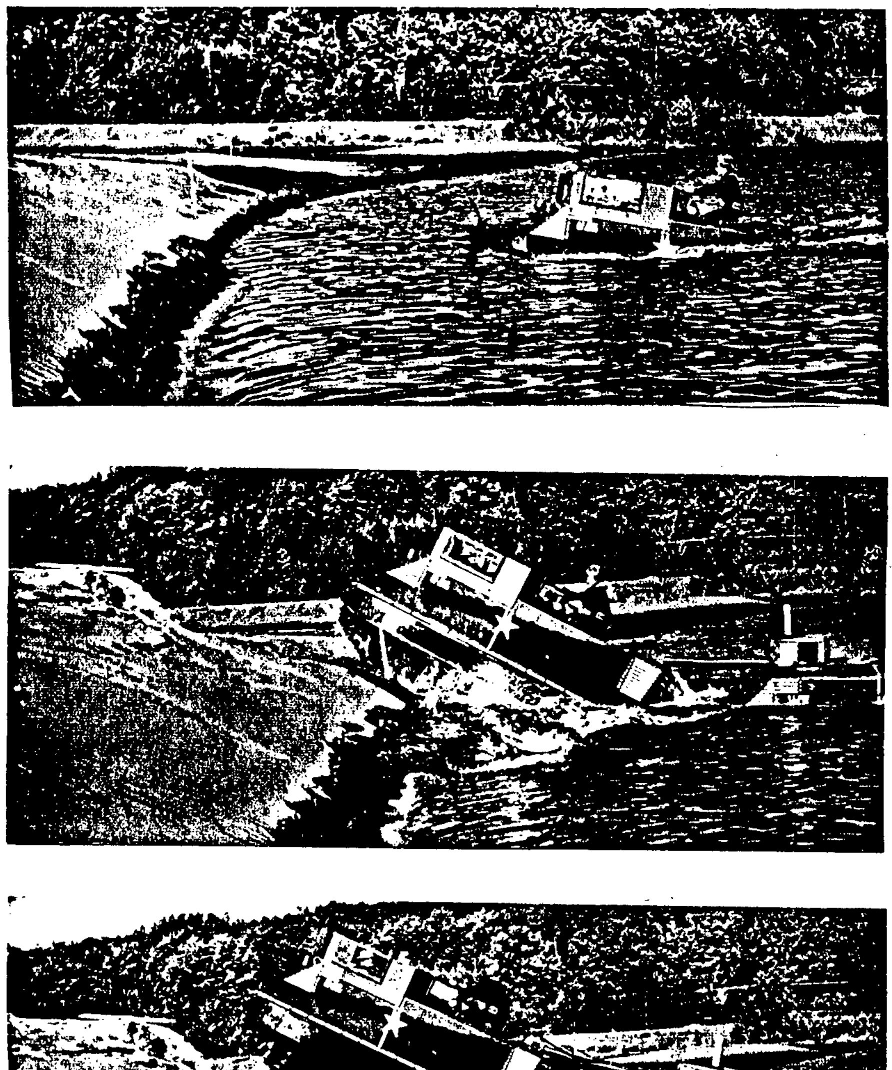


FIG. 18. WATER EXIT, BANK PROFILES.



COUPLED VEHICLES EXITING AT SITE 4

FIGURE 19

The increased capability of the coupled vehicle can be attributed to the ability of keeping its track in compliance with the bank throughout the exiting process, to its added momentum and to the absence of a reflected wave.

When the single vehicle engages the bank, the soil under the front portion of the track must support a higher than normal loading. This fails the soil and makes an exit difficult. The coupled vehicle is able to pitch up and engage the bank over almost the entire track length. The track loading is no more than normal and the tendency for soil failure is reduced.

A single M-113 pushes a large bow wave ahead of the vehicle. The reflection of this wave from the bank acts on the vehicle before the vehicle can exit. The reflected wave is often strong enough to push the vehicle several feet back from the bank. The pitched up configuration of the coupled vehicle considerably reduces the reflected wave and is therefore able to use its relatively unreduced momentum to assist the vehicle over the first, critical stages of exiting. Again, it is obvious that the ease and speed with which pitch control can be accomplished is a critical factor in exiting so that the full speed and momentum capability can be utilized to maintain smooth and continuous optimum compliance of all tracks with the bank profile during the complete exiting process. These tests were performed in still water; it remains the objective of future tests to determine the extent to which the combinations of increased speed, power yaw articulation and positive pitch control will aid in the exiting process from a swiftly flowing river under a variety of "real" environmental conditions.

9. Negotiation of Rigid Step Obstacles:

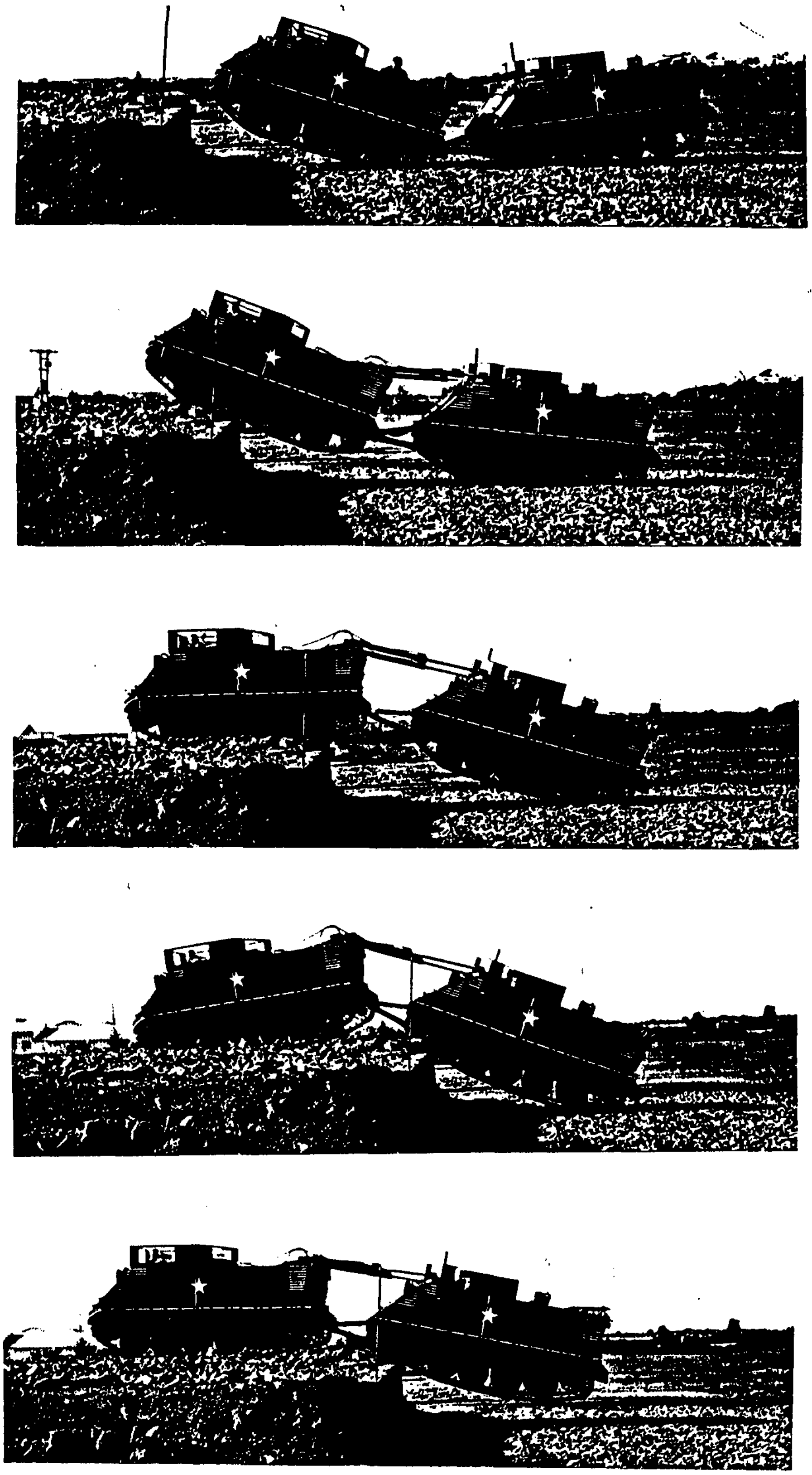
Obstacle negotiation tests were performed on obstacles constructed at the Keweenaw Field Station. The obstacles have a front face constructed of hardwood timbers, backfilled with mine tailings and soil. Although, their sharp leading edge presents a more severe demand on the track and suspension than a natural obstacle would, their construction ensured that their shape would remain constant under prolonged comparative testing.

To date, the coupled vehicle has successfully negotiated vertical rigid step obstacles up to five feet in height (Figure 20). By comparison the single unit is limited to mounting a two-foot high step.

Most of the obstacles have been negotiated both with and without the force-feedback system. So far the drivers reactions are mixed as to which mode is better. It was determined, however, that two and three foot obstacles can be driven over with the force-feedback system acting as an autopilot. It is only necessary for the driver to lift the front end of the lead unit onto the step, from that point on the "zero-force-seeking" characteristics of the force-feedback system are sufficient to provide the proper command signals for the vehicle to drive over the step "hands-off".

But, when climbing the higher obstacles, the force-feedback system can no longer be used as an autopilot. In fact, the "drive over" technique is no longer possible, but a "lift over" technique has to be used. (Note the last sequence of Figure 20). After the front unit is lifted onto the obstacle, the edge will support the front unit as it is driven over, and the forward portion of the rear unit's track must be lifted onto the edge by a maximum pitch-down articulation. Once the first roadwheels, or at least the drive sprockets of the rear unit, are engaged with the edge of the obstacle, "pitch-up" actuation is necessary, thereby pivoting the rear unit about the edge of the step until the angle of approach with respect to the top surface of the obstacle is such that the traction is sufficient to drive the units forward. At this point, the vehicles are about level but with a slight inter-vehicle pitch-up attitude.

At times, when the surface of the obstacles was slippery, the drive over technique was not possible, especially on the four foot step, because the forward unit could not generate enough traction to help pull the rear unit onto the step. With the lift over technique this problem does not exist because all of the lifting is done by the hydraulic inter-vehicle assist. This clearly demonstrates the value of supplementing the traction



5' HIGH STEP OBSTACLE

FIGURE 20

forces with inter-vehicle assist forces. The response of the hydraulic system is sufficient that the 2 and 3 foot obstacles can be driven over at between 2 and 5 mph without hesitation. It is also clear that natural soil obstacles of greater height could be negotiated without loss of momentum, because they would not create the high concentrated impact loads which the hard step obstacles impose on tracks and suspension.

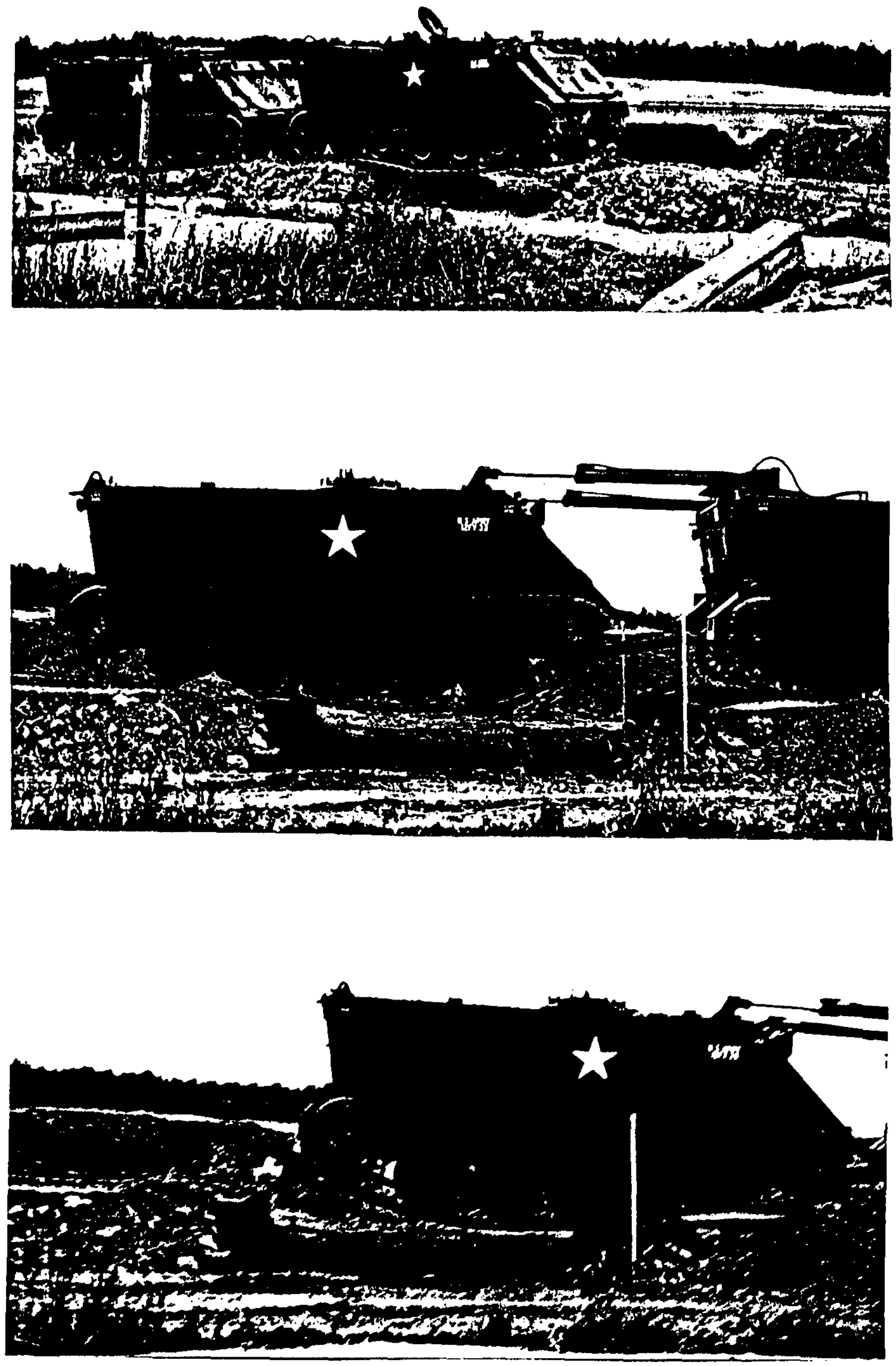
10. Trench Crossing:

The performance was compared over a series of V-ditches and two foot deep vertical wall trenches. The single vehicle easily traversed the five foot ditch, but barely traversed the eight foot V-ditch; it was immobilized by the seven foot wide trench, from which it had great difficulty in extricating itself. By comparison, the coupled units traversed all ditches including the eleven foot wide trench without difficulty, (see Figure 21). Observations made while descending the two and three foot high step obstacles indicate that it should be possible to cross ditches up to 14 feet wide. It is very difficult to assess whether the force-feedback system is a help or hinderance in the trench crossing operation. Obviously, the wider trenches cannot be crossed hands-off because the zero-force seeking characteristic will cause the front end to drop into the trench. When operating without force-feedback, it is easier to cross hands-off since the units will stay straight and level, the extra rigid length being all that is necessary to span the trench. Whether this holds true for trenches wider than eleven feet will have to be determined in future tests.

CONCLUSION

In the authors opinion, the system presented here produced a quantum jump in performance when exiting from a body of water, in the crossing of step obstacles and trenches and in cross-country ride quality. The price to be paid are the increased complexity, and vulnerability therefore less reliability, and in reduced steerability (but not necessarily maneuverability).

In most other areas of cross-country operation the true net gain depends largely on the particulars of terrain and environment.



11 FOOT WIDE TRENCH

FIGURE 21

IN-ARM SUSPENSION

Presented By

M. KAIFESH
U.S.

IN-ARM SUSPENSION

INTRODUCTION

THE IN-ARM SUSPENSION PROGRAM IS A DIRECTED EFFORT TO DEVELOP IMPROVED SUSPENSION SYSTEMS FOR TRACKED VEHICLES, FOR THE POST 1980 ARMY.

SLIDE I:

RECENT VEHICLE CONCEPT DEFINITION STUDIES HAVE SHOWN THAT HIGH CROSS-COUNTRY SPEED IS ESSENTIAL FOR HIGH COMBAT CAPABILITY. CONCURRENT STUDIES IN THE SUSPENSION AREA HAVE SHOWN THE PRINCIPLE CHARACTERISTICS NEEDED FOR HIGH MOBILITY ARE HIGH WHEEL TRAVEL, LOW SPRING RATE, AND OPTIMUM DAMPING.

OBJECTIVE

THE GOAL OF THE IN-ARM SUSPENSION PROGRAM IS TO DEVELOP A SUSPENSION SYSTEM LOCATED OUTSIDE THE VEHICLE HULL THAT HAS THESE CHARACTERISTICS. THE COMPLETE SUSPENSION SYSTEM IS TO BE CONTAINED WITHIN THE LIMITED SPACE OF THE ROADARM ENVELOPE.

THE IN-ARM SUSPENSION WILL PROVIDE SEVERAL ADVANTAGES, IN ADDITION TO HIGH MOBILITY. MAINTENANCE WILL BE FACILITATED DUE TO THE EASE OF INSTALLATION AND REMOVAL. THE UNITS WILL BE ATTACHED DIRECTLY TO THE OUTSIDE OF THE HULL WITH FOUR TO SIX FASTENERS. THESE MODULAR TYPE SUSPENSION UNITS COULD BE APPLIED TO VARIOUS STATIONS ON AN INDIVIDUAL VEHICLE AND POSSIBLY TO DIFFERENT VEHICLES.

1 OFF/2 ON:

HIGH WHEEL TRAVEL, A MINIMUM OF 15 INCHES STATIC TO JOUNCE, IS REQUIRED IN THE TECHNICAL SPECIFICATION. OTHER LIMITING DIMENSIONS ARE SHOWN IN THIS CHART. THEY INCLUDE:

- A ROADARM LENGTH OF BETWEEN 15.5 and 18 INCHES.
- A MAXIMUM OF 5 INCHES OF TRAVEL FROM HORIZONTAL TO FULL JOUNCE
- A MAXIMUM STATIC ANGLE OF 40°

A RELATIVELY SOFT SPRING RATE IS BEING SOUGHT BY SPECIFYING A NATURAL FREQUENCY IN

BOUNCE, OF 40 TO 50 CPM. THIS PARAMETER, ALONG WITH A STATIC WHEEL LOAD OF 4500 POUNDS VERTICLE PER WHEEL STATION DEFINES THE SPRING RATE AT STATIC.

A DAMPING RATIO OF .5 TO 16 CRITICAL AT STATIC IS A REQUIREMENT IN THE TECHNICAL SPECIFICATION. THIS DAMPING RATE MUST BE VARIABLE, AND CONTROLLED BY THE MOTION OF THE ROADARM. DAMPING IS TO BE A MAXIMUM AT VEHICLE CRAWL SPEEDS UP TO 5 MPH, AND IS TO DROP OFF TO 0 AT VEHICLE VELOCITIES OF 20 MPH.

2 OFF:

APPROACH

TWO DIFFERENT TYPES OF SYSTEMS ARE BEING INVESTIGATED. ONE, THE HYDROPNEUMATIC, IS AN OUTGROWTH OF PREVIOUS DEVELOPMENT WORK. THE OTHER APPROACH, WHICH UTILIZES A COMPRESSIBLE FLUID AS THE SPRINGING MEDIA, IS DESIGNATED THE HYDROMECHANICAL IN-ARM SUSPENSION SYSTEM. BOTH SYSTEMS MUST MEET THE SAME PERFORMANCE REQUIREMENTS.

------------------------------------ P A U S E ------------------------------------

CONCEPT DESCRIPTION

3 ON:

THE HYDROPNEUMATIC IN-ARM CONCEPT FUNCTIONS VERY SIMILARLY TO THE MBT AND XM1 PROTOTYPES. SPRINGING IS PROVIDED BY A HYDRAULICALLY ACTIVATED AIR SPRING.

WHEN A BUMP IS ENCOUNTERED THE WHEEL AND ARM ASSEMBLY MOVES UPWARD. THE ACTUATOR PISTON IS HELD IN PLACE BY THE STATIONARY CRANK AND AS THE ARM MOVES UP THE CYLINDER MOVES INWARD OVER THE ACTUATOR PISTON.

THIS ACTION FORCES HYDRAULIC FLUID, FROM THE ACTUATOR CYLINDER THROUGH THE DAMPING MANIFOLD, INTO THE ACCUMULATOR CYLINDER. THE INCREASE IN PRESSURE ON THE ACCUMULATOR PISTON COMPRESSES THE NITROGEN AND PROVIDES THE RESISTING FORCE.

DAMPING IS OBTAINED, BY REGULATING THE FLOW OF HYDRAULIC FLUID BETWEEN THE ACTUATOR AND ACCUMULATOR CYLINDER THROUGH THE DAMPING MAINFOLD. JOUNCE DAMPING IS PROVIDED BY A DAMPING ORIFICE IN PARALLEL WITH A RELIEF VALVE. A REBOUND CHECK VALVE, ASSURES NO REBOUND DAMPING. THIS DEVICE DIFFERES FROM PREVIOUS HYDROPNEUMATIC

SUSPENSION SYSTEMS FUNCTIONALLY IN THAT EACH UNIT CONTAINS A FIXED VOLUME OF FLUID. THE UNITS ARE NOT INTERCONNECTED HYDRAULICALLY AND ARE BEING DESIGNED TO OPERATE INDEPENDENTLY AS ISOLATED UNITS. NO HYDRAULIC CONNECTIONS ARE REQUIRED THROUGH THE VEHICLE HULL. ANOTHER MAJOR DIFFERENCE IS THAT THE ROADARM OF THE IN-ARM DESIGN IS USED AS BOTH A STRUCTURAL MEMBER AND A HOUSING.

3 OFF/4ON:

P A U S E

THIS SYSTEM PROVIDES 15-1/4 INCHES OF JOUNCE TRAVEL AND 3-1/4 INCHES OF REBOUND. THE UNDAMPED NATURAL FREQUENCY, OF THE SYSTEM IS 38 CYCLES PER MINUTE. THE SPRING RATE IS VARIABLE WITH A RATE OF AT STATIC RATE OF 180 POUNDS PER INCH.

THE DAMPING ORIFICE WILL PROVIDE A DAMPING LEVEL OF .5 OF CRITICAL DURING THE INITIAL PORTION OF THE DAMPING LEVEL, VERSUS FLOW RATE CURVE. THE DAMPING VALVE POPPET AND FLOW PASSAGES HAVE BEEN DESIGNED TO PROVIDE MAXIMUM FLOW INDUCED LIFT FORCES NEAR THE HIGH END OF THE FLOW CAPABILITY OF THE VALVE. THIS LOWERS THE DAMPING LEVEL AT HIGH FLOW WHICH CORRESPONDS TO THE HIGHER VEHICHE VELOCITIES.

THE HYDROPNEUMATIC IN-ARM SUSPENSION HAS GONE THROUGH A CONCEPT DESIGN PHASE AND A DETAILED DESIGN PHASE. ONE PROTOTYPE UNIT IS CURRENTLY BEING FABRICATED.

4 OFF/5 ON:

P A U S E

THE HYDROMECHANICAL IN-ARM SUSPENSION CONSISTS OF A TRUNNION MOUNTED SPRING/DAMPER STRUT AND A TRUNNION MOUNTED ROADARM. THE LOCATION OF THE STRUT RELATIVE TO THE ROADARM FORMS A CRANK AND SLIDER TYPE LINKAGE. SPRINGING IS PROVIDED BY A FIXED VOLUME OF COMPRESSIBLE FLUID.

WHEN THE WHEEL AND ARM ASSEMBLY IS DEFLECTED UPWARD THE SPRING PISTON IS FORCED INTO THE LIQUID SPRING CYLINDER CAUSING THE SILICON FLUID TO COMPRESS. THE COMPRESSIBILITY OF THE FLUID PROVIDES THE RESISTING FORCE.

A RESERVIOUR TO PROVIDE THE NECESSARY VOLUME OF FLUID IS ATTACHED TO THE SPRING STRUT AND PROJECTS INTO THE HULL.

THE DAMPING PISTON IS CONNECTED AXIALLY TO THE SPRING ROD AND DAMPING IS CAUSED BY THE PISTON PUSHING OIL THROUGH THE VALVING IN THE DAMPING MANIFOLD TO THE OTHER SIDE OF THE CYLINDER. JOUNCE DAMPING IS PRODUCED BY A COMBINATION OF A SMALL ORIFICE AND TWO RELIEF VALVES IN PARALLEL.

----------------------------------P A U S E ----------------------------------

5 OFF/6 ON:

THIS SYSTEM ALSO PROVIDES THE REQUIRED 15 InCHES OF JOUNCE TRAVEL AND HAS THREE INCHES OF REBOUND TRAVEL. ITS UNDAMPED NATURAL FREQUENCY IS 57 CPM. ITS STATIC SPRING RATE IS 554 LB/IN AND IT PROVIDES A VARIABLE SPRING RATE WHICH IS SLIGHTLY STIFFER THAN THE HYDROPNEUMATIC SYSTEM.

THE DAMPING VALVING PROVIDES AN APPROXIMATE VISCOUS DAMPING LEVEL OF .5 OF CRITICAL AT LOW VEHICLE SPEEDS. AN INERTIA BYPASS VALVE IS USED TO LIMIT DAMPING, AT HIGH VEHICLE VELOCITIES. VALVING TO INCORPORATE LOCKOUT AND VARIABLE HEIGHT ARE INCLUDED IN THE DAMPING MANIFOLD.

THE HYDROPNEUMATIC IN-ARM SUSPENSION DESIGN HAS GONE THROUGH A CONCEPT DESIGN PHASE AND THE DETAILED DESIGN PHASE IS CURRENTLY BEING COMPLETED. A DECISION TO PROCEED WITH FABRICATION OF A PROTOTYPE UNIT WILL BE MADE UPON COMPLETE ANALYSIS OF THE DETAILED DESIGN PHASE.

6 OFF

IN-ARM SUSPENSION

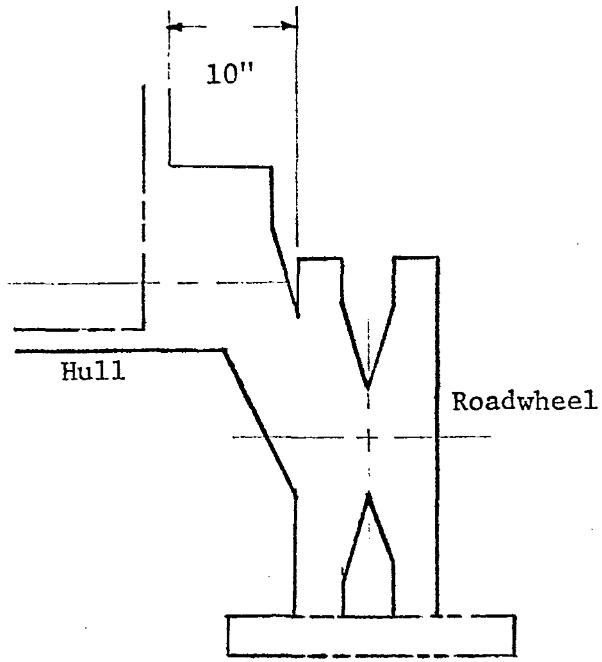
OBJECTIVE: TO DEVELOP AN IMPROVED SUSPENSION SYSTEM FOR TRACKED VEHICLES.

CHARACTERISTICS: HIGH WHEEL TRAVEL.
LOW SPRING RATE.
OPTIMIZED DAMPING.

ADVANTAGES: PROVIDES FOR HIGH MOBILITY.
REQUIRES LITTLE INTERNAL HULL SPACE.
LENDS ITSELF TO MODULAR APPLICATION.

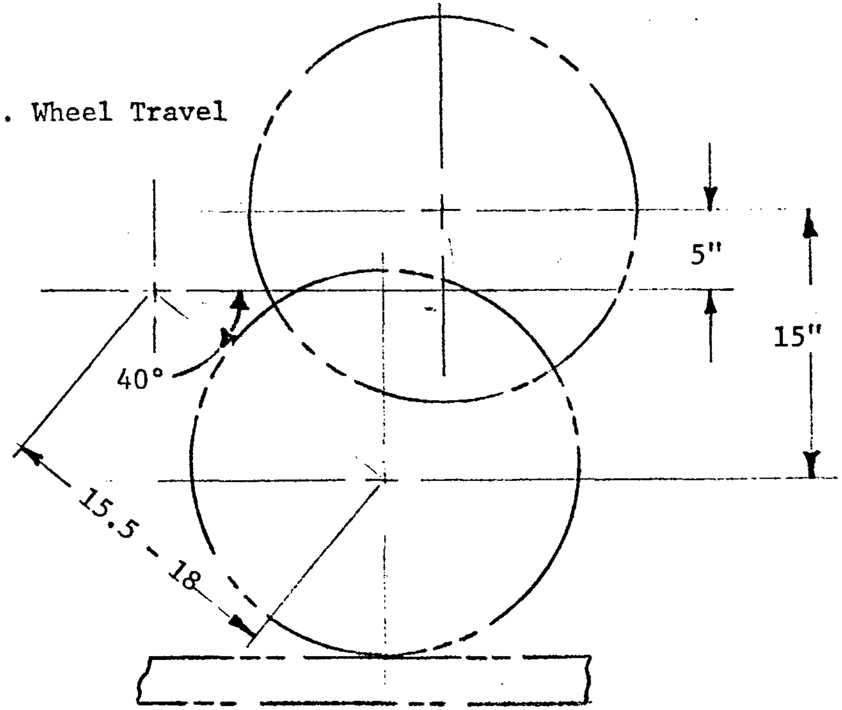
SLIDE 1

IN-ARM SUSPENSION SPECIFICATIONS



Maximum Space Allocation

A. Wheel Travel



B. Spring Rate

Static Wheel Load - 4500 pounds
Natural Frequency - 40-50 CPM

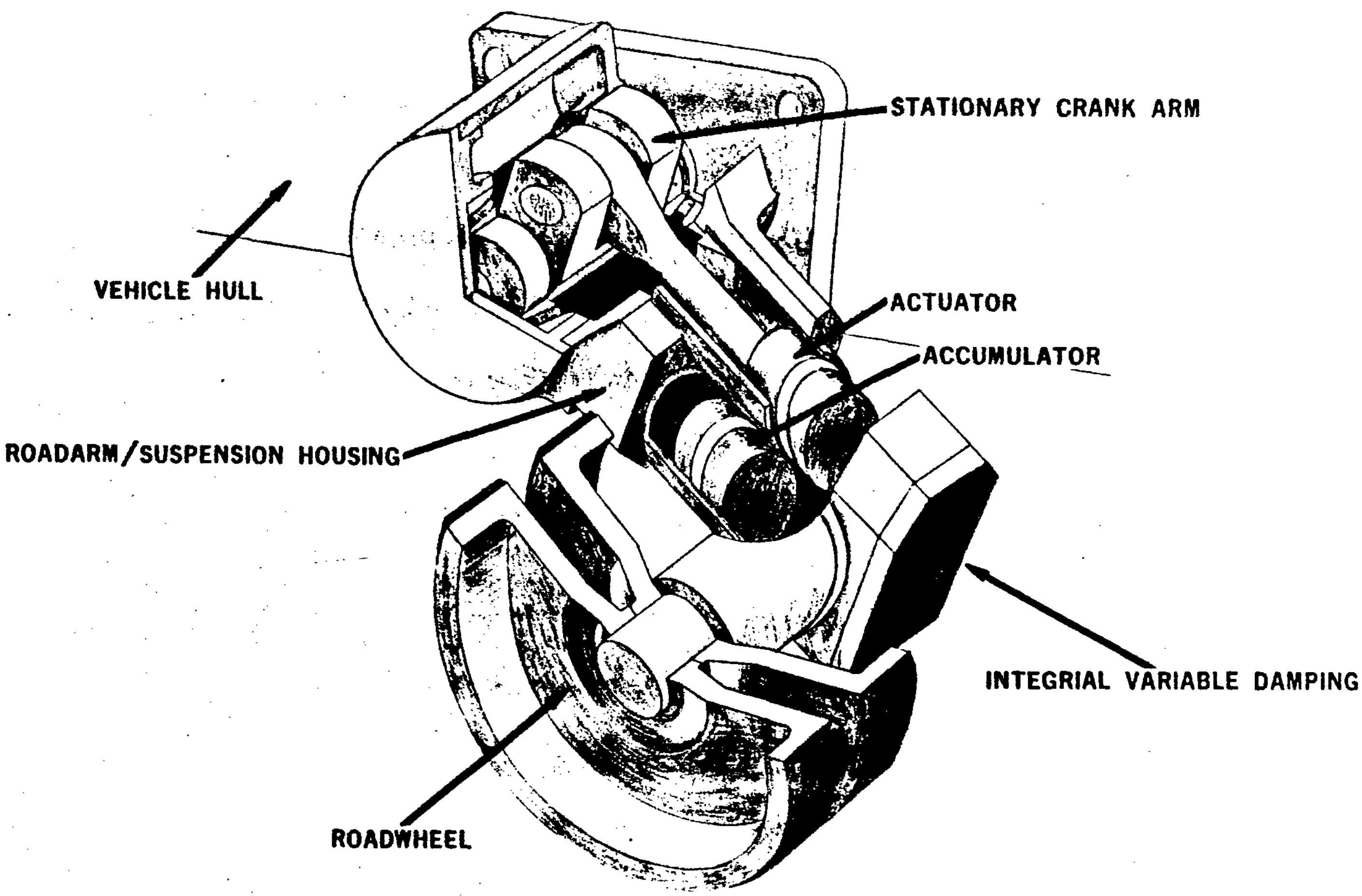
C. Damping

Jounce Damping Only

Damping Ratio - .5 to .6 at Static
Variable - Maximum at 3 MPH
Minimum at 20 MPH

SLIDE 2

IN-ARM SUSPENSION•HYDROPNEUMATIC



SLIDE 3

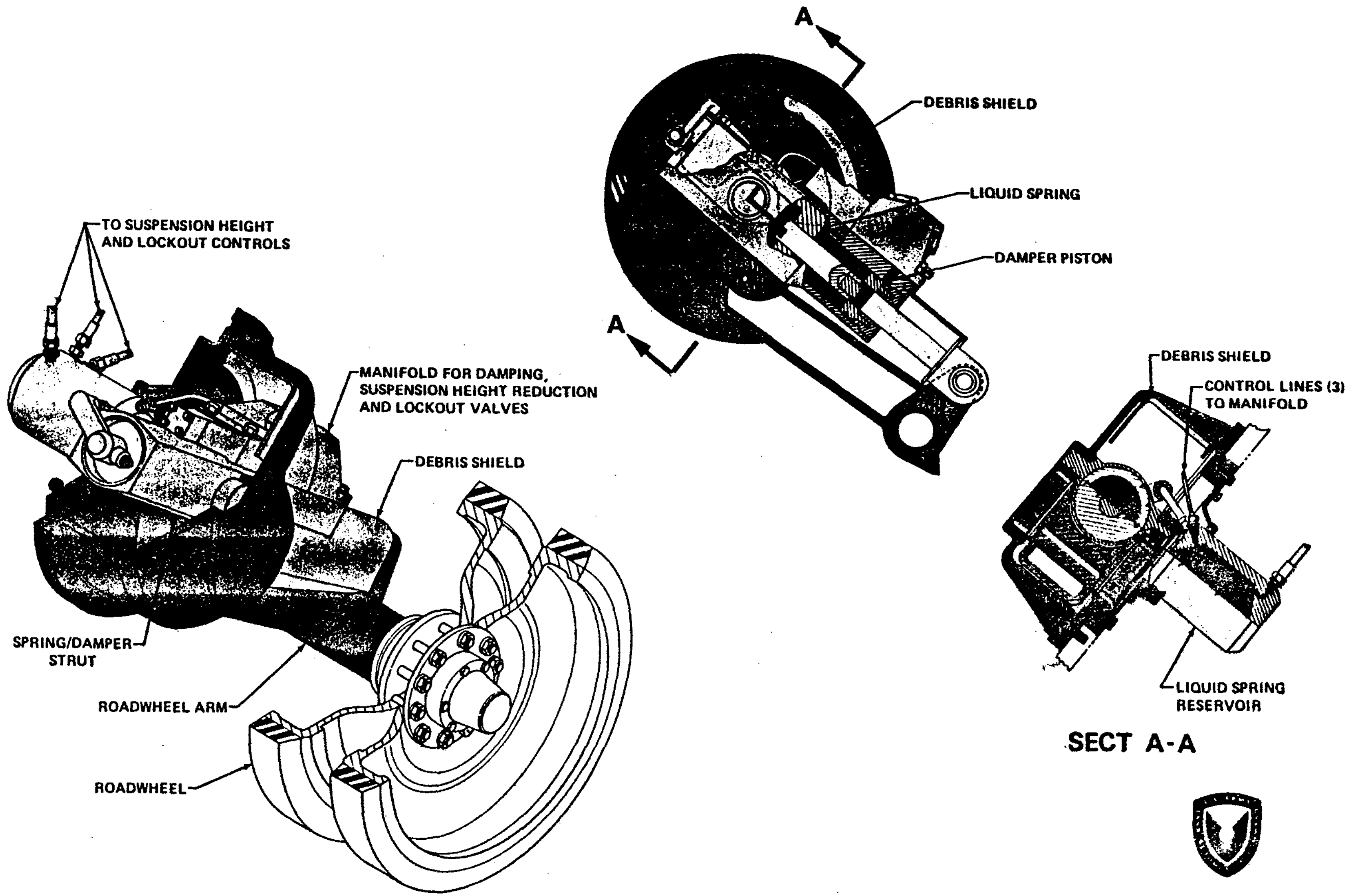
HYDROPNEUMATIC IN-ARM SUSPENSION

COMPLIANCE WITH PRINCIPAL SPECIFICATIONS

	CRITERIAL	SPECIFICATION	PROVIDED BY DESIGN
1.	Hull to inner edge of roadwheel dimension	10 inch maximum	8 inches
2.	Roadarm length between wheel spindle and mounting trunnion center lines	14 to 18 inches	16.0 inches
3.	Vertical jounce motion of wheel	15 inches minimum	15.24 inches
4.	Jounce motion above the horizontal	5 inches maximum	4.95 inches
5.	Static position of roadarm below horizontal	40° maximum	40°
6.	Vertical rebound motion of wheel below static position	Not specified	3.25 inches
7.	Load capacity at static position	4500 pounds	4500 pounds
8.	Instataneous Natural Frequency at Static Position in Bounce	40 to 50 CPM	38 CPM
9.	Spring Rate (STATIC)	Not specified	180 lbs/in
10.	Unidirectional Damping	Jounce Damping Only	Yes
11.	Damping Ratio (STATIC)	.5 to .6 at 3 MPH	0.5
12.	Variable Damping	Maximum at 3 MPH Zero at 20 MPH	Yes Drops to minimum

SLIDE 4

IN-ARM SUSPENSION • HYDROMECHANICAL



SECT A-A

SLIDE 5

HYDROMECHANICAL IN-ARM SUSPENSION

COMPLIANCE WITH PRINCIPAL SPECIFICATIONS

	CRITERIA	SPECIFICATION	PROVIDED BY DESIGN
1.	Hull to inner edge of roadwheel dimension	10 inch maximum	10 inches
2.	Roadarm length between wheel spindle and mounting trunnion center	14 to 18 inches	18 inches
3.	Vertical jounce motion of wheel	15 inches minimum	15 inches
4.	Jounce motion above the horizontal	5 inches maximum	5 inches
5.	Static position of roadarm below horizontal	40° maximum	34°
6.	Vertical rebound motion of wheel below static position	Not specified	3 inches
7.	Load capacity at static position	4500 pounds	4500 pounds
8.	Instataneous Natural Frequency at Static Position in Bounce	40 to 50 CPM	57 CPM
9.	Spring Rate (STATIC)	Not specified	554 lb/in
10.	Unidirectional damping	Jounce damping only	Yes
11.	Damping ratio (STATIC)	.5 to .6 at 3 MPH	0.5
12.	Variable Damping	Maximum at 3 MPH Zero at 20 MPH	Yes Cutoff above 20 MPH

SLIDE 6

POWERED WALKING BEAM SUSPENSION

Presented By

W. BRANDT
U.S.

POWERED WALKING BEAM SUSPENSION SYSTEMS

The overall objective in development of future tactical vehicles is to achieve a mission effectiveness which will be compatible with that of future combat vehicles. Mission effectiveness includes reliability, availability, maintainability, durability (RAM-D), mobility in a wide variety of terrain conditions, and capacity to carry and deliver cargo at some required rate. The specific objective in a current TACOM research and development task is to field test and evaluate the powered walking beam drive and suspension systems designed, fabricated, and installed on two standard U.S. military trucks by ENGESA of Sao Paulo, Brazil.

Figure I shows a general over-view of the drive and suspension configuration from ENGESA. Field operation observation of similar systems on Brazilian Army trucks indicated improvements in off-road mobility. The utilization of basically commercial differential axles and gearing has good potential for low cost. The system is quite readily adapted to existing standard military trucks as well as future development trucks which may be commercial design. The darker shaded portions of Figure I indicate the component areas in both the 2-1/2 and 5 ton U.S. military test trucks which were modified by installing the ENGESA hardware. The major change was made in the rear suspension and drive area. The original M35A2 and M813 trucks had two axles arranged in tandem, or what is commonly referred to as a bogie type suspension. The modified vehicles have a single, larger axle with powered walking beams at each end. Also, where dual tires were previously installed, single tires of larger size are now used.

Figure II illustrates the mechanism of an ENGESA transfer case. Power input from the transmission flows through the high (H) or low (L) range selector gear and proceeds through a constantly engaged path to the output for the rear axle. The other path transmits power to the engage and disengage shift gear for the front axle. A neutral (N) position between the high and low range selector gear positions is utilized when tractive effort is not needed and some accessory power, such as for a winch, is required. The power-take-off is an available option which is not present in the two modified U.S. trucks currently being tested. The power-take-off drive shaft and external flange connection are shown by the shaded portion and phantom lines respectively.

Figure III shows a top view of the rear drive and suspension area. Some features to note are as follows:

1. Emergency Brake.
2. Differential Lock to prevent wheels from spinning on one side only.

3. Two spring mounting pads (surfaces with four hole pattern located between differential lock and walking beams) for conventional multi-leaf spring assemblies; also, see Figure I.
4. Four Brake Actuators.
5. Powered Walking Beams.

Figure IV shows a full section view of a representative portion of the Walking Beam with its internal power train. A solid differential axle shaft projects outward through the walking beam at the center of the beam span and connects through splines to a coupling at the outboard end of the shaft. The coupling in turn engages two torque tubes. The tube nearest the axle shaft connects to the differential lock mechanism and the other tube drives the spur gear trains which engage the individual wheel axle shafts at the ends of the walking beam.

Figure V displays the approximate ramp height capability differential between standard and modified 5 ton cargo trucks (M813). The front suspensions and drive axles of both are basically the same. Therefore, the difference in obstacle negotiation capability is primarily due to the relative magnitudes of vertical wheel travel limits in the rear suspensions. Full jounce into the bump stop on the standard vehicle with cross-country payload will provide approximately 6-1/4 inches wheel travel (static to bump) as compared to approximately 22 inches on the modified vehicle. A 45° ramp was somewhat arbitrarily assumed because this was approximately the angle of repose for a gravel pile used as a trial obstacle at TACOM. It was determined by scaled layout that approximately a 35 inch elevation change could just be cleared by the standard vehicle while the modified vehicle could clear approximately a maximum of 61 inches (see upper left and right views in Figure V). Referring to the lower left hand view on Figure V, note that as the front wheels of the standard vehicle rear suspension proceed up the ramp a point is reached where the bump stops are engaged. From this point on the rear wheels of the rear suspension are progressively raised from the terrain surface and therefore tend to finally lose the ability to support any of the load. Since the load must transfer to the front wheels of the rear suspension, penetration into a soil slope generally results. Vehicle immobilization occurs during most first attempts. Backing down and repeating attempts to negotiate such soil slopes usually modifies the slope configuration to the point where the vehicle can proceed over the obstacle. Referring to the lower right hand view in Figure V it is noted that a"bump-out" condition does not occur. First run negotiations of such gravel slopes with the experimental vehicle were not difficult to achieve with minimal experience.

Vehicle experience thus far indicates that a high wheel travel suspension does provide a cross-country mobility capability which a standard vehicle does not possess. The Powered Walking Beam system as described is not expected to provide a very high average speed over cross-country, but rather the ability to negotiate rough terrains where standard vehicles could be immobilized or provide extremely poor ride qualities.

A short color film taken by ENGESA showed the modified 2-1/2 and 5 ton (M35A2 and M813) cargo trucks during functional check-out runs in Brazil.

TOTAL TRACTION SUSPENSION FOR TRUCKS

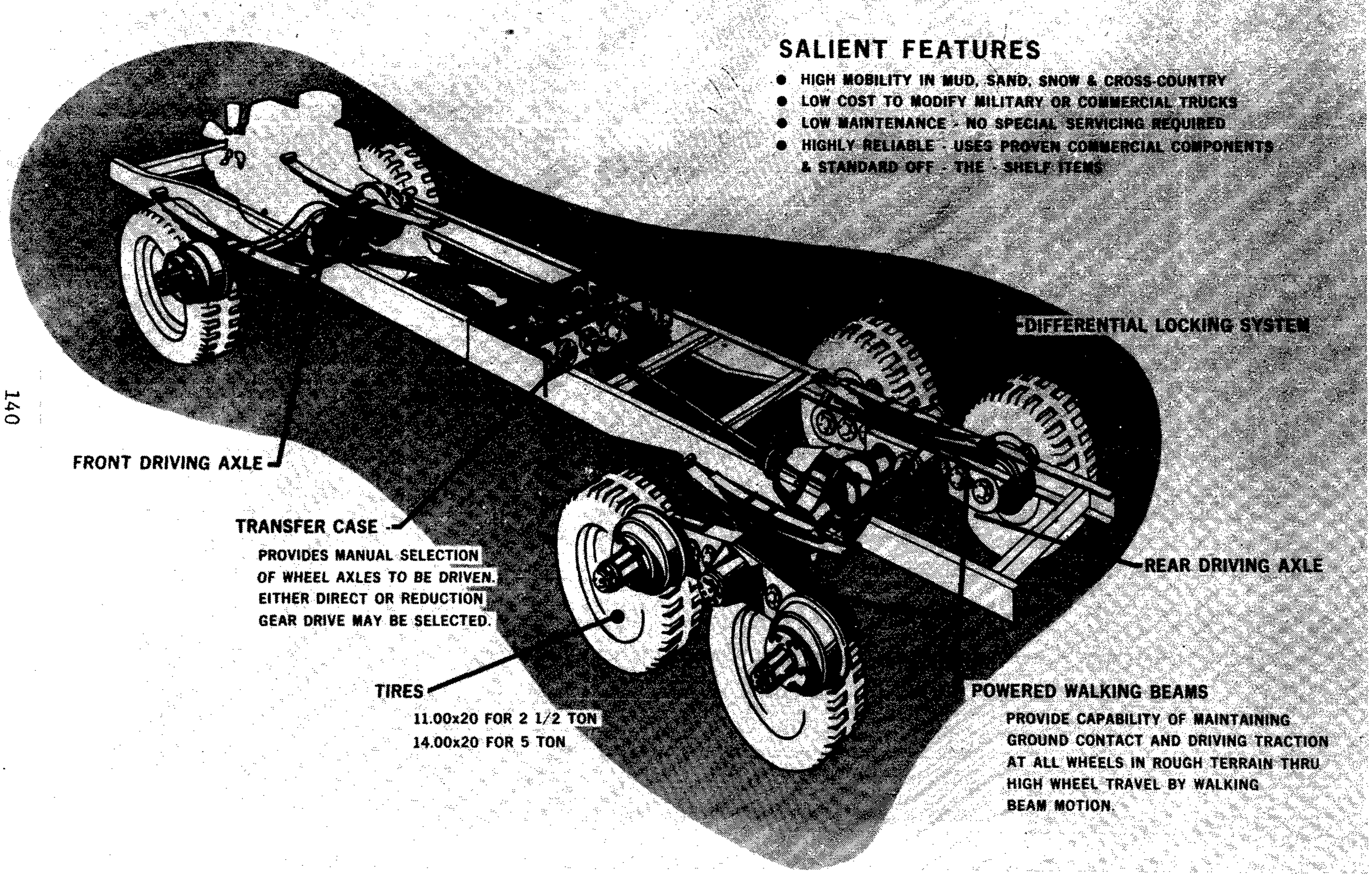


FIGURE I

TRANSFER CASE DRIVE
(ENGESA)

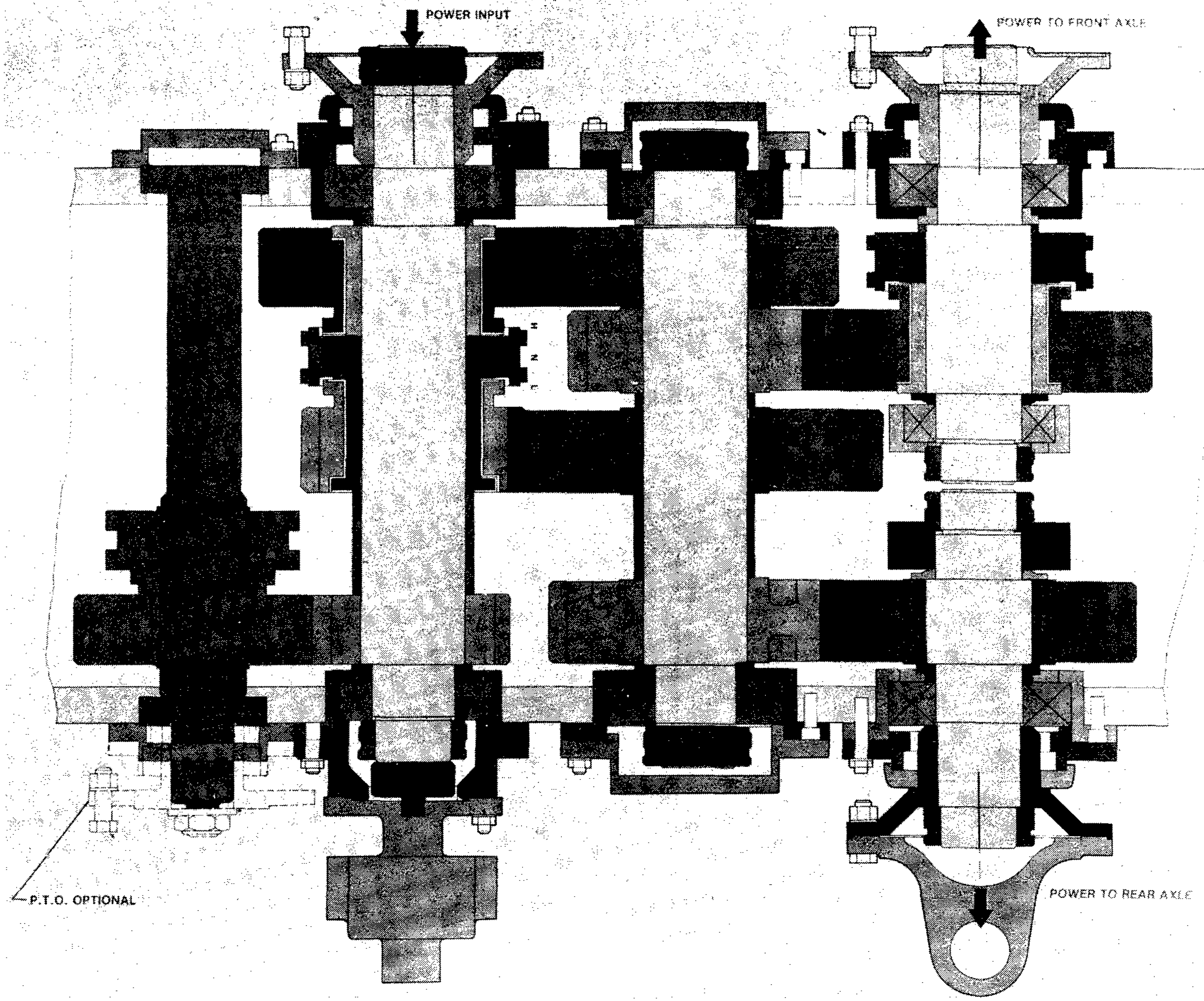


FIGURE II

REAR DIFFERENTIAL AND POWERED WALKING BEAMS

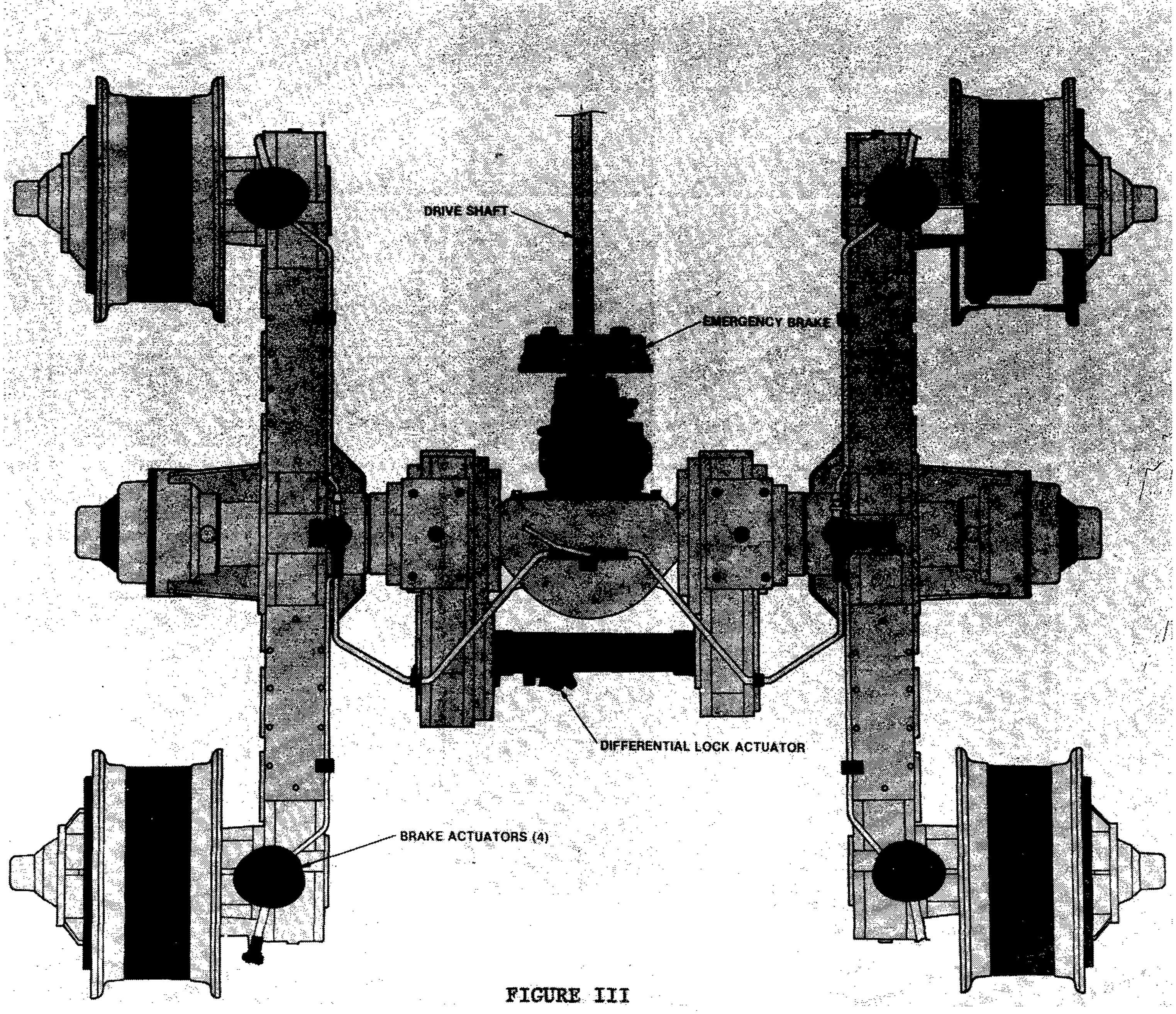


FIGURE III

WALKING BEAM DRIVE MECHANISM

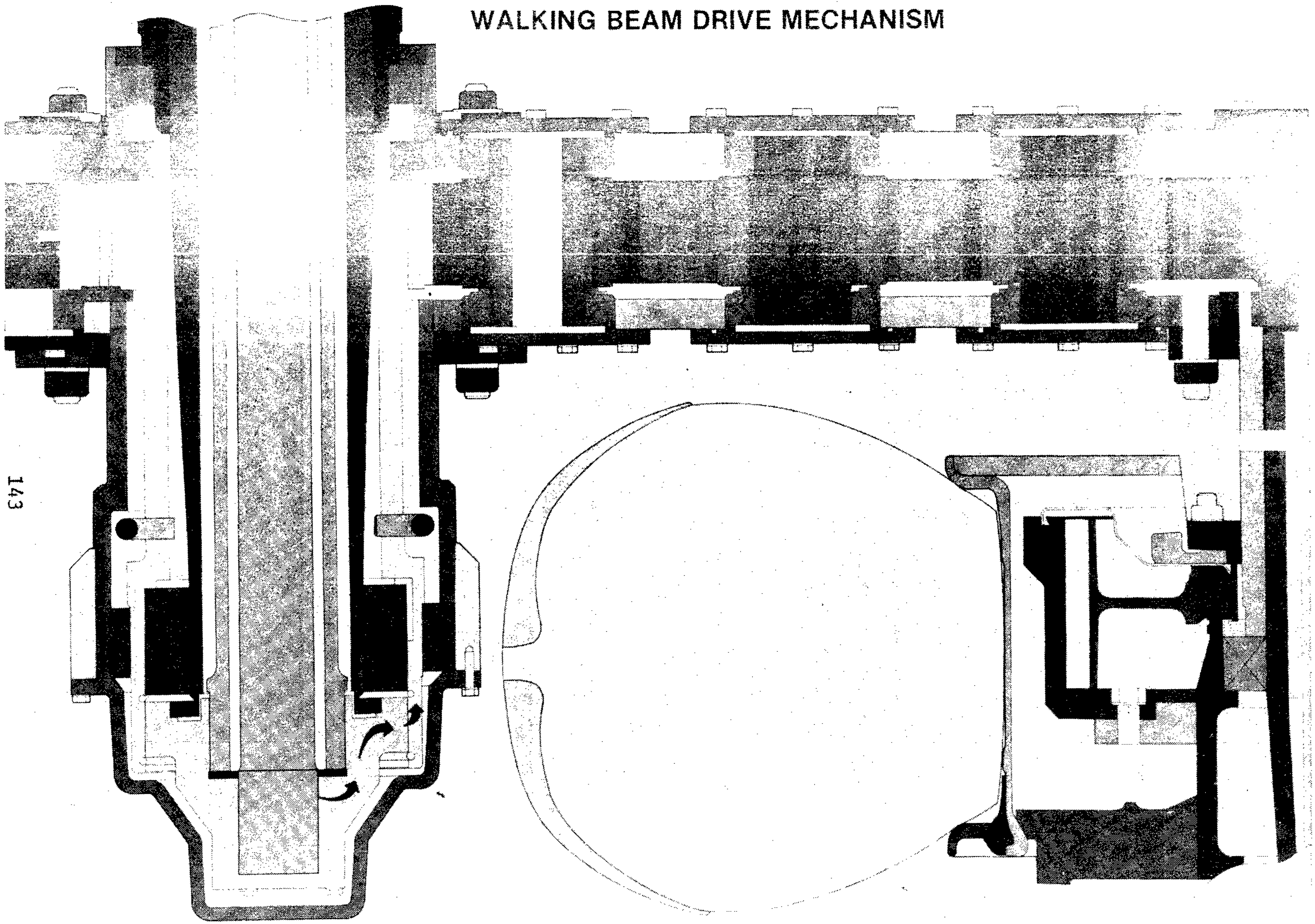


FIGURE IV

COMPARATIVE RAMP HEIGHT CAPABILITIES

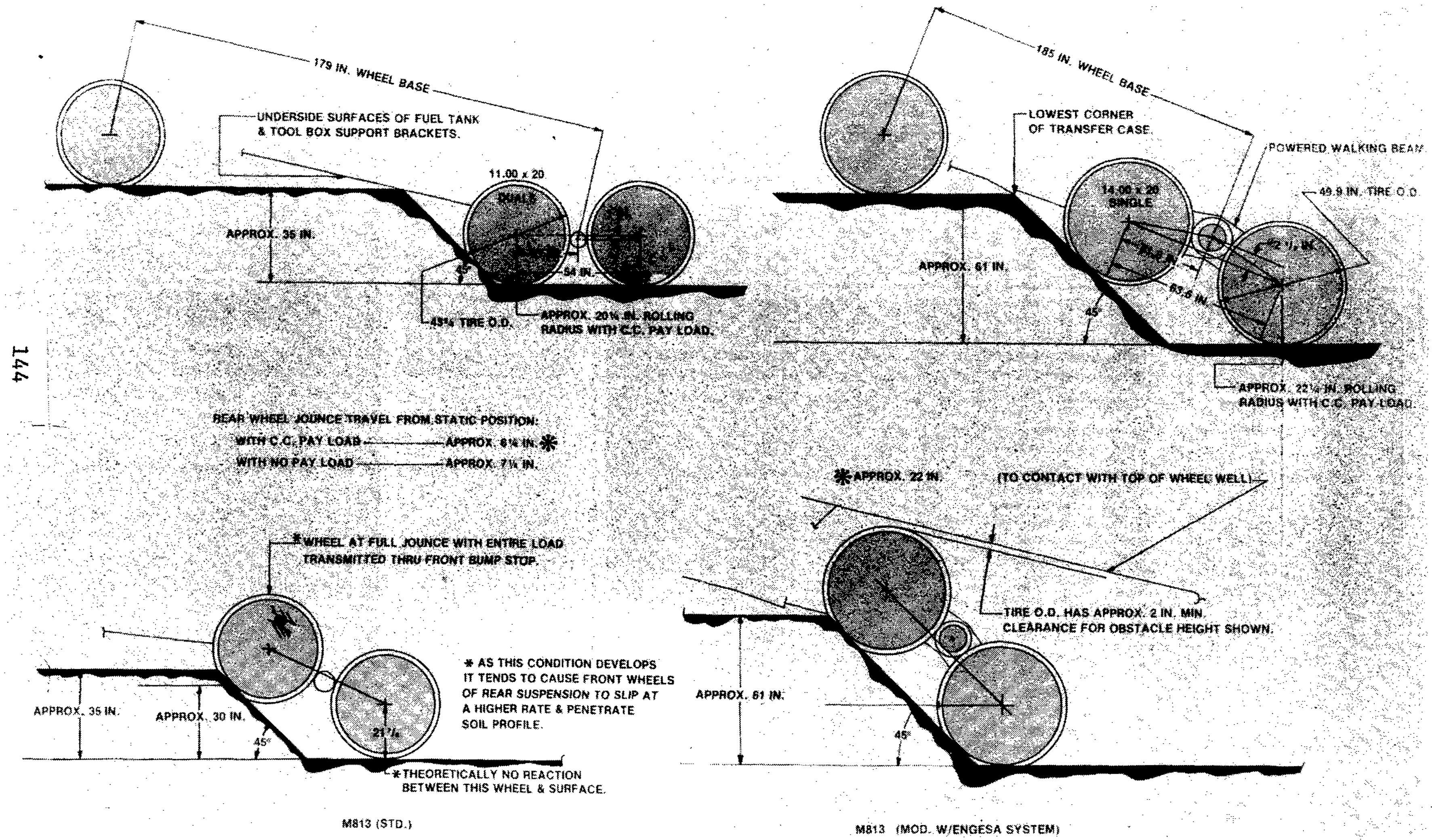


FIGURE V

CONFIGURATION OF TRACK AND SUSPENSION SYSTEMS
FOR FUTURE TRACKED MILITARY VEHICLES

Presented By

J. VON FREYMANN
FRG

J. von Freymann, Diplom Ingenieur (approximately equivalent to: Master of Science), Oberregierungsbaurat

55 Trier, April 30, 1975

CONFIGURATION OF TRACK AND SUSPENSION SYSTEMS FOR FUTURE TRACKED MILITARY VEHICLES

In order to answer the questions arising from this problem, the tasks to be performed by the track and suspension systems of military vehicles have to be defined, placing emphasis on main battle tanks.

The function of the track and suspension system is to contribute to the mobility of a vehicle. Mobility can be defined as follows:

> "Mobility is the capability of a vehicle to change its location within a given time, for a certain duration of time, including change of direction, under different road conditions".

Efforts are made to increase the protection factor for future vehicles by increasing their mobility to "high mobility". This "high mobility" is defined as follows:

> "High mobility is the capability of a vehicle to change its location over a long distance within a very short time, for a long duration of time, including random changes of direction, under as many existing combinations of road conditions as possible".

Before examining the configuration of a track and suspension system capable of achieving high mobility, one must determine which environmental parameters affecting mobility must be coped with by the track and suspension system. On-road operation which does not involve any problems can therefore be disregarded in discussing this problem.

Environmental parameters affecting mobility (pertaining to tracks and suspensions) are:

traction behavior

track and suspension performance

obstacle negotiation

These three quantities include:

traction behavior - tractive pull
- lateral forces

track and suspension performance
- behavior on microprofiled terrain
- behavior on ramp-like obstacles

obstacle negotiation
- ditch crossing capability
- obstacle climbing capability

The quantity of the transferable tractive pull and lateral forces (sufficient soil strength is assumed) is almost exclusively determined by parameters that are designed into the overall vehicle system. The following examples are given:

- track length on ground
- track width
- vehicle tread
- vehicle weight
- position of CG

A configuration of tracks in accordance with aspects of soil mechanics is nearly prevented by the requirement to use rubber pads in order to minimize road damages. As a further requirement rubber pads have to be replaceable in order to increase the life time of the track. In addition requirements for high life time and low weight affect the track design in such a way that a configuration aimed at good traction behavior will not be feasible. Optimizing the overall system is given a higher priority than optimizing an assembly.

Based on the track and suspension performance, results are obtained concerning the mean off-road speed which can be achieved by a vehicle on a microprofiled terrain, when the maximum speed is only limited by the vibrational behavior of the vehicle and the maximum acceleration tolerable to the driver. When determining the track and suspension performance, no singular obstacle, such as ditches, steps, water obstacles and woods will be used as a basis.

The determination of track and suspension performance is based on the assumption that off-road speed is primarily affected by the vehicle's behavior on sinusoidal-like profiles causing pitch and vertical vibration as well as by the capability to negotiate ramp-like obstacles at the highest possible speed.

In order to achieve the highest possible speed on sinusoidal-like profiles, it will be necessary to keep the pitch angle of the hull in the pitch resonance phase or the bounce distance of the hull in the vertical vibration resonance phase to a minimum, and to use all engineering means to keep the maximum pitch angle of the hull or the bounce distance as large as possible.

These objectives will be achieved by:

- high relative damping
- long spring travel
- long track length on ground
- large distance between idler wheel and road (with rear track drive sprockets)
- favorable hull configuration (hull contour does not project over track and suspension contour)

In order to negotiate ramp obstacles of different inclination at high speeds, the following requirements have to be met.

- long spring travel, high spring forces and high energy absorption of springs
- high damping forces
- long track length on ground
- large distance between idler wheel and road (with track drive sprockets at the rear of vehicle)
- large track rising angle

How can we now provide the characterists required for high track and suspension performance?

1. Long spring travel can be achieved by hydropneumatic suspensions, torsion bars and tube-over-bar springing. The possible total spring travel depends on the strength and overall concept.

2. High spring forces can be easily achieved by a hydropneumatic suspension. In the case of torsion bar springs and modified systems, only limited forces can be achieved by providing overload springs. This also applies to the energy absorption capability.

3. High relative damping and high damping forces can in principle be achieved by hydraulic means. Optimum spring-damping balance based on concept requirements is feasible. Due to its unfavorable articulation kinematics, telescopic shock absorbers do not conform to the present requirements. The high temperatures to be expected can only be controlled by an appropriate forced cooling system.

4. The track length on ground can only be increased within narrow limitations, since it is based on the concept requirements and will affect the length tread ratio.

5. If the idler wheel is installed at the highest possible location above the road, it will affect the vision, location of crew seats and hull configuration and can, therefore, only be considered within the overall vehicle concept.

6. A large track rising angle can be achieved by an appropriate arrangement of the idler wheel in relation to the first road wheel, which will, however, affect the rolling resistance when the vehicle bellies-in deeply.

The parameter "position of CG" which almost exclusively affects the obstacle negotiation in a main battle tank concept, can only be influenced to a very limited extent. The height of the idler wheel above the ground and the spring characteristics, which also has some influence on the obstacle negotiation, are usually established in accordance with other aspects.

Summarizing, it can be said that the capability of a vehicle (here MBT) to negotiate ditches and steps cannot be much influenced.

The track and suspension system of a vehicle can only be considered in its interaction with other problem areas, especially with problems of human factors engineering and stabilization of weapons and optical equipment. Consequently, an increase of mobility beyond the present state seems to be practical only if provisions are made to assure that -

- crew members are able to perform their functions (driving, observation, loading and aiming) during fast off-road vehicle operation)

- crew members are from the viewpoint of human factors engineering not subjected to excessive stress

- the stabilization of weapons and optical equipment can compensate for the disturbance introduced into the vehicle.

- controlability of the vehicle within the military formation is assured.

The implementation of recognized necessities with regard to the increase of mobility by changing the configuration of the track and suspension system will have such an impact on the overall concept that this would result in a complete change of the present tank configuration.

FLUIDIC CONTROLLED SHOCK ABSORBER

Presented By

M. STEELE
U.S.

Introduction

The TACOM Armor and Components Division has the Army mission of furthering technical progress in the field of tracked and wheeled vehicle suspensions. This necessitates a continuing effort to extend the technology base. Present and planned future suspension systems include the hydraulic or hydropneumatic type. Such systems require a high degree of control. Fluidic technology is a potential method of providing the hydraulic control advantageously.

Harry Diamond Laboratories has the mission for extending fluidic technology Army-wide. In cooperation with HDL, this division is pursuing a R&D program leading toward the application of fluidic technology to suspension system control. Under immediate consideration are suspension springing and vibration damping controls.

Present Damping Control Systems

A shock absorber (linear or rotary) is the standard vibration damping device now used on most wheeled and tracked vehicles. Spring loaded valves form an integral part of such damping devices, providing the control function. Opening and closing of orifices provides a hydraulic resisting force proportional to piston velocity. Once designed, constructed, and installed, damping characteristics (based on computed damping coefficients) remain fixed. The damping force provided by the damper does not match the demand at all velocities nor can the damper anticipate demand exceeding its capacity. Thus, matching performance over the entire operating spectrum is not provided and incipient or imminent failure to damp is not protected against.

Energy absorbed is dissipated as heat. When energy imparted to the device is more than can be dissipated, failure of seals and fluid occurs. Since the valves used are close tolerance fitted and spring loaded, operation leads to moving part failure. Overloading magnifies the failure rate. Problems inherent in present damping devices are; lack of finite control of damping response over the entire operating range, absence of control of energy absorption exceeding capacity, no anticipation of damping demand, and presence of moving parts.

Fluid Logic Damping Control Systems

Fluidics provides a potential solution to damping device problems. Fluidic devices can provide non-moving part continuous logic, anticipatory sensing and control. Inherently, such devices provide a high degree of RAM-D.

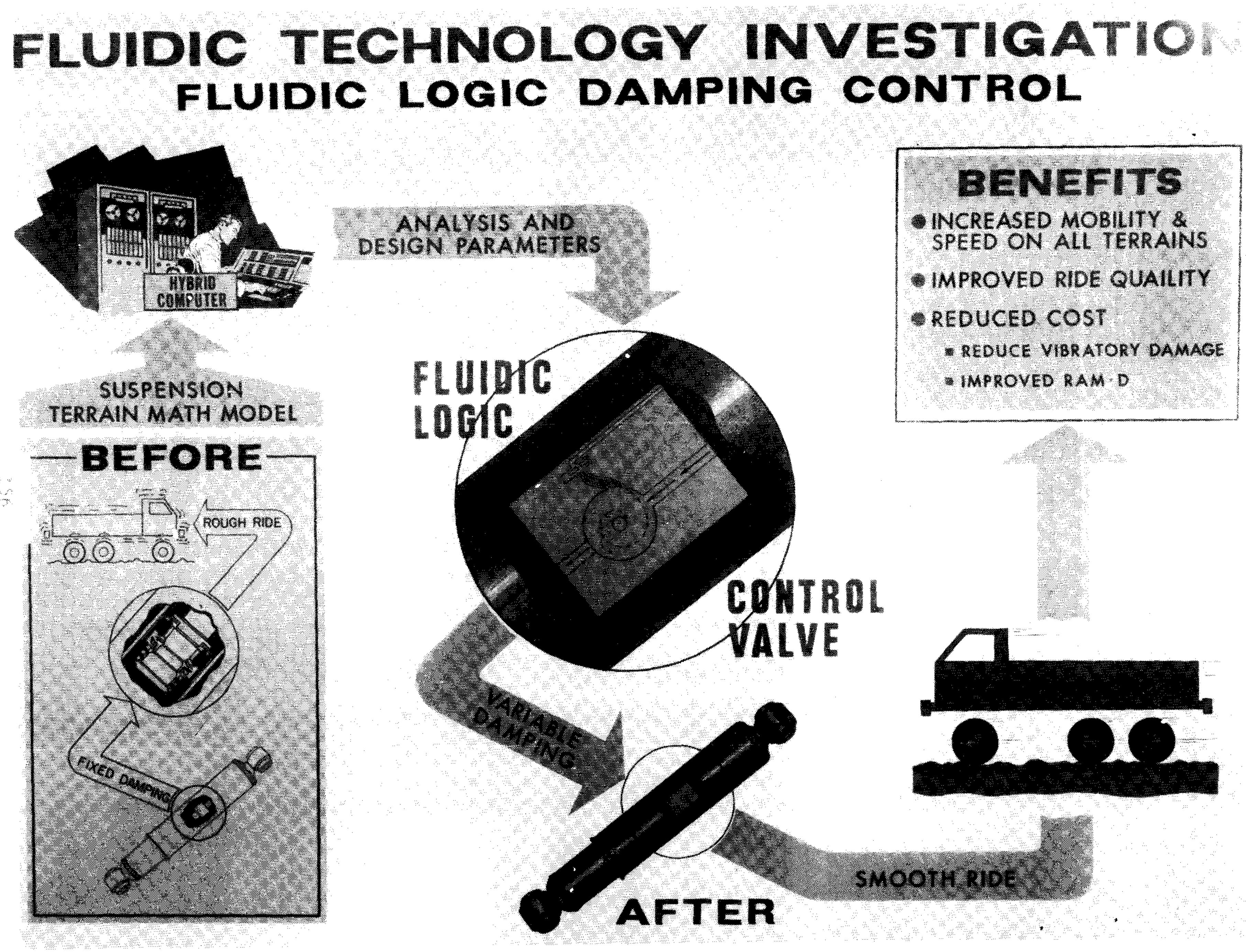
Current efforts have been devoted to hybrid computer analysis of the damping requirement at various vehicle speeds over varying terrains. Optimum damping parameters have been sought, based on not exceeding the level of vibration forces imparted to the driver in excess of his human tolerance level. This is measured in watts, with six watts selected as the evaluation criteria. Previous studies by TACOM have confirmed this human tolerance level.

The damping parameters arrived at by computer analysis will be used in the design of a fluidic logic control valve concept. The valve, so designed, is intended to replace the current spring loaded valve and include fluidic anticipatory (accelerometer) sensing. Thus, the valve will be capable of providing a damping force matching the vehicle damping requirement. When the anticipated requirement exceeds capacity, damping will be reduced. The driver (the final logic) will then, due to excessive vibratory force, take evasive action by reducing speed and/or changing direction, etc. When required damping returns to within the boundaries of damping capacity, matching damping forces will be once again instituted. This will tend to eliminate failures due to mechanical or physical inability to perform as required.

Since damping will always be available to the vehicle, and the damping force matches the requirement, improved ride quality will be experienced. This will result in increased mobility and speed on all terrains and reduced vibratory damage to the vehicle, its cargo and occupants. (When all advantages are summed, increased vehicle mission availability will result.)

Fluid Logic Springing and Damping Control

By continuation of the damping control effort, a hybrid computer study is underway to provide optimum design parameters for fluidic logic control of springing and damping. This extension of fluidic technology is expected to lead to concept design and development of a complete suspension system. In the future this development could lead to application in such developments as the In-Arm Suspension, Hydropneumatic Suspension, Compensating Track Tensioners, etc. Since fluidics as applied in such suspension systems draws little to no power (from the drivetrain) for control, it is expected to prove extremely advantageous. All other advantages previously mentioned (see Fluid Logic Damping Control) would also accrue.


FLUIDIC TECHNOLOGY INVESTIGATIO
FLUIDIC LOGIC DAMPING CONTROL
HYBRID COMPUTER
ANALYSIS AND DESIGN PARAMETERS
SUSPENSION TERRAIN MATH MODEL
BEFORE
ROUGH RIDE
FIXED DAMPING
FLUIDIC LOGIC
CONTROL VALVE
VARIABLE DAMPING
AFTER
SMOOTH RIDE
BENEFITS
INCREASED MOBILITY & SPEED ON ALL TERRAINS
IMPROVED RIDE QUAILITY
REDUCED COST
REDUCE VIBRATORY DAMAGE
IMPROVED RAM-D

IMPROVED SHOCK ABSORBERS AND VARIOUS TORSION SPRING SYSTEMS

Presented By

R. SIOREK
U.S.

US/FRG SYMPOSIUM PROCEEDINGS INPUT

Participant: Richard W. Siorek, Senior Program Engineer, AMSTA-RKT
Topic: Suspension Components Research and Development Project

SLIDE I: The objective of the project is to provide suspension building blocks to allow upgrading of currently fielded vehicles and to assure higher performance and RAM-D levels for developmental vehicles (i.e., our replacement fleet for the 1990 time frame). Much of our current work is aimed at maintaining the currently attained mobility levels while reducing component and system weight and keeping cost as low as possible. Improvement of RAM-D characteristics is a constant consideration.

The project is broken up into several tasks: the dampers task which addresses the problems of improving dampers with respect to RAM-D and performance; the roadwheels task which is charged with providing improvements in the life expectancy and a performance capabilities of tracked vehicle roadwheels; the spring task which is charged with evaluating new materials for spring application; and the embrionic controls task which is charged with development of futuristic active suspension systems. Of these, I will today present short composite summaries of the damper and spring tasks.

SLIDE II: Within the area of dampers there is primarily one major problem which precipitates the bulk of incidents causing equipment failure. This is one of energy management. A damper is basically an energy management device taking the physical hull motion caused by vehicle operation over terrain or, more basically, power created by the engine and converting or controlling that motion and power by turning it into heat. That heat in turn must be dissipated largely by radiation and convection and since in most cases it is generated faster than it is dissipated, the unit temperatures quickly rise to burnout levels.

SLIDE III: On tracked vehicles our approach has been development of the rotary damper. This unit with its increased fluid volume and controlled force generation capabilities has shown the ability to run cooler while producing proper damping over an extended range and for long periods of time. This has largely cleared up the problem for new development vehicles which will replace the current fleet.

SLIDE OFF: Linear dampers, better known as "shock absorbers", representing the bulk of the equipment currently fielded, are another story. Work is now underway in several areas to improve the life characteristics of these pieces. Improved fluids, better seal materials, and most importantly the development of "smart" valves, which shut down the unit by taking it off line hydraulically when it is overloaded, and other thrusts are attempting to correct the known obvious conditions of failure. In an attempt to correct the not so obvious, work is also underway on the development of a computerized analysis and prediction capability. The eventual objective of this effort will be to provide a capability to test damper designs before they are ever committed to hardware. At the present time the program can only handle heat flow throughput, but additions are being made to the program to allow it to accept and to generate force inputs and outputs.

Results so far have shown good correlation with actual Yuma field data. Current laboratory tests are also to be correlated with the computer output and the field data. Continuing efforts are planned in this area so that proper damping in terms of both magnitude and energy capabilities can be made available for all new development vehicles on an "off-the-shelf" basis.

SLIDE IV: The next task involves development of new springing systems, the upgrading of present springs and the evaluation of newly marketed materials for use in springs. Much effort in recent years has been directed to the design, testing, and evaluation of spring systems comprised of a torsion bar coupled in series with a torsion tube. Basic to this effort is the development of springing systems that will significantly increase the cross-country mobility of tracked combat vehicles. Torsion bar and tube systems of the type shown here are capable of increasing roadwheel travel from the typical six to seven inches of a simple torsion bar to fourteen to nineteen inches.

SLIDE V: Coupled with improved spring rates these systems provide the requisite characteristics needed to allow vehicles to traverse terrain at much higher speeds. Further they can accomplish this with much improved ride, thereby allowing greater equipment reliability and lessened crew fatigue.

SLIDE VI: Problems, of course, can be expected with any new, improved system that is fielded. There is, in increasing vehicle capabilities, a weight penalty to be paid. And there are still physical limitations to capability although these occur at a much higher level of performance. Finally there is the old gremlin of cost, the calculation of cost effectiveness and the determination of what you are willing to pay for increased performance.

Two of these items are being addressed under separate efforts under the springs task, weight, and cost. Capability limitations are currently a function of things other than the suspension componentry itself and there is little to be done in this area in the near future.

SLIDE VII: The first area being addressed is the cost and complexity of manufacturing the tube. Numerous techniques have been proposed and three of the most promising have been put under evaluation. Two of these have so far shown themselves to be non-viable in the near term future, the contractor's lofty promises not withstanding, on the basis of problems encountered during manufacture of the prototypes. Lab testing is currently underway and results are not yet available.

SLIDE VIII: Another thrust, as mentioned earlier, is the evaluation of new materials for use in springs. Titanium as shown here is an example of the types of things being looked at. Tests are underway on this material to identify the manufacturing and application problems that use of this type material may entail. Though this particular material may never see vehicle use, it is indicative of the type of work being pursued.

SLIDE OFF: There are other efforts also on-going under this task. They involve development of new testing and evaluation techniques, such as resonants testing, which could speed up life cycle testing by a factor of 60. There is the constant hunt for new materials for evaluation and other related efforts.

This is the basic work undertaken with these tasks. If there is interest or specific questions we will be glad to entertain them.

FUTURE SUSPENSION COMBAT VEHICLES

1975-1980 COMBAT FLEET IS LATEST STATE OF ART

NEED BUILDING BLOCKS FOR 1980-90

SUSPENSION FEATURES

MAINTAIN PRESENT MOBILITY ADVANCES AND

- REDUCE WEIGHT
- REDUCE COST
- IMPROVE RELIABILITY

DAMPENERS

PROBLEMS - INSUFFICIENT DEVELOPMENT

LIMITATION OF ENERGY ABSORBTION

HEAT DISSIPATION

DAMPENERS

OBJECTIVE - LIFE

PERFORMANCE

APPROACH - IDENTIFY MECHANICS OF FAILURE

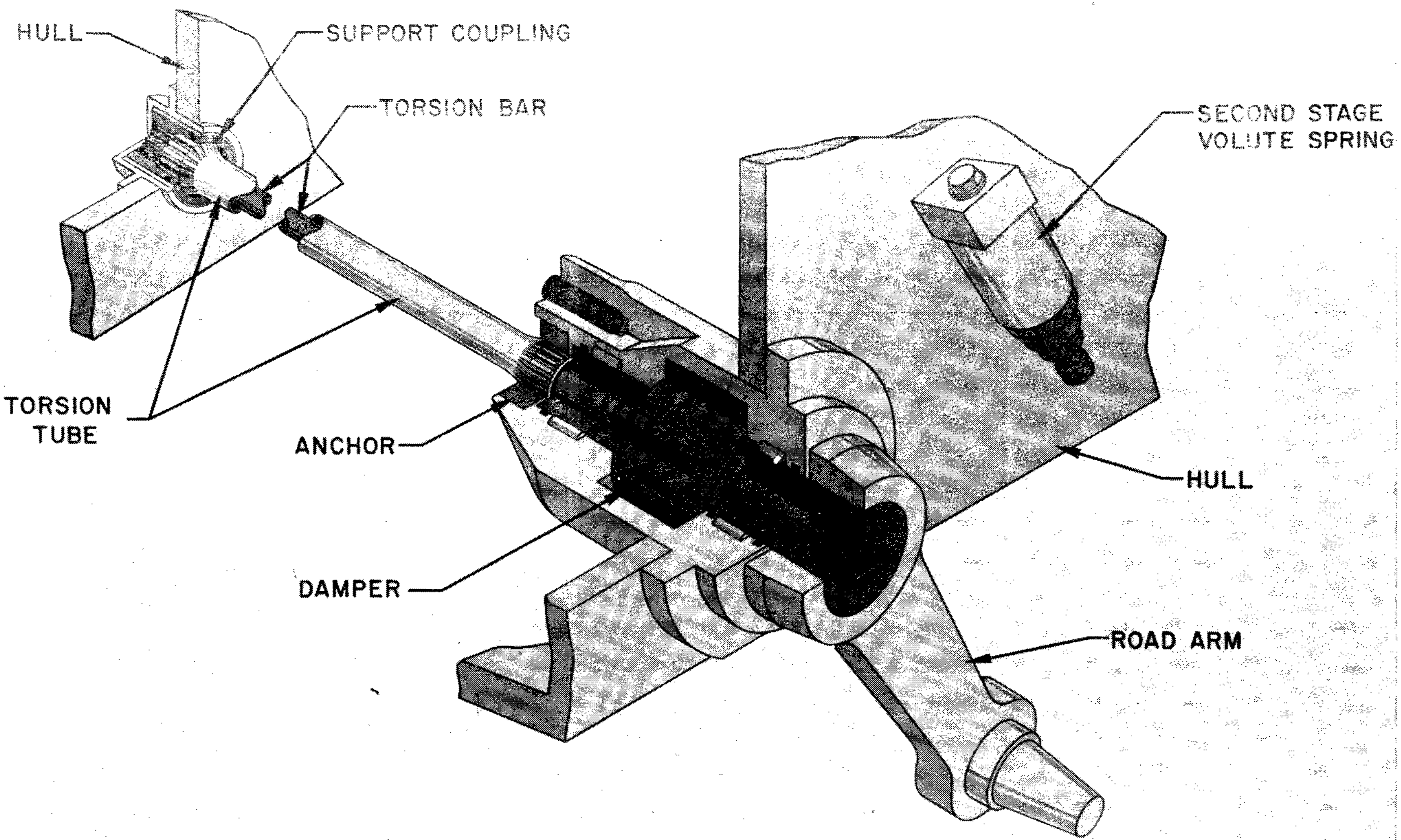
INVESTIGATE ALTERNATE DAMPERS

TORSION BAR & TUBE

OBJECTIVE - INCREASE CROSS-COUNTRY MOBILITY OF COMBAT VEHICLES

APPROACH - HIGH WHEEL TRAVEL
SPRING CHARACTERISTICS
TORSION BAR & TUBE SPRING

DUAL RATE SPRING SUSPENSION



SURFACE MOBILITY DIVISION

TORSION BAR & TUBE

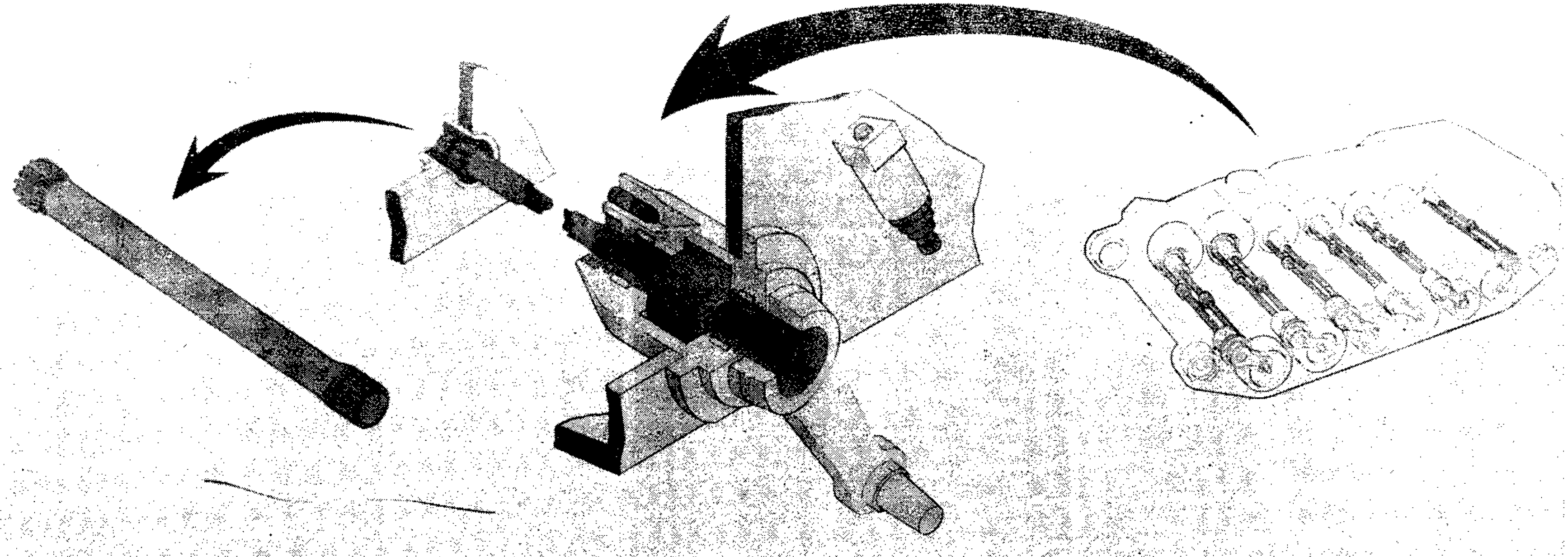
PROBLEMS - WEIGHT

BAR & TUBE TORSION SPRING

WIND-UP LIMITATION

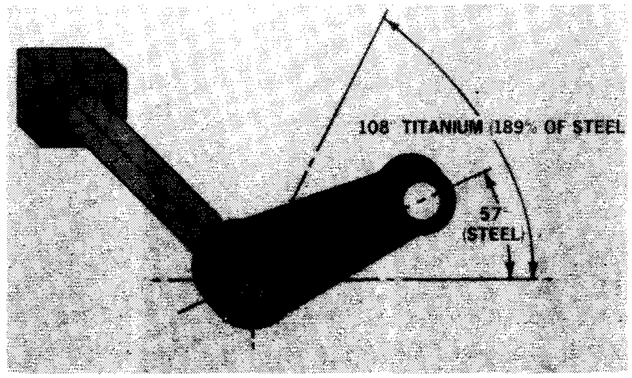
COST

MMCT: TORSION TUBE SPRING FABRICATION



PROGRAM OBJECTIVES	DEVELOPMENT OF FABRICATION TECHNIQUES
REDUCTION OF FABRICATION COSTS	HOT FORMING
IMPROVMENT OF COMPONENT CHARACTERISTICS	ROTARY SWAGING
EXPANSION OF PRODUCTION BASE	HOT UPSETTING
	WARM ,HYDRAFILM EXTRUSION
	OTHER FORMING PROCESSES

TITANIUM TORSION BARS TO IMPROVE RELIAB\LITY & DESIGN



PROPERTIES	E4340 STEEL	38644 TITANIUM
DENSITY	.283 LBS/IN^3	.174 LBS/IN^3 (62% OF STEEL)
COST	\$1.00/LB	\$6.00/LB
LIFE	45,000 CYCLES	40,000 CYCLES
S_S (TORSIONAL SHEAR)	140,000 PSI	125,000 PSI
G (MODULUS OF RIGIDITY)	11.6 X 10^6 PSI	5.9 X 10^6 PSI
T_S (TENSILE)	150,000 PSI	215,000 PSI
ULTIMATE	200,000 PSI	220,000 PSI
YIELD	185,000 PSI	206,000 PSI

COMPONENT STUDIES AND DEVELOPMENT EFFORTS MADE
IN THE FIELD OF TRACK AND SUSPENSION SYSTEMS
FOR FUTURE TRACKED MILITARY VEHICLES

Presented By

D. HAUG
FRG

Ministry of Defence
Rü VII 6 - Az 9o-25-5o

Bonn,den 1.April 1975
Ext. 4135

Subject:

U.S. Army/FRG Army Mobility Symposium from 8 to 11 April 1975

Component Studies and Development Efforts Made in the Field of Track and Suspension Systems for Future Tracked Military Vehicles

Gentlemen:

1. I would like to begin with a few general remarks.

The track and suspension system provides support between the ground contacting elements and the hull of a vehicle. One of its main tasks is to absorb shock forces generated by uneven terrain and to prevent them from acting on the hull.

In the past, the approach to these problems has been a more or less empirical one not based on reliable knowledge of the technical and physical factors and relationships involved. Today, with the requirement for greater mobility, such a time-consuming and costly approach is no longer considered satisfactory because it would not meet the need for ad hoc component development and ultimately fail to produce a vehicle that meets the operational requirements in an optimum way. Computer and test bench simulation techniques available today and field experiments using test vehicles and suitable measuring and evaluation methods offer ways of identifying economical solutions.

Currently, studies are being carried out in the Federal Republic of Germany on the following basic problems:

a. Terrain Features - Track and Suspension System Characteristics

This study is designed to investigate the possibility of establishing firm physical relationships between the stochastically distributed terrain features and soil types and the engineering design of a track and suspension system.

To this end, detailed terrain data are being compiled and evaluated. Initial results indicate that it may soon be possible to provide valid statistical data on the distribution of terrain features. Such data are essential not only for the design of the track and suspension system components but also for the design of a number of other vital vehicle components such as crew stations, optronic observation and sighting devices, gun stabilization systems etc.

While it will thus soon be possible to provide valid statistical data about terrain features, we are confronted with a number of problems in the field of soil mechanics which, for the time being, will not permit generalizations to be elaborated about soil types.

To obtain data on the behavior of and loads on the track and suspension system in high-speed (up to 20 m per second) cross-country operations involving high lateral accelerations (up to 7 m per sq. second) a wide variety of cross-country test runs are being conducted using various test vehicles.

b. Track and Suspension System Design Study

This study is to identify the design characteristics a track and suspension system for combat vehicles should have if optimum use is to be made of the technical possibilities for increased mobility.

By means of computer models the optimum design is determined through a process of almost infinitesimal variation of parameters and changing combinations of parameters. Through structural analyses the feasibility of the design is assessed. The design found feasible is reduced to hardware and installed in a test vehicle.

These studies will enable us to determine in a comprehensive and universally valid way possible improvements to the track and suspension systems of vehicles currently in service and what they will gain us in terms of mobility or what the design of the track and suspension system of a new vehicle to be developed should be like to ensure optimum mobility.

From what I have said so far you will have gathered that we envisage the following phases for the development of effective future track and suspension systems or system components meeting the performance requirements:

(1) Software Phase

- Computer simulation of the operating characteristics of track and suspension system components or entire systems.

(2) Hardware Phase

- Test bench simulation (under realistic conditions) using individual track and suspension system components or even entire systems.
- Field tests on individual track and suspension system components or entire systems installed in test vehicles.

2. Having made these preliminary remarks I will now discuss the current status and the possibilities for improvement of some track and suspension system components. There is not enough time to discuss all of them; so I will confine my remarks to the following three components:

- suspension and damping system;
- track adjuster, and
- track.

a. <u>Suspension and Damping System</u>

Basically, the track and suspension system operates through spring-supported roadwheels linked to the hull. Dampers acting parallel to the springs are to reduce the vibration of the vehicle body resulting from the spring suspension.

In view of the considerable increase in cross-country speeds that is envisaged for future combat vehicles, high demands have to be placed on suspension and damping systems, or new solutions to be found, since angular and linear disturbances of the hull should desirably be reduced or at least not be increased.

Both

- torsion bar suspension and
- hydropneumatic suspension

are today regarded as suitable future suspension systems, the latter currently being preferable from the point of view of technical performance, the former from the point of view of reliability and cost-effectiveness.

Therefore, we in the Federal Republic of Germany have decided to develop and improve both systems: The torsion bar system by using better materials and improved processing techniques, or by combinations, such as the dual torsion bar and the tube-over-bar system.

The hydropneumatic system through research on storage reservoir bulbs, e.g. on bulb materials, fatigue under pulsing loads, behavior at low and high fluid temperatures, and studies on a number of other problem areas, e.g. case sealing; relief valve, and comparative studies on one-cylinder and two-cylinder systems.

While in the hydropneumatic system the damping unit is integrated into the suspension system, the damper in the torsion bar system is mechanically linked to the roadwheel arm or the torsion bar.

The following types of dampers are used or being developed for use in connection with torsion bars:

- hydraulic linear shock absorbers;
- hydraulic rotary shock absorbers, and
- laminated rotary friction shock absorbers.

Owing to its design characteristics the rotary shock absorber has greater potential for development than the telescopic shock absorber which does not leave sufficient scope for improvement, particularly where roadwheel travel is long.

This can be demonstrated as follows: Calculations have shown that the optimum damper articulation point must be located on the extension of the secant through the two points of contact between the damper linkage and the roadwheel arm. Taking into account the geometry of the damper itself (length of travel and installed length), the optimum articulation points for modern vehicles would lie outside the track contour. On Figure 1, D1 is the optimum articulation point and D2 the marginal feasible articulation point. From this it is evident that for large bounce distances the damping force acting on the wheel arm will be zero because of the unfavorable conditions.

One of the major problems with dampers is their temperature behaviour. This is one of the reasons why in recent years the FRG, apart from seeking to further improve conventional suspension/damping systems, has been studying a new (Fig. 2) suspension system working on the energy storage principle. In field tests this new suspension system has been compared with conventional systems. The energy storage principle has been realized both in the dual torsion bar system

(Figure 3) and the two-cylinder hydropneumatic system (Figures 4 and 5. and tested in bench simulations and field experiments.

Compared with conventional suspension systems the energy storage type system promises to offer the following major advantages:

- There will be almost no loss of damper work and thus
  - almost no transformation of damping energy into heat; instead, the energy is stored and recycled into the vehicle's propulsion process so that
  - little propulsion energy is lost, which is especially important in cross-country operations.
- The generation and transmission to the hull of high-frequency vibrations will largely be eliminated which has considerable advantages as far as stabilized optical and weapon slaving devices are concerned.

Let me outline some of the results of the bench and field tests:

(1) Amplitude Increase Function

Figure 5 shows the amplitude increase function (the ratio of response vibration to primary vibration) of a conventionally dampened hydropneumatic suspension system compared with that of a hydropneumatic energy storage type system. In this respect, the performance of the energy storage type system is slightly better than that of the conventional system.

It should be noted - and this applies to all other results as well - that the energy storage type systems tested were basic experimental models. Efforts are being made to develop an improved system based on an analysis of these initial results.

(2) Vertical Acceleration

To evaluate vertical motion, accelerations were measured at the front and at the rear and in the center of gravity of the vehicles. Test measurements carried out on test vehicle ET 703 (Figure 7) showed that with the energy storage type suspension the vertical acceleration values in the vehicle's center of gravity are worse owing to the absence of vertical damping. By integrating additional dampers at the central pairs of roadwheels (in vehicles with conventional damping systems the loads on these dampers are comparatively low) it has been possible to achieve vertical accelerations very similar to those achieved with conventional damping systems.

Fig. 8 shows the vertical acceleration values for the front and the rear hull.The values for the energy storage suspension system are located between the values of the single torsion bar and the dual torsion bar combined with the hydraulic linear shockabsorber.

(3) Pitch behavior

Excessive pitch may cause the weapon to hit the end stops. Moreover, the travel distance of the front and rear suspension elements are limited to such a degree that hard stops of the idler wheels and the hull become increasingly frequent. Therefore, the pitch will have to be limited to an optimum level.

As Figure 9 shows, the pitch characteristics of the torsion bar energy storage type suspension system in test vehicle ET 708 are not inferior to those of conventional dampers even at the current state of development, and it should be added that the design provides great spring travel distances. By comparison, the pitch characteristics of a vehicle using a conventional damping system with only one effective torsion bar are markedly worse.

Tests conducted on test vehicle ET 703 have shown the hydropneumatic energy storage type suspension system to be markedly better; no further improvements were obtained by providing additional damping at the third and fourth roadwheels.

(4) Heat Generation

The most serious problem with conventional damping systems, and particularly the hydropneumatic system, is that of heat generation through the transformation of kinetic vibration energy into heat. With increasing mobility this problem almost presents a sort of "sound barrier".

With the energy storage type system this problem cannot arise.

Figure 10 shows the temperature increase curves obtained in cross country test runs carried out at Schleißheim near Munich. They show the energy storage type system to be clearly superior in this respect.

(5) Loss of Energy

Loss of energy is caused by the transformation into heat of the kinetic vibration energy generated in the suspension system. Bench simulation of terrain carried out on a torsion bar element alternately equipped with an energy storage unit and a linear telescopic shock absorber showed a ratio of 1:.5 between the amount of energy lost with the energy storage system and the amount of energy lost with the shock absorber. The amount of energy lost for the energy storage suspension exclusively consists of bearing friction. If one takes into account that this energy has to be produced by the propulsion system, it is evident that the low-loss energy storage type system is preferable in this respect as well.

(6) Comparison between Conventional and Energy Storage Type Suspension Systems

Figure 11 shows an essentially qualitative comparison between conventional and energy storage type suspension systems. It can be seen that neither system can at this time be shown to be absolutely superior. Rather, the superiority of one system or the other depends on the relative importance assigned to the various criteria.

A particularly favorable feature of the energy storage type system in this connection is the low level of heat generation and the low loss of energy. These are characteristics that are essential in high mobility vehicles. As regards the prestabilization of the hull a specific development effort may produce improvements. Improvements in the cost effectiveness and reliability of the energy storage type suspension system appear also possible.

b. Track Adjuster

Especially during circular maneuvering and weaving movements in rough terrain track tension must not be allowed to slacken to such an extent that the track lifts off the idler wheel and the first roadwheel. This might result in the track being thrown off or at least cause considerable damage to roadwheel tires and the center guides of the track. To prevent this from happening high-mobility vehicles should in any case be equipped with a track adjusting system.

The semiactive combined mechanical and hydraulic system sketched out in Figure 12 has completed a first brief test in test vehicle ET 703 to obtain measurement data. The results achieved are extremely satisfactory. Figure 13 summarizes the results of the film evaluation of the track slack both during the approach to an obstacle and the bounce following the crossing of the obstacle. Shown are the averages of several test runs at varying speeds both for a system using this track adjuster and, for comparison, also for a rigid system.

From the hydraulic pressure in the track adjuster, the displacement of the adjuster piston, the bounce of the first roadwheel, and the associated geometrical conditions a rough calculation can be made of the track tension (leaving the internal friction of the track adjuster and dynamic factors out of consideration). Figure 14 shows this tension as a function of time for a vehicle moving over a corrugated course, again both for the rigid system and for a system using the semi-active track adjuster. Figure 15 shows the frequency distribution of track tensions for linear cross-country movement (over the corrugated course mentioned) and Figure 16 for cross-country weaving movement at 3 different speeds.

It was not possible in these tests to vary the damping of the track adjusting system. There is, however, reason to believe that it may even be possible to increase the effectiveness of the track adjuster by introducing appropriate throttle points in the storage line or changing the diameter of the line in accordance with the results of theoretical optimization efforts.

The film that will be shown afterwards will give you a visual impression of the comparative field tests carried out on test vehicle ET 703 with regard to hydropneumatic suspension and the track adjuster.

c. Track

The studies currently under way in the Federal Republic of Germany in respect of tracks will be dealt with by Mr.Rausch in the subsequent presentation.

3. Problems of Shielding the Crew from the Effects of Hull Motion

The crew of a high mobility combat vehicle will be able to accomplish its tasks only if it operates under tolerable conditions. The increased speed of such high-mobility vehicles leads to increased vibration of the entire vehicle body. Since this vibration will not always be absorbed by an advanced track and suspension system (which may sometimes even be inexpedient), the crew must be provided with suitable seats that shield it from such vibration.

There are several known methods of achieving this aim. In test vehicles FVT, VT and GVT sprung individual seats or platforms accommodating serveral seats were installed. If presently available components (spring and damper) are used, long spring travel will be needed for effective suspension which will require additional vehicle space. The seat suspension used in the present case employs a combination of a spring damper system with an additional frictional damper and automatic adjustment of the static position of the seat. The frictional damper prevents seat suspension action until a predetermined acceleration is reached so that it is possible to obtain adequate shielding of the crew from hull motion with a total spring travel of about 130 mm. A model of this system has been tested in a vehicle. The observation system used is a stabilized articulated panoramic periscope (Figure 17) whose eyepiece is fixed to the sprung seat. The relative motion between the observer's eye and the eyepiece is insignificant so that even with a small exit pupil continuous target tracking during high-speed cross-country movements is possible as has been demonstrated by laser firing tests. To permit effective target tracking not only the observer must be shielded from hull motion but the controls as well, preferably by attaching them to the sprung seat.

During cross-country test on the FVT the following vertical accelerations were measured:

	50 % value	68 % value (1σ)
Seat mounting in the hull	6.4 m/s^2	7.8 m/s^2
Sprung seat bucket	4.0 m/s^2	4.9 m/s^2
Observer's eye	3.2 m/s^2	4.1 m/s^2

If the acceleration value measured at the hull is 100 per cent, the reduction ratio is 100:63:50. It should be noted here that the seat was not optimized at the time when these tests were carried out (June 1973). With improved seats we hope to achieve a reduction of the acceleration to about 30 per cent at the observer's eye.

Figure 18 shows another observation system for the commander or the gunner. Here, too, the observer sits on a sprung seat.

For combat surveillance a stereoscopic revolving mirror vision device is used, whereas for target acquisition a seat-mounted TV monitor is used which is fed from a stabilized panoramic periscope with an attached TV camera. Responsive target assignment from the revolving mirror device to the periscope/monitor is achieved via a diopter sight.

Thank you.

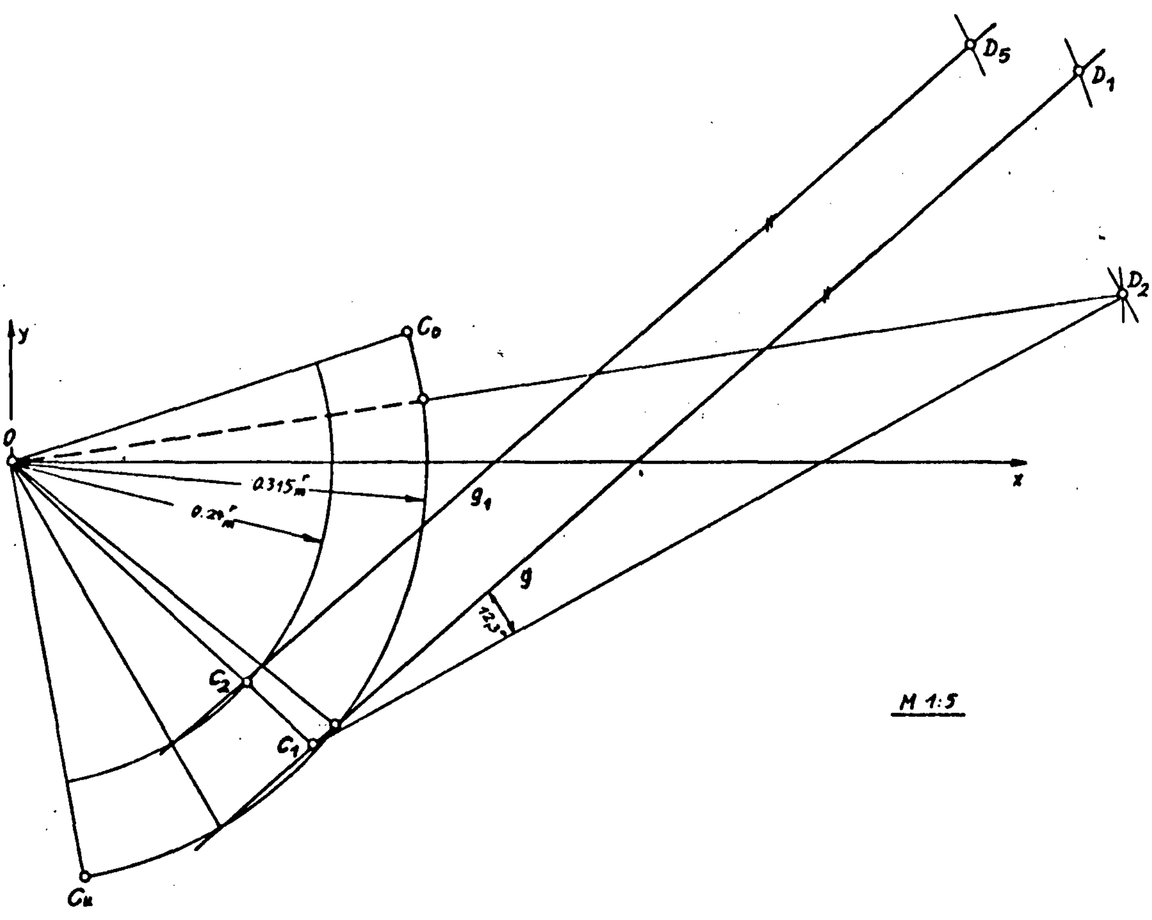
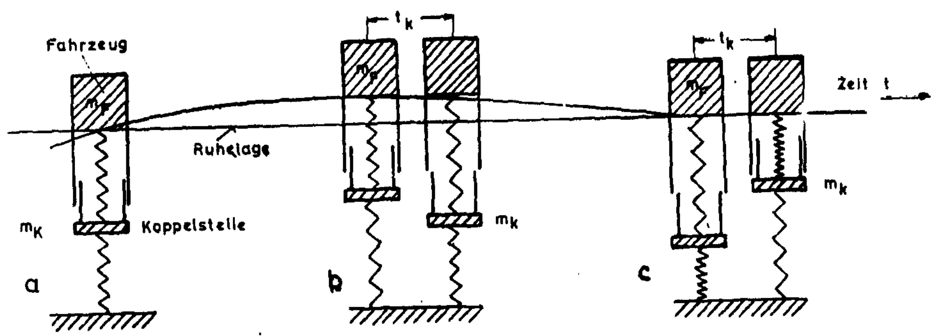


Fig. 1

Übersichtbild zur Optimierung des Schwingenanlenkpunktes



Schematische Darstellung der Arbeitsweise

der SSF

Fig. 2

IABG-TV
Lichtenau

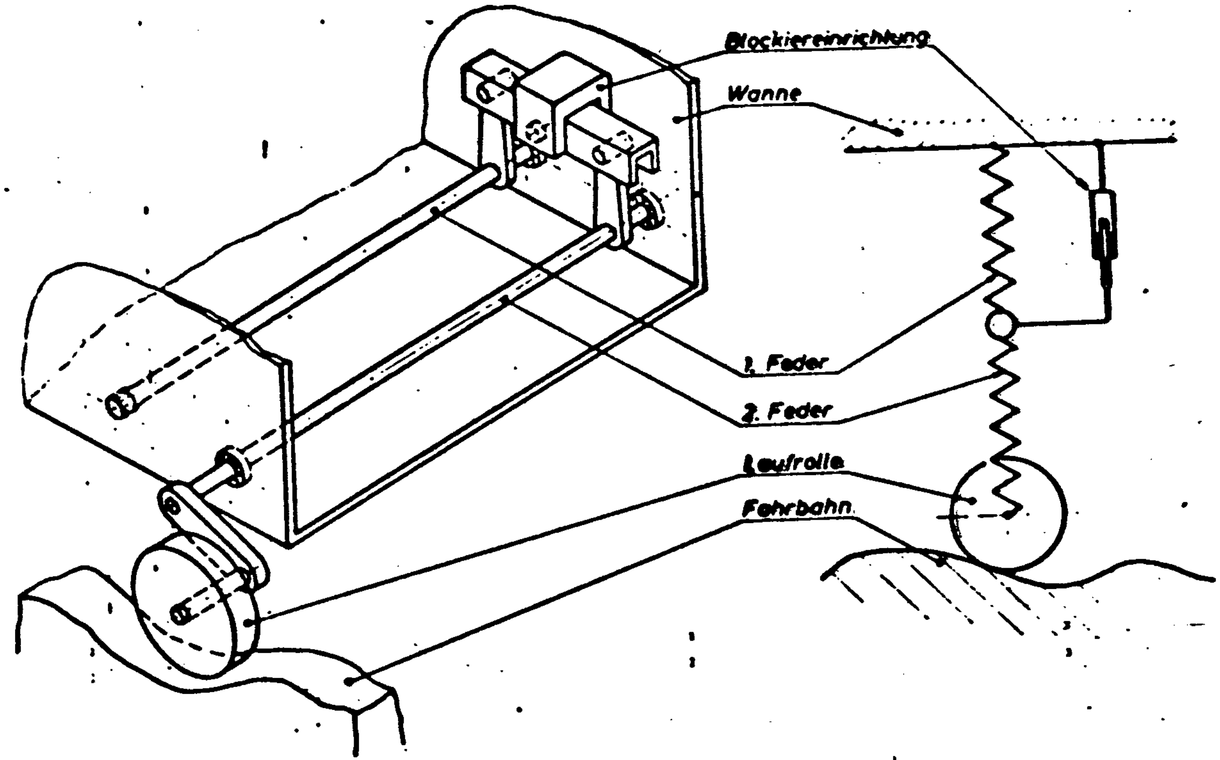

Blockiereinrichtung
Wanne
1. Feder
2. Feder
Laufrolle
Fahrbahn

Fig. 3

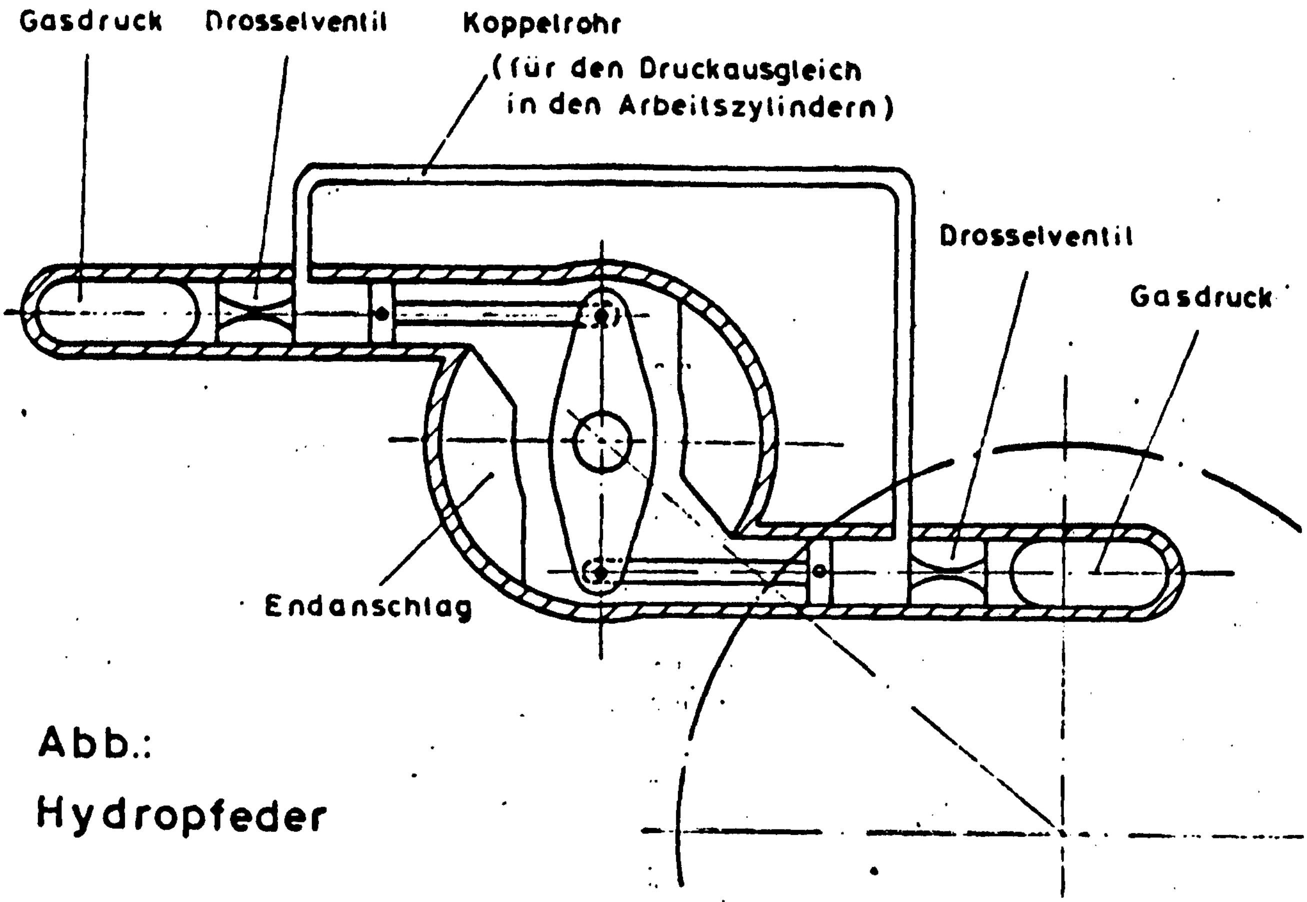


Abb.:
Hydropfeder

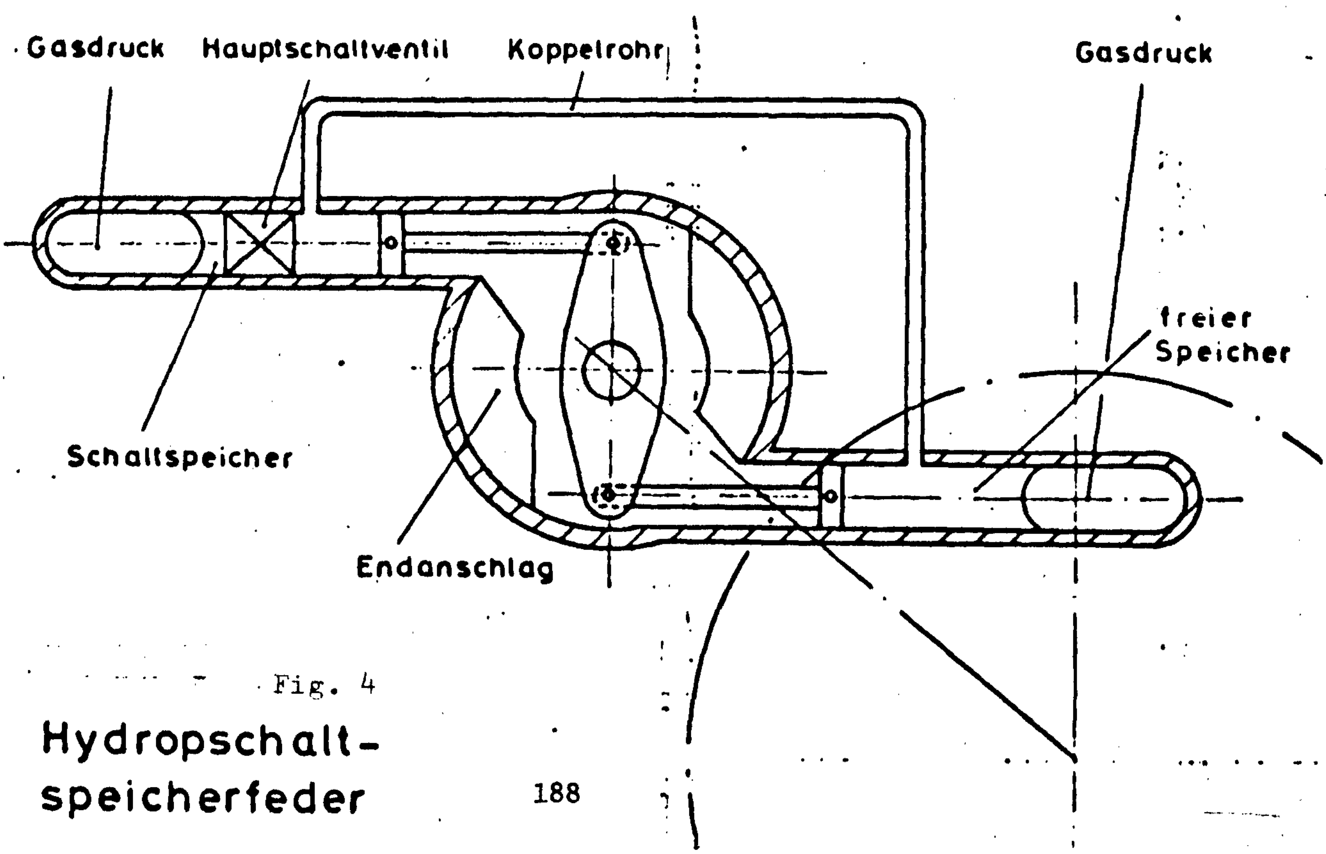
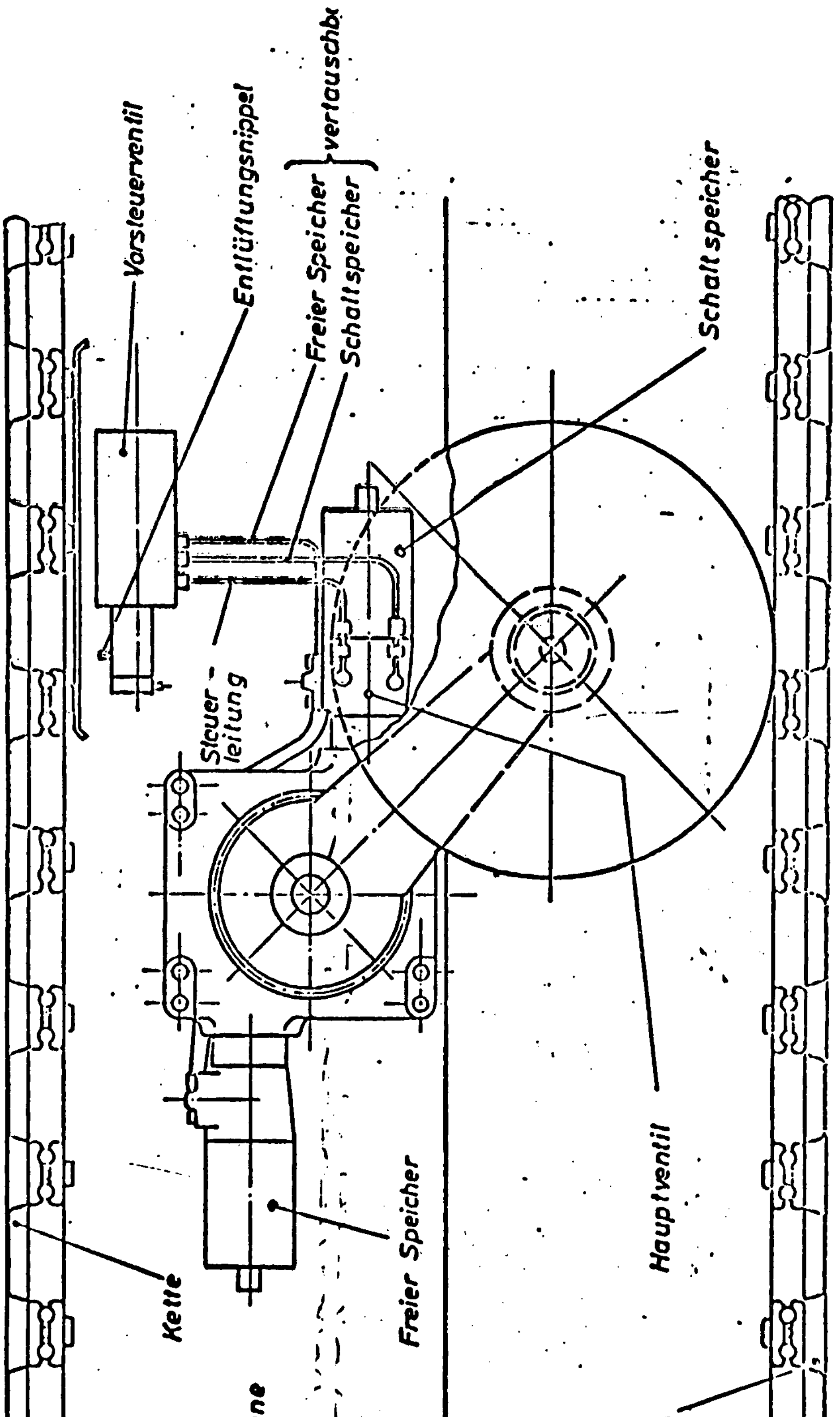


Fig. 4

Hydropschalt-
speicherfeder



Vorsteuerventil
Entlüftungsnippel
Freier Speicher
Schaltspeicher
vertauschbar
Schaltspeicher
Steuer-
leitung
Hauptventil
Freier Speicher
Kette
Wanne
Kette

Fig. 5

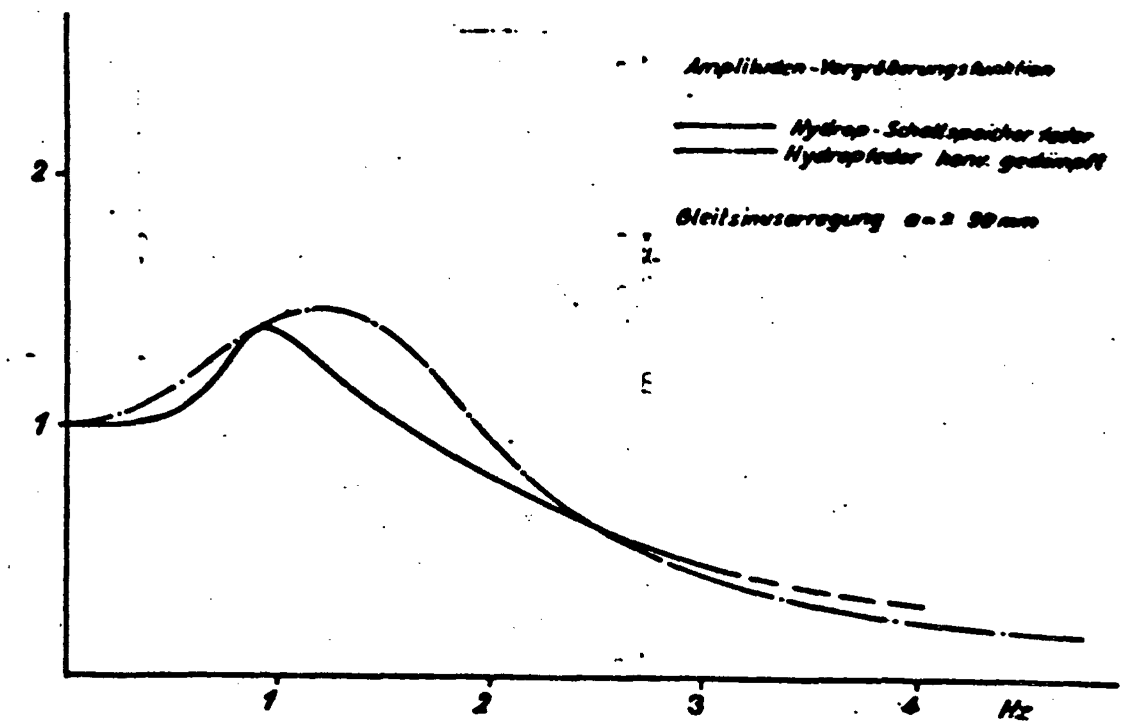


Fig. 6

Vergleich der experimentell ermittelten Überhöhungsfunktion für konventionelle Hydropfeder und HSSF

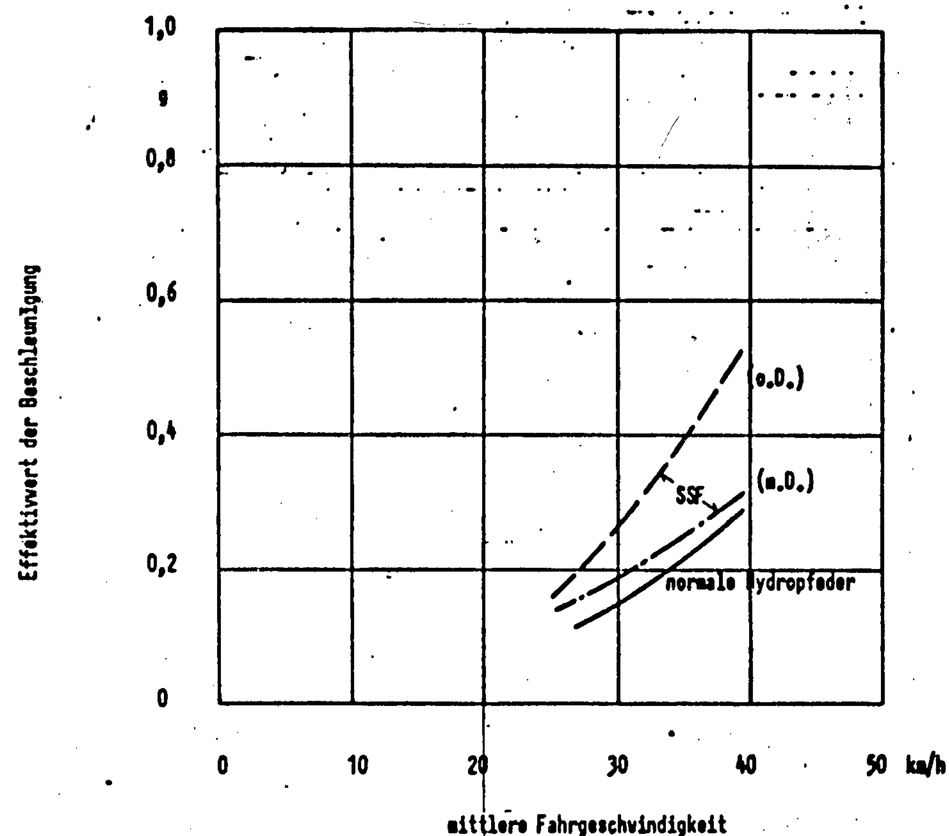


Fig. 7

Vertikale Beschleunigung im Schwerpunkt des ET 703 mit normaler Hydropfederung und mit Schaltspeicherfederung (SSF)

ohne Dämpfungsventile (o.D.)

mit Dämpfungsventilen an den Schwingen 3 und 4 (m.D.)

(Versuchsfahrten auf dem Gelände in Lichtenau)

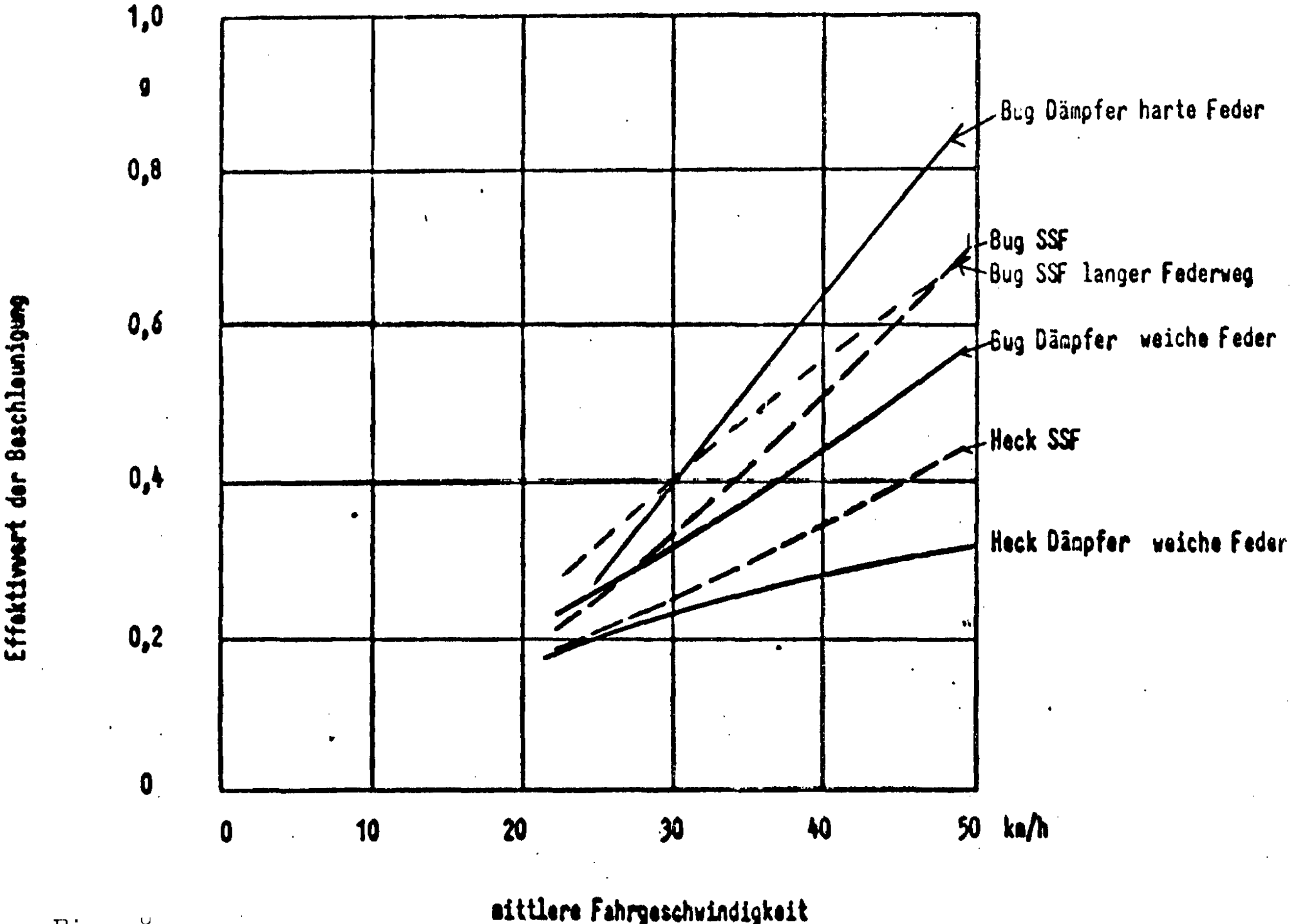


Fig. 8

Vertikale Bug- und Heckbeschleunigung des ET 708 mit konventionellen Lineardämpfern und mit Schaltspeicherfederung
(Versuchsfahrten auf dem Gelände Schleißheim)

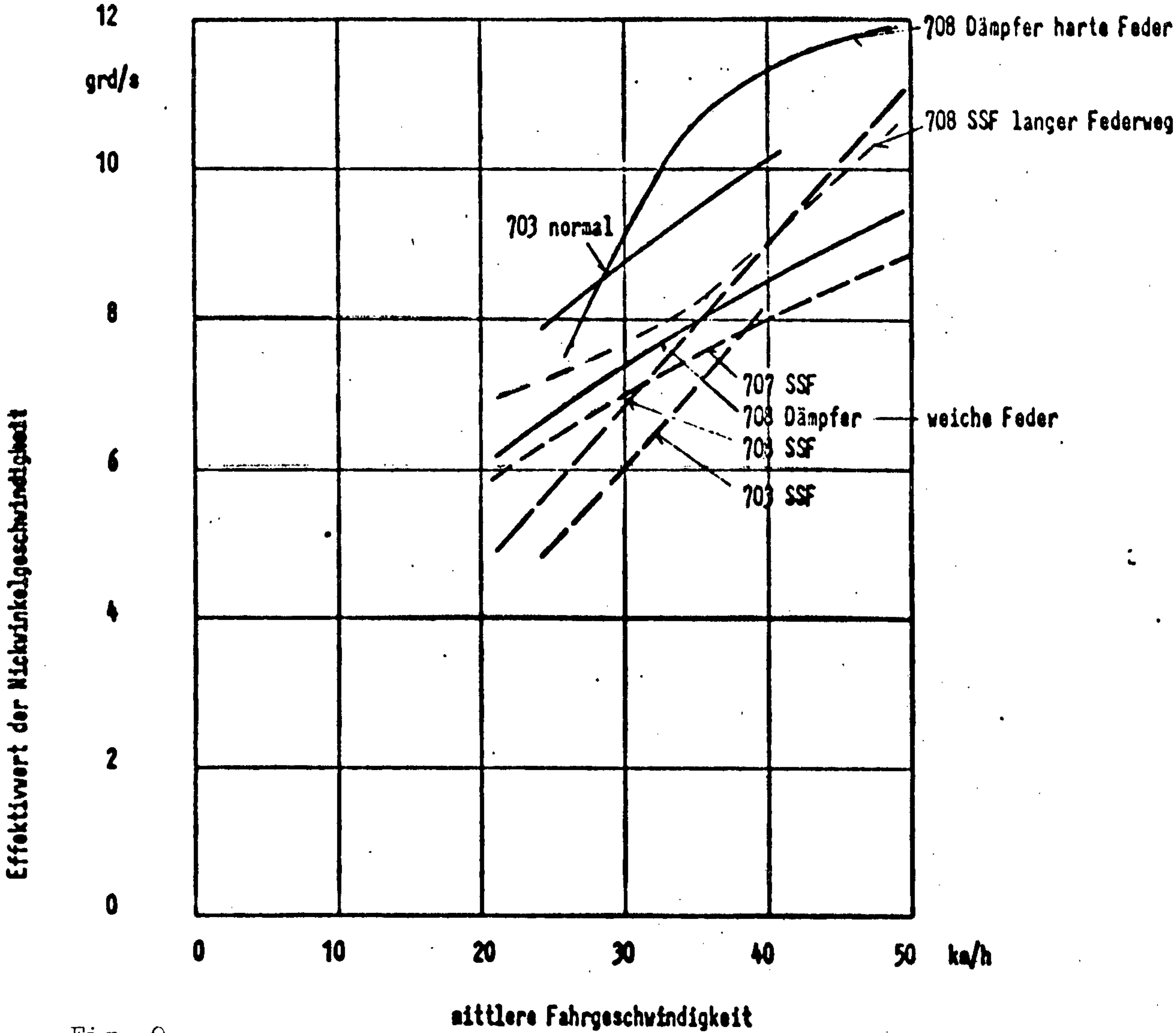


Fig. 9

Vergleich der Nickschwingungen der Fahrzeuge ET 703, ET 707 und ET 708 mit konventioneller Dämpfung und mit Schaltspeicherfederung (SSF)

ET 707: nur Meßwerte mit SSF vorhanden

ET 703: in Ausführung SSF mit zusätzlicher Dämpfung an Laufrolle 3 + 4 ergibt dieselben Werte wie ohne Dämpfung

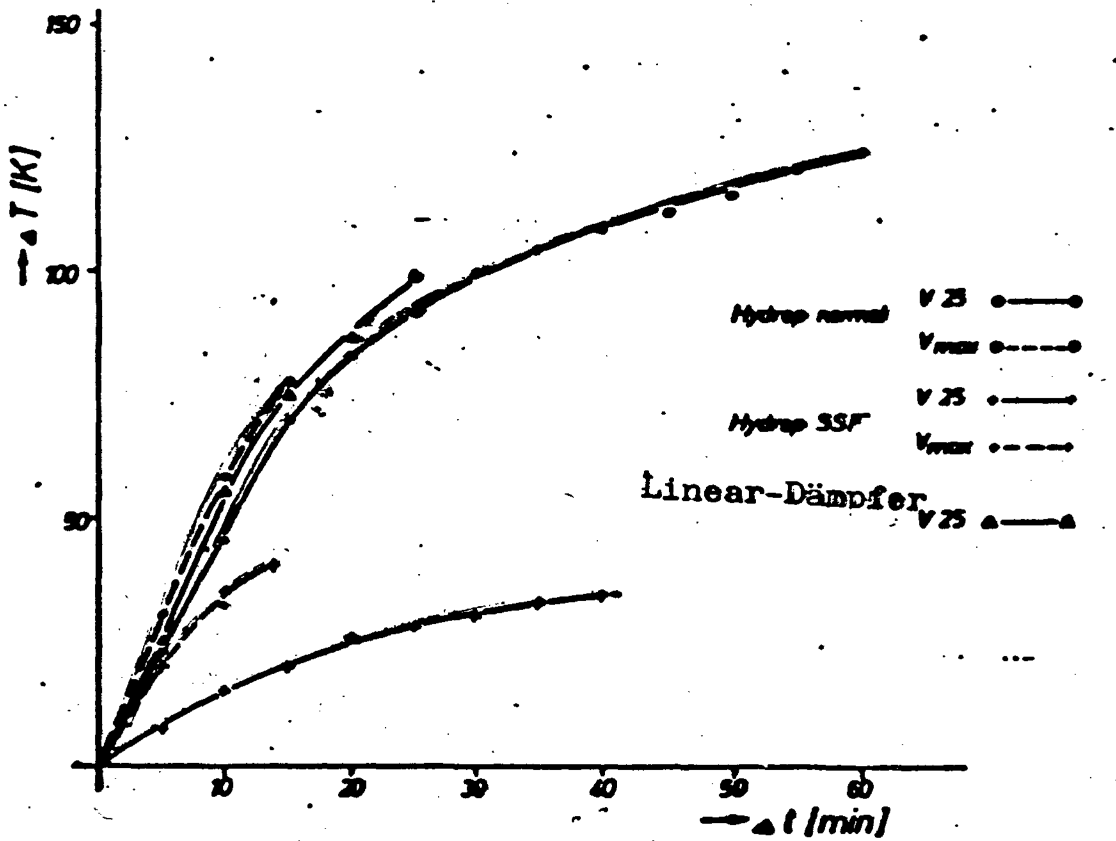


Fig. 1o Temperaturanstieg am Dämpfer bei konventioneller Dämpfung u. SSF

	Konventionelle Dämpfung		Schaltspeicherfeder	
	Drehstab	hydropneumatisch	Drehstab	hydropneumatisch
möglicher Federweg (mm)	ca. 320	550	485	550
Hubschwingungsabbau	Bezugsgröße	besser	im höheren Geschwindigkeitsbereich besser	schlechter
Nickschwingungsabbau Anfahren von Hindernissen	Bezugsgröße	schlechter	teils schlechter teils besser *	besser
Temperaturproblem	Bezugsgröße	größer	nicht vorhanden	wesentlich geringer
Bauvolumen	Bezugsgröße	kleiner	größer	kleiner
technischer Aufwand	Bezugsgröße	höher	höher	viel höher
Kosten	Bezugsgröße	höher	höher	höher
Verbesserung durch weitere Entwicklung	geringfügig	wesentlich	möglich	wesentlich

Fig. 11

Vergleich von Drehstab- und Hydrop-Federung mit konventioneller Dämpfung und Schaltspeichereffekt, bezogen auf Messungen mit ET-Fahrzeugen.

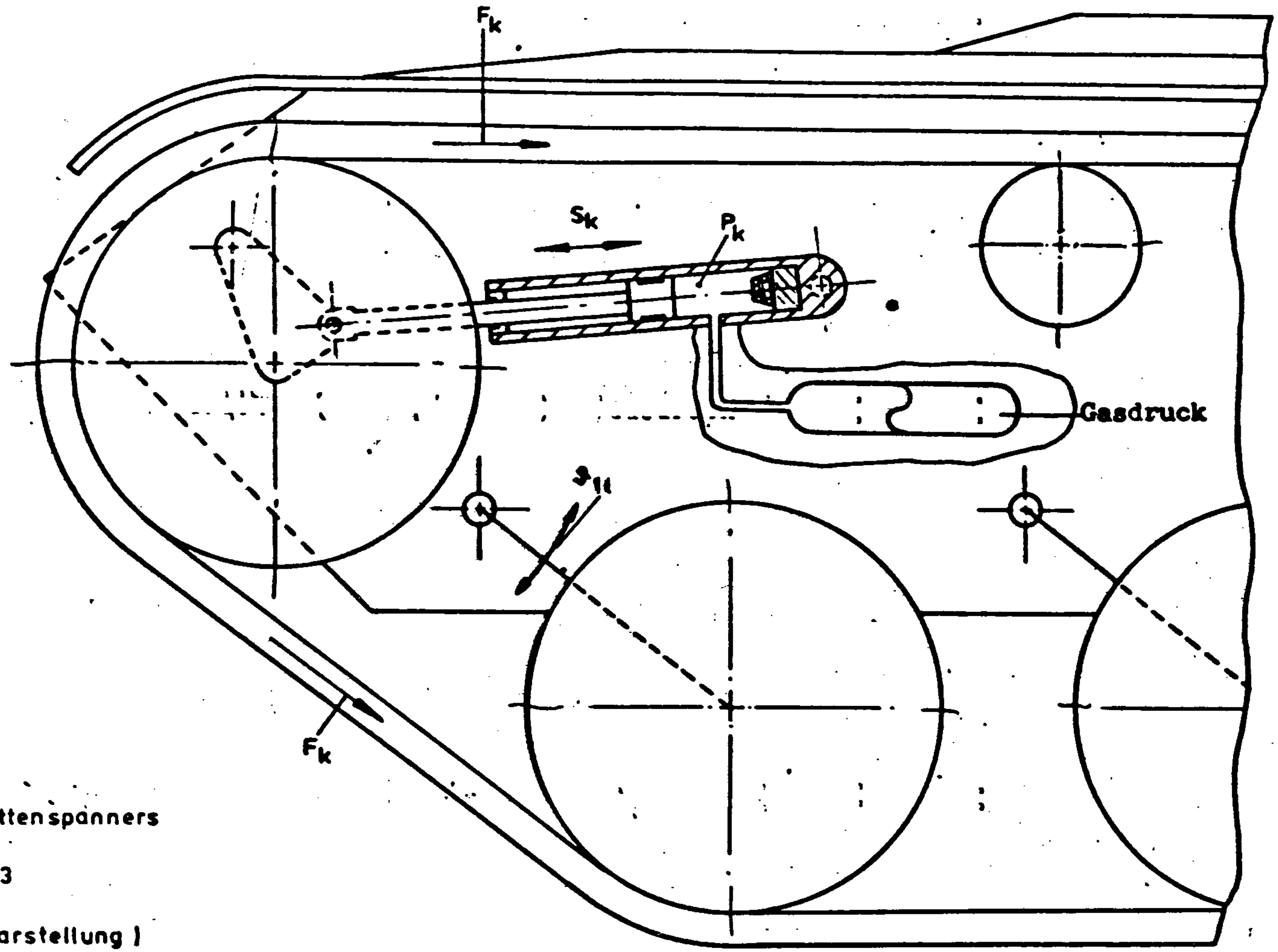


Fig. 12

Anordnung des Kettenspanners

am ET 703

(schematische Darstellung)

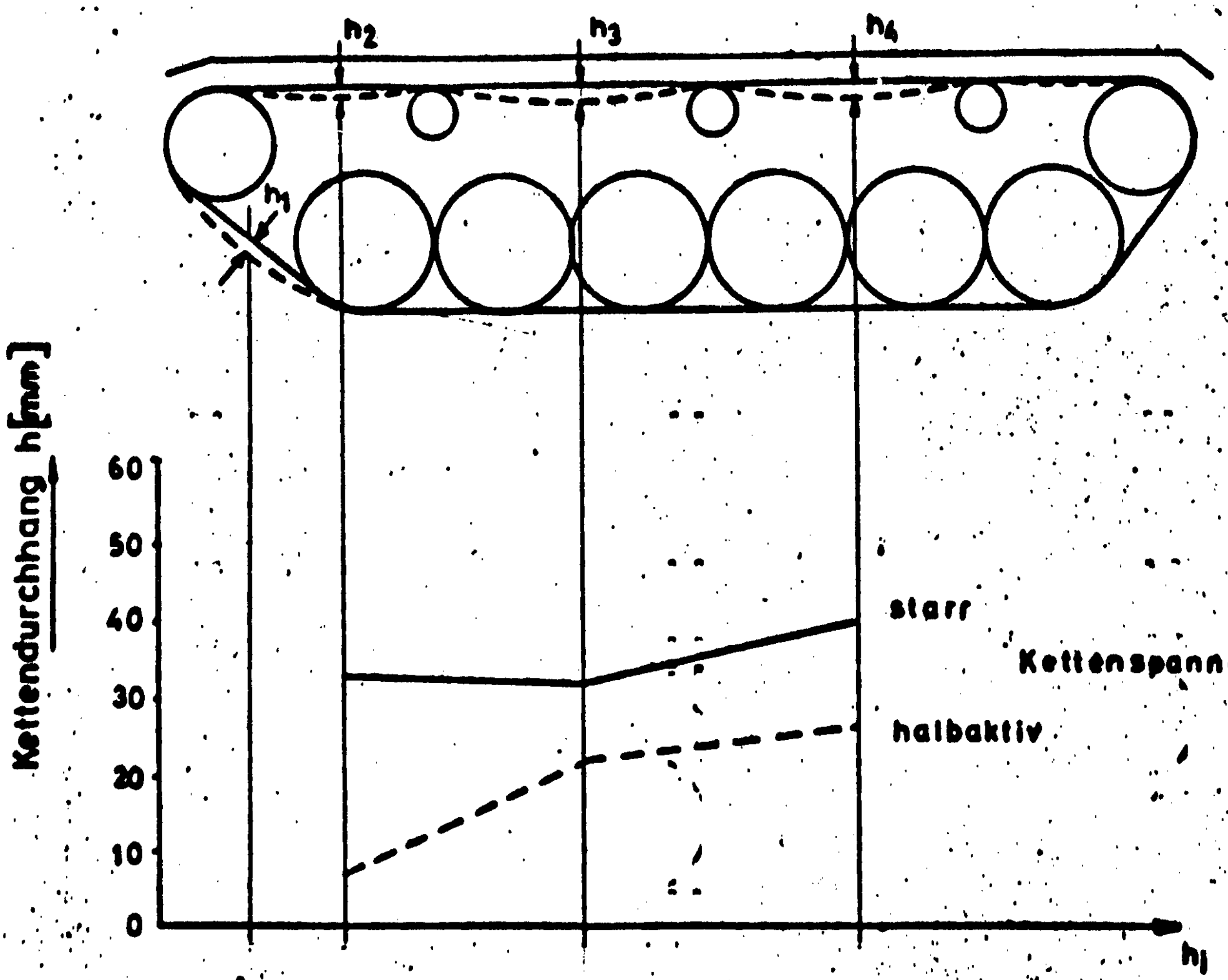


Abb. : Auffahrt auf ein Hindernis

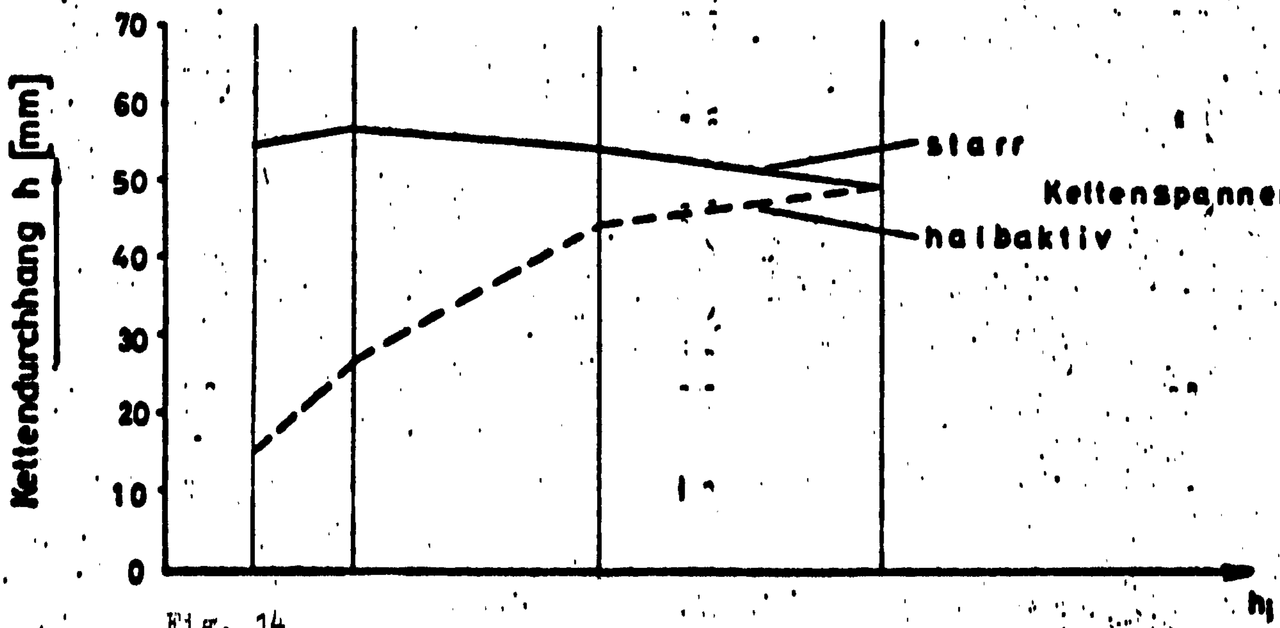
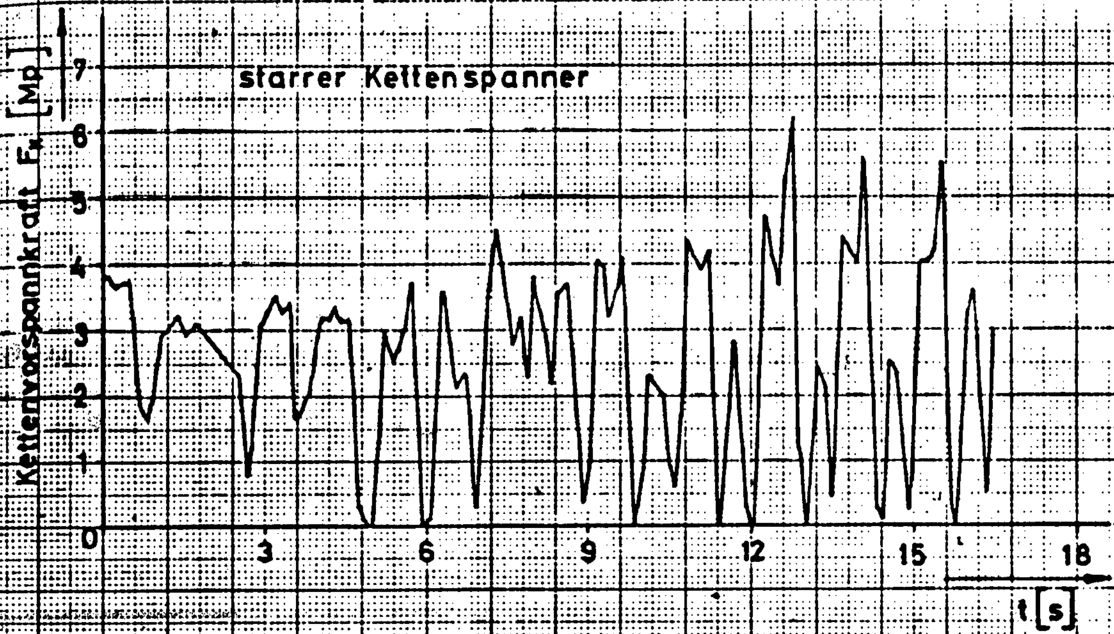


Fig. 14 : Einfedern nach Hindernisüberfahrt



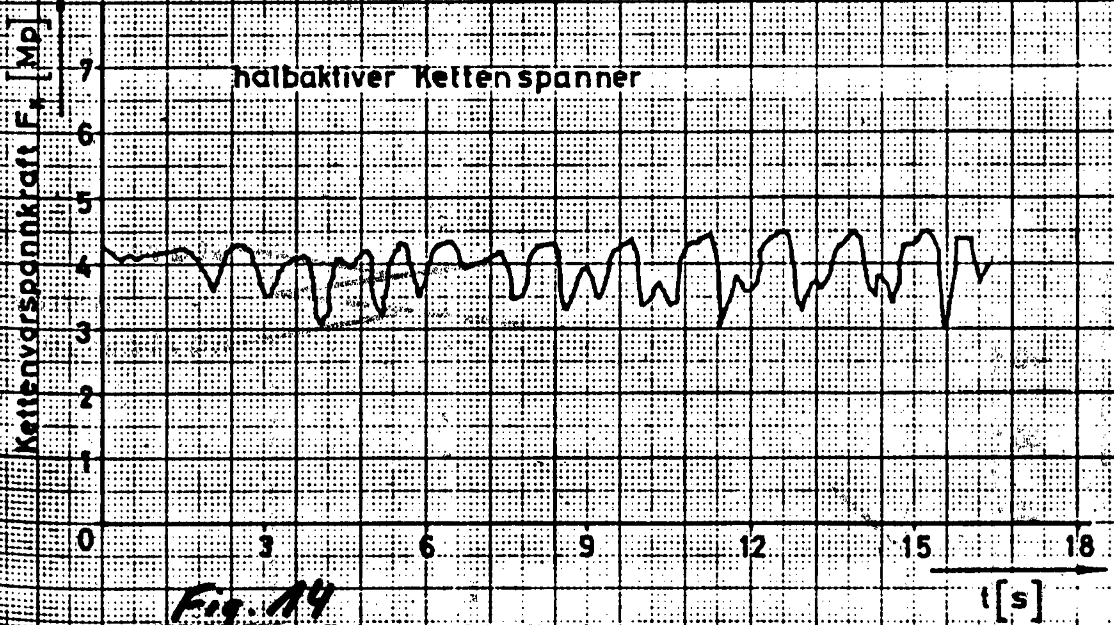


Fig. 14 : Fahrt auf Wellenbahn

V max ≈ 42 km/h

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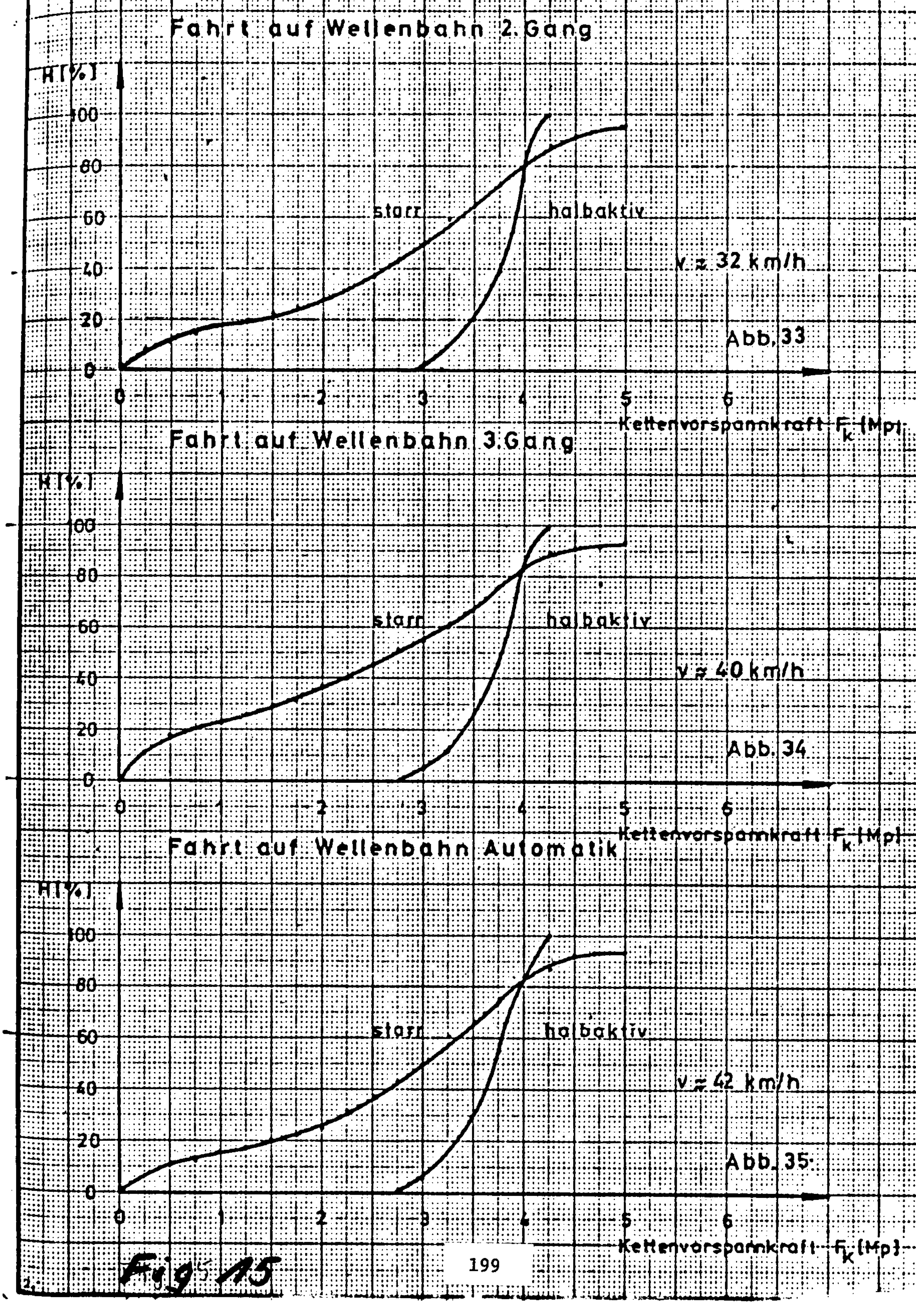


Fig. 15

IABG TV

Ber.-Nr.: 98-5/74
Blatt: 27

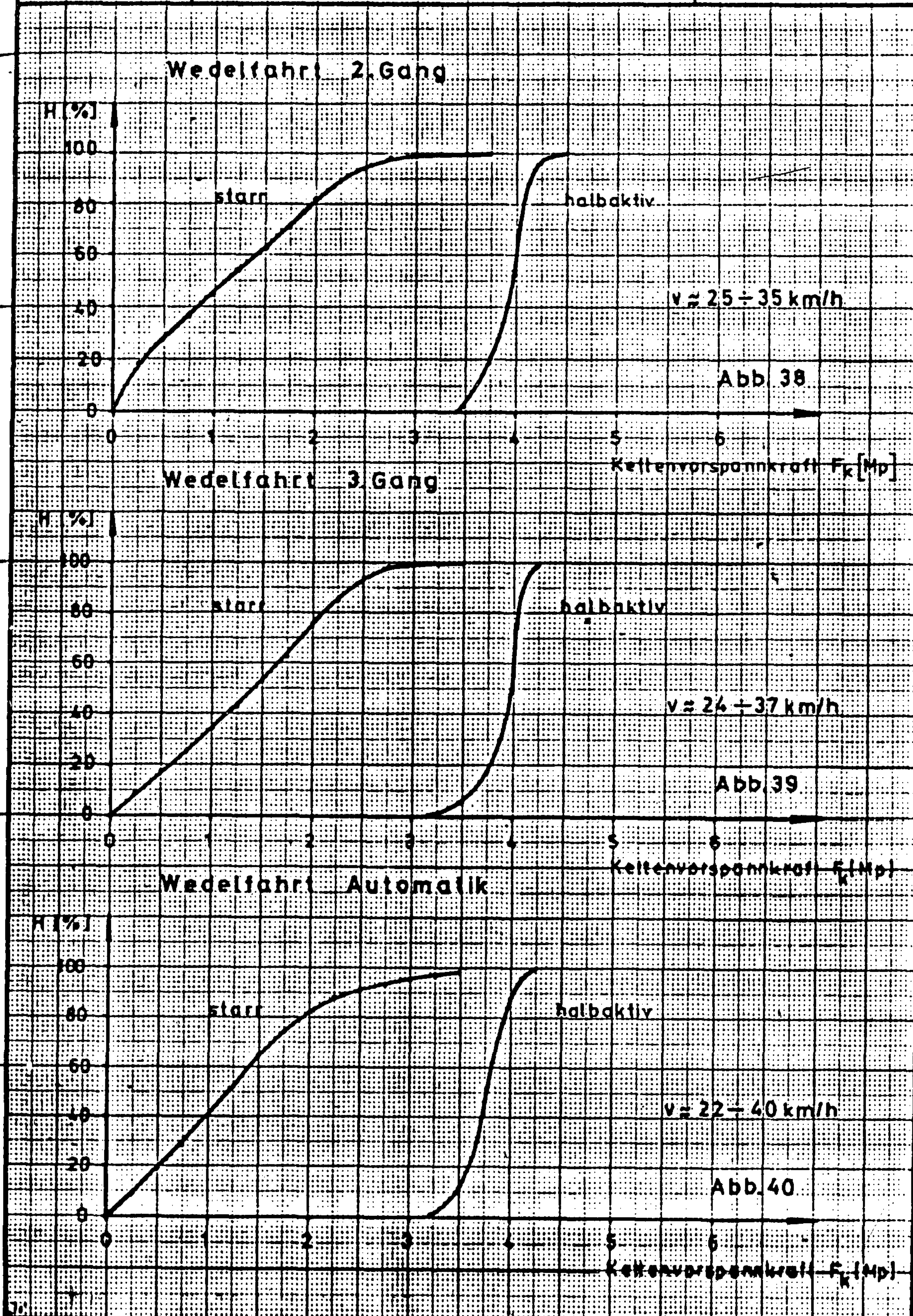


Fig. 16

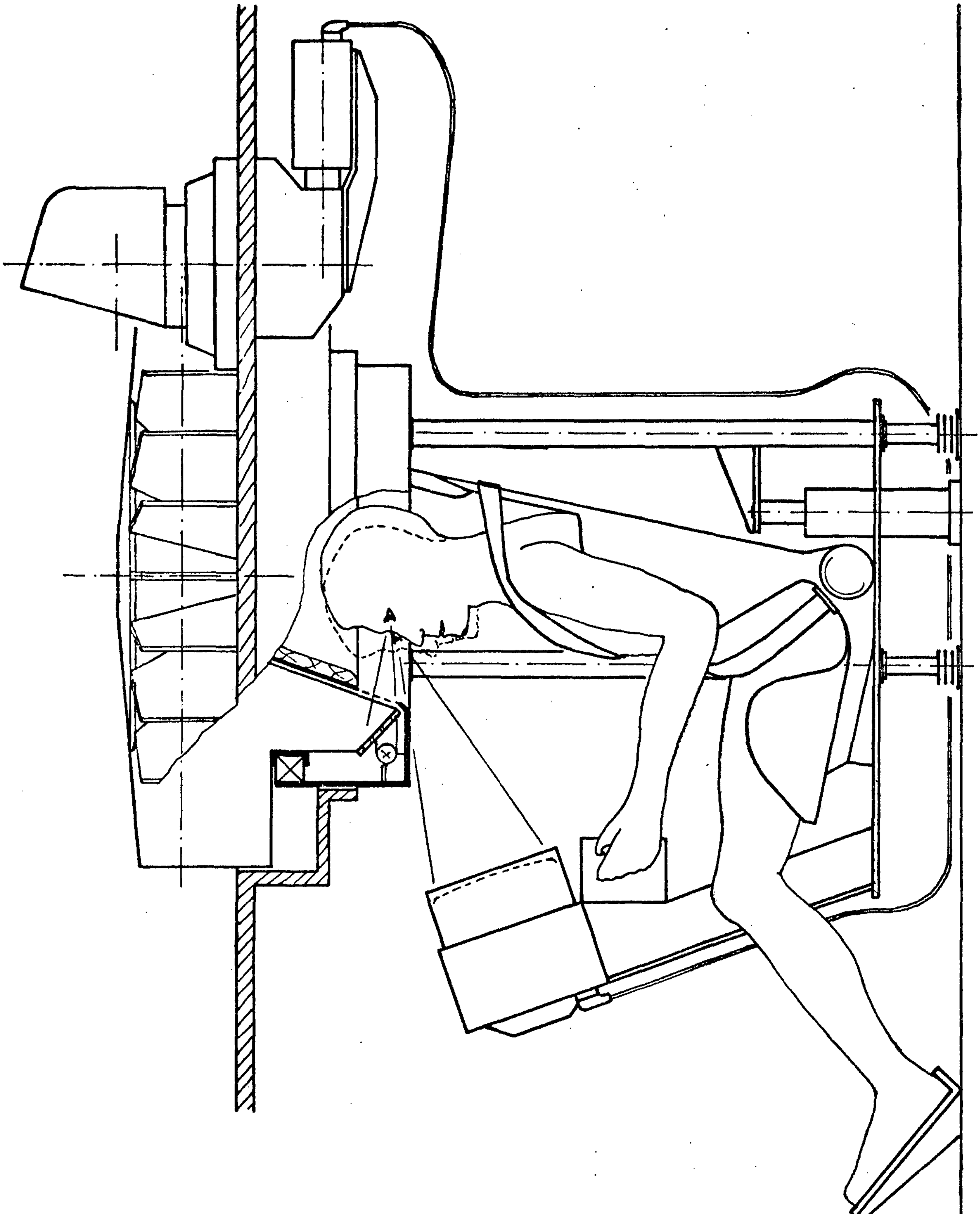
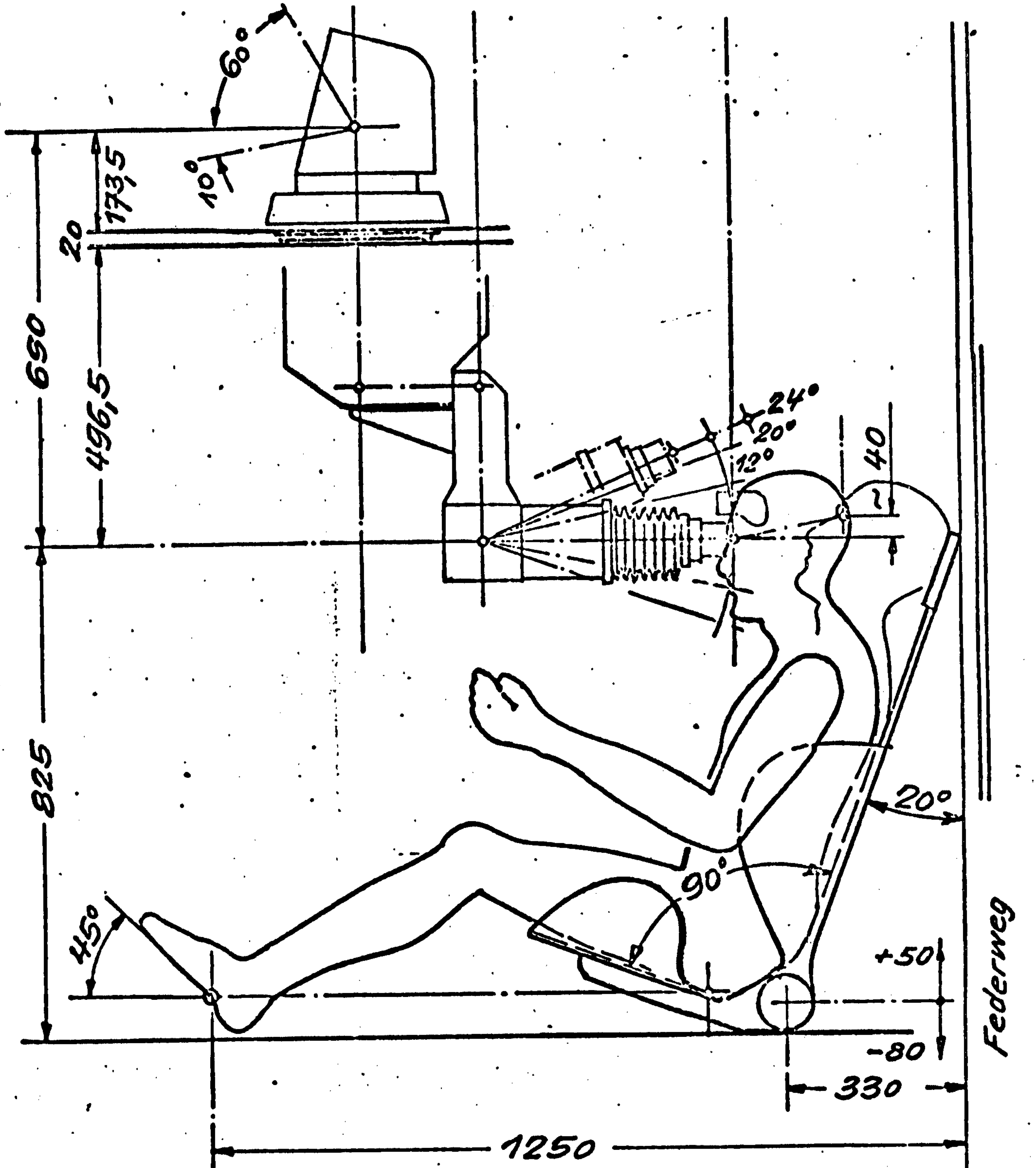


Fig. 18



Zuordnung PERI, Beobachter und Sitz

Fig. 19

SIMULATION OF USE BY TEST PLANTS

Presented By

W. RAASCH
FRG

Simulation of Use by Test Plants

1. General Aspects

Beside the simulation by computer also the simulation of vehicle use by test plants got more and more important.

Because of the great complication of modern track-laying-vehicles the testing in special test plants is a essential contribution to the development of these vehicles.

This is founded in the fact that

- the design ideas have to be checked before the mere development of a component starts

- performance datas have to be investigated

- new designed components have to be tested to prove their function

- the interaction of components and partial systems must be proved also for the operating conditions are extreme

- and that not at last the reliability of components and partial systems have to be proved in endurance tests

The test in laboratories realize these demands by the following advantages

- it is not necessary to have built the whole system for running a laboratory test

- the test itself can be as a simulation of the missions load as a synthetic test routine

- it is possible to simulate extreme situations which can not be realized with the real vehicle with out getting danger for men and machine

- the conditions for measurements are much better then at working on vehicle in the provingground

- test may be reproduced at any time later without lost of accuracy

- endurance tests can be done with much lower costs

For this IABG tries to run copmputer simulation as well as simulations of mission's loads in the laboratory connected one with the other to investigate the problems existing in the running gear till yet.

These laboratory tests shall be pointed out with the following few examples.

2. Investigation on Tracks

It is rather difficult to measure the different loads acting on a track while the vehicle is running. Therefor we have built a testplant with which we can analyze the effect of the several loads (figure 1). Each one of the decisive loads can act one the track element. As you can see the loads are set up by hydraulic actuators, for each direction one. The bearing plate can be moved by three adjustable struts. All loads are measured with electrical dynamometers.

Another task which can be hardly resolved at a running tank is to measure the frictional forces between wheel and tooth of track. We got this by the following laboratory simulation:

The track is fixed on a large drum (figure 2) which has a diameter of 4 meters. The teeth of the track are positioned outside in that way, that the wheel can roll on the track as it rolls at the vehicle. The drum is driven by a controlled d.c-motor so that different velocities can be realized. The wheel is fixed with a strut on a platform which is moveable in the vertical direction and the weight of which is that which acts on one wheel of the vehicle. With a hydraulic actuator the wheel can be pressed against the teeth of the track and the reaction force in the running direction is measured with a dynamometer situated in the horizontal strut.

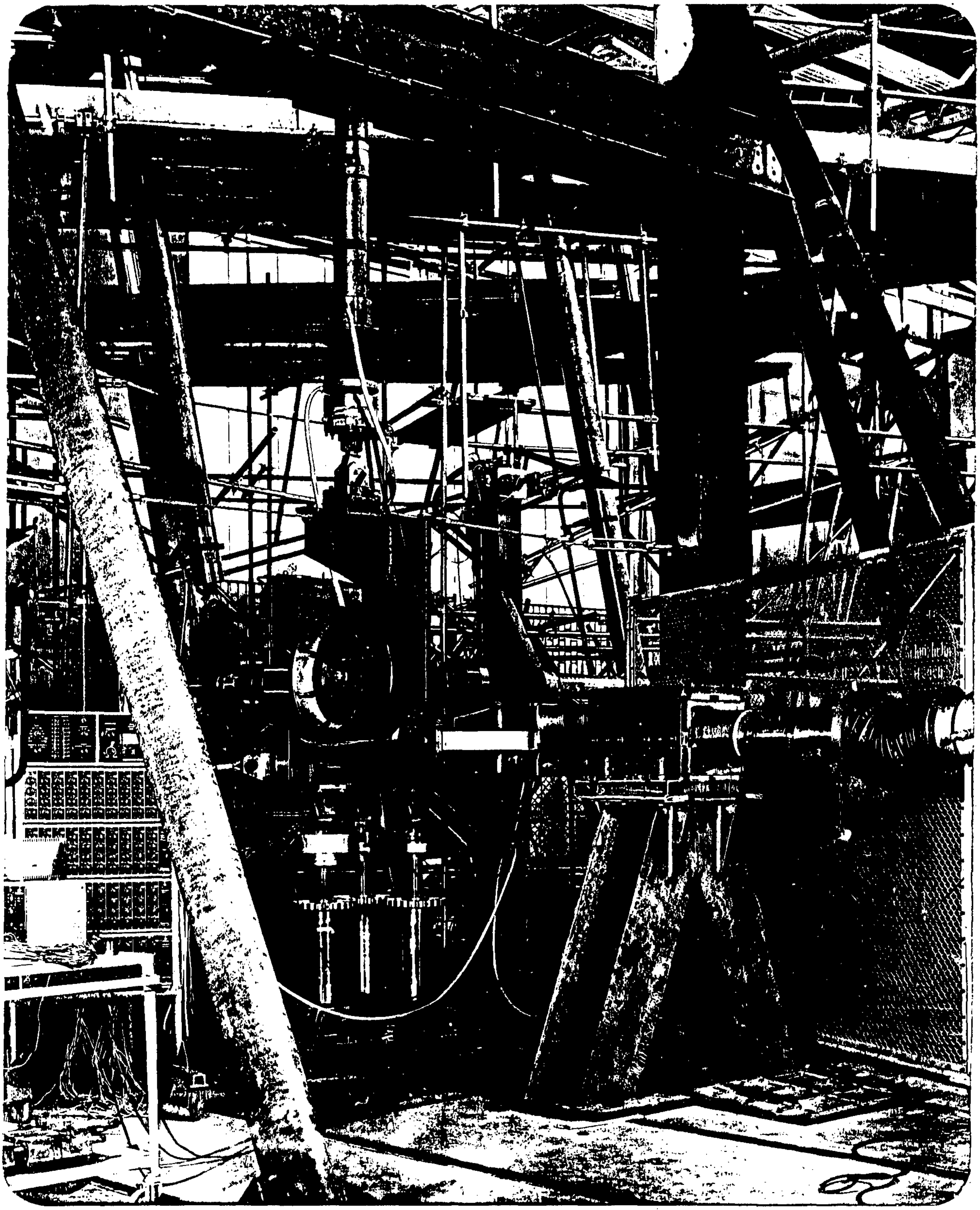
Figure 3 shows the sight of this testplant. It is also possible to put mud and water inside. It is also possible to investigate the vibrations which are generated by the track.

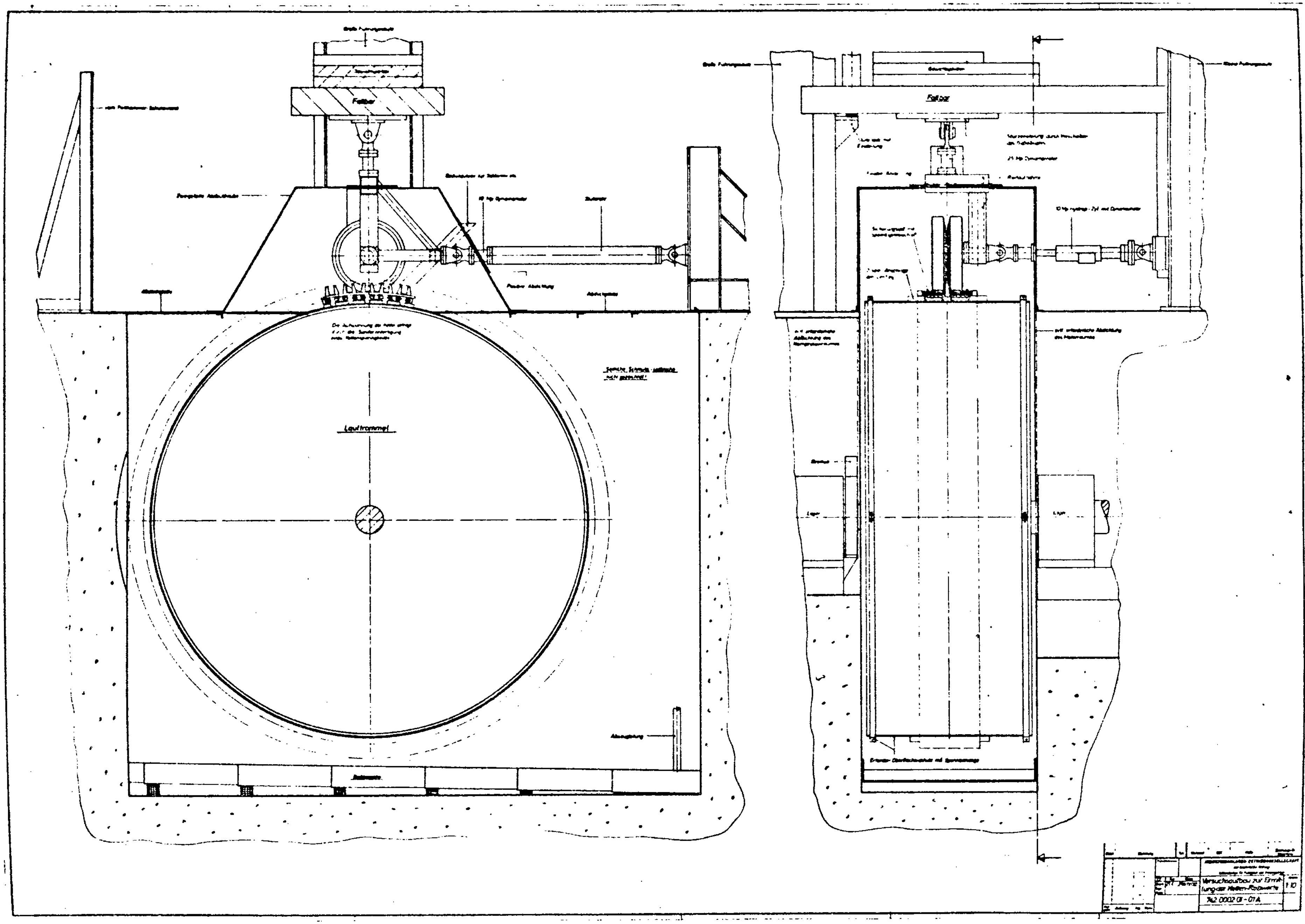
Problems which happens during the dynamic running of the vehicle on the track can be resolved in a lot of details. So it is possible to run test on fatigue-problems which result out of the action an reaction of all parts of the running gear.

Looking for this problems we have designed a kind of track-rig (figure 4).

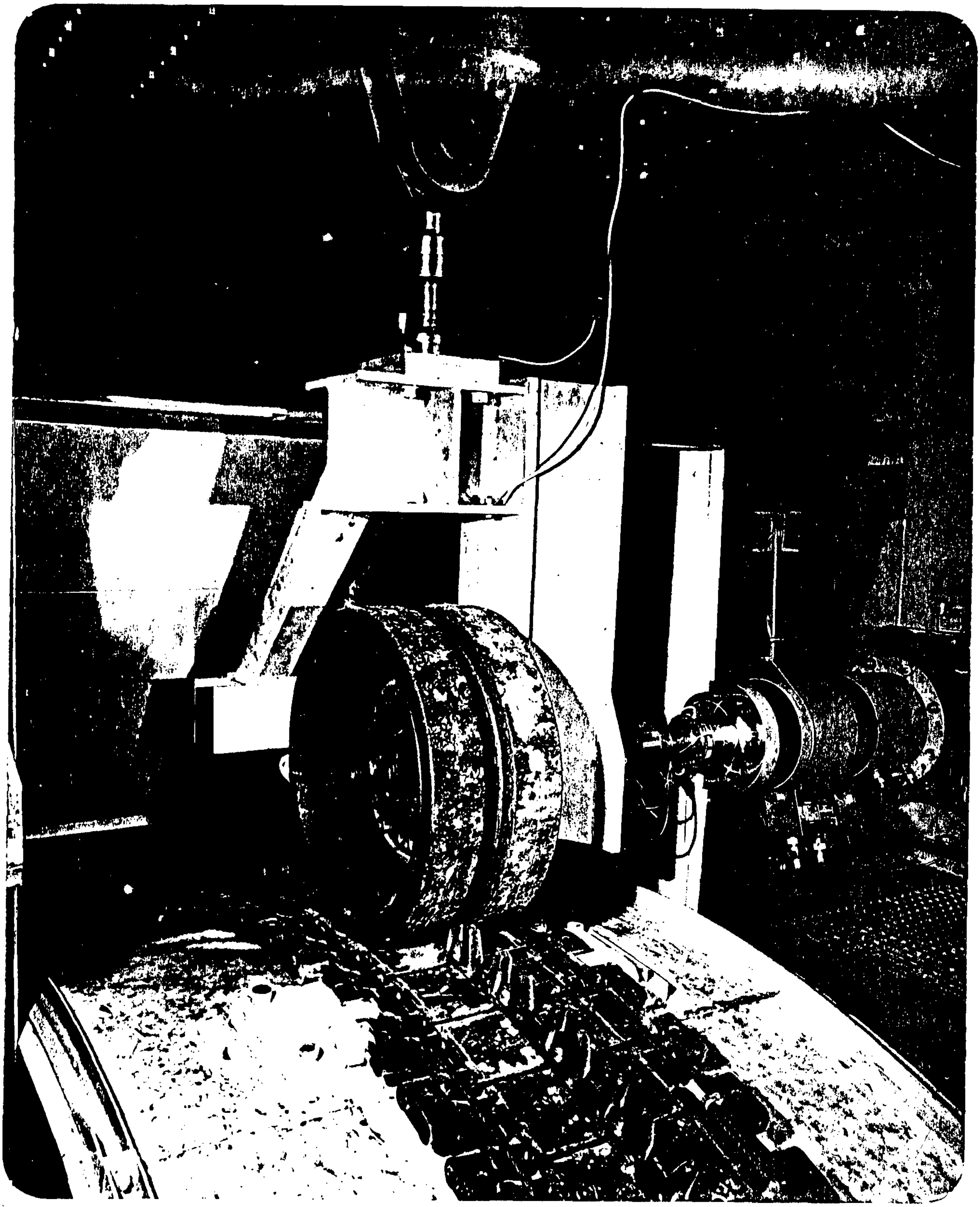
The track is running across the sprocket, the wheels, the teensioner, and the jackwheels. Instead of the first wheel there is another sprocket to span the track. The span-forces are generated by a chain drive where at a hydraulic actuator spans this chain and here by the vehicle's track. The other loads are produced to the track by moving the rollerpath under the track (figure 5).

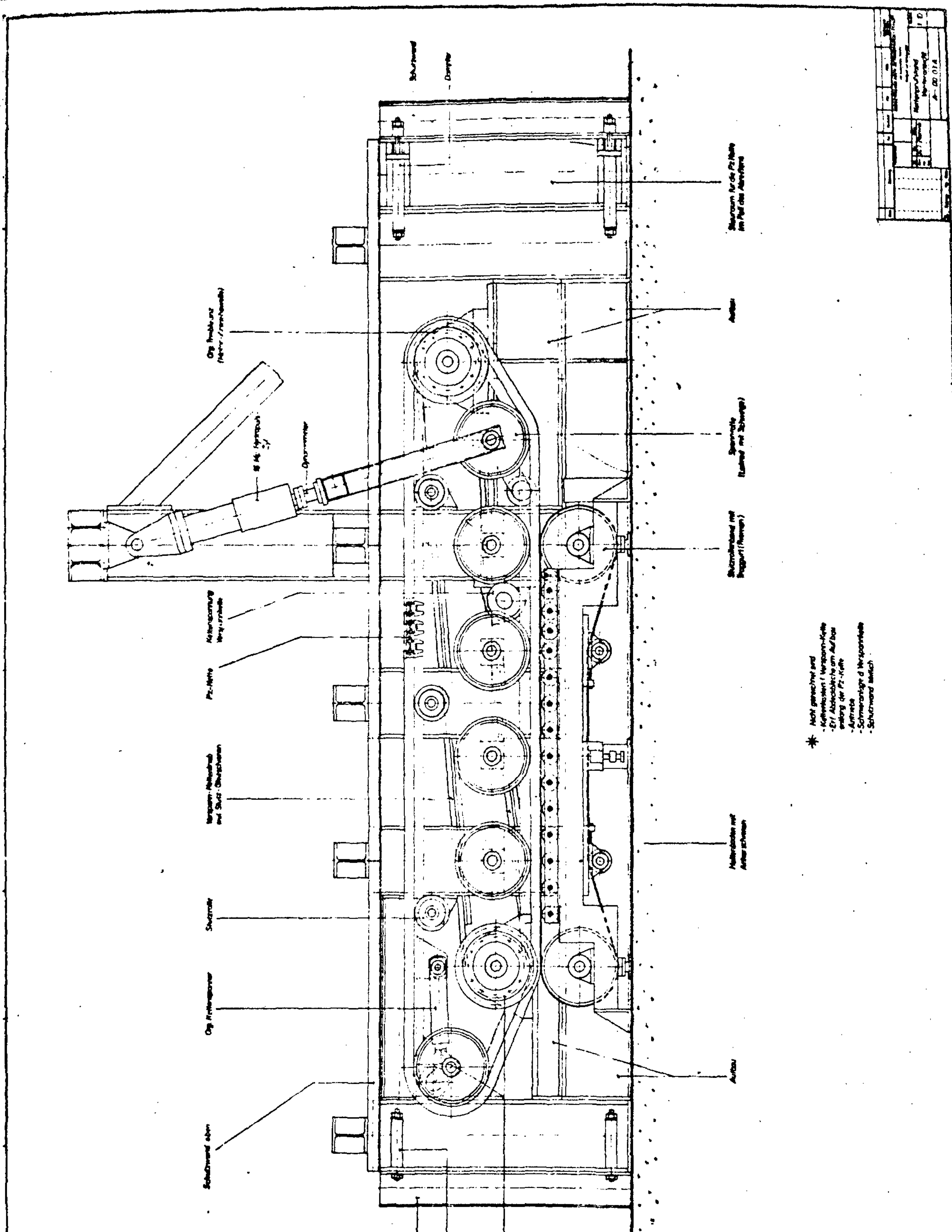
This test equipment shall be ready in the autumn of this year.



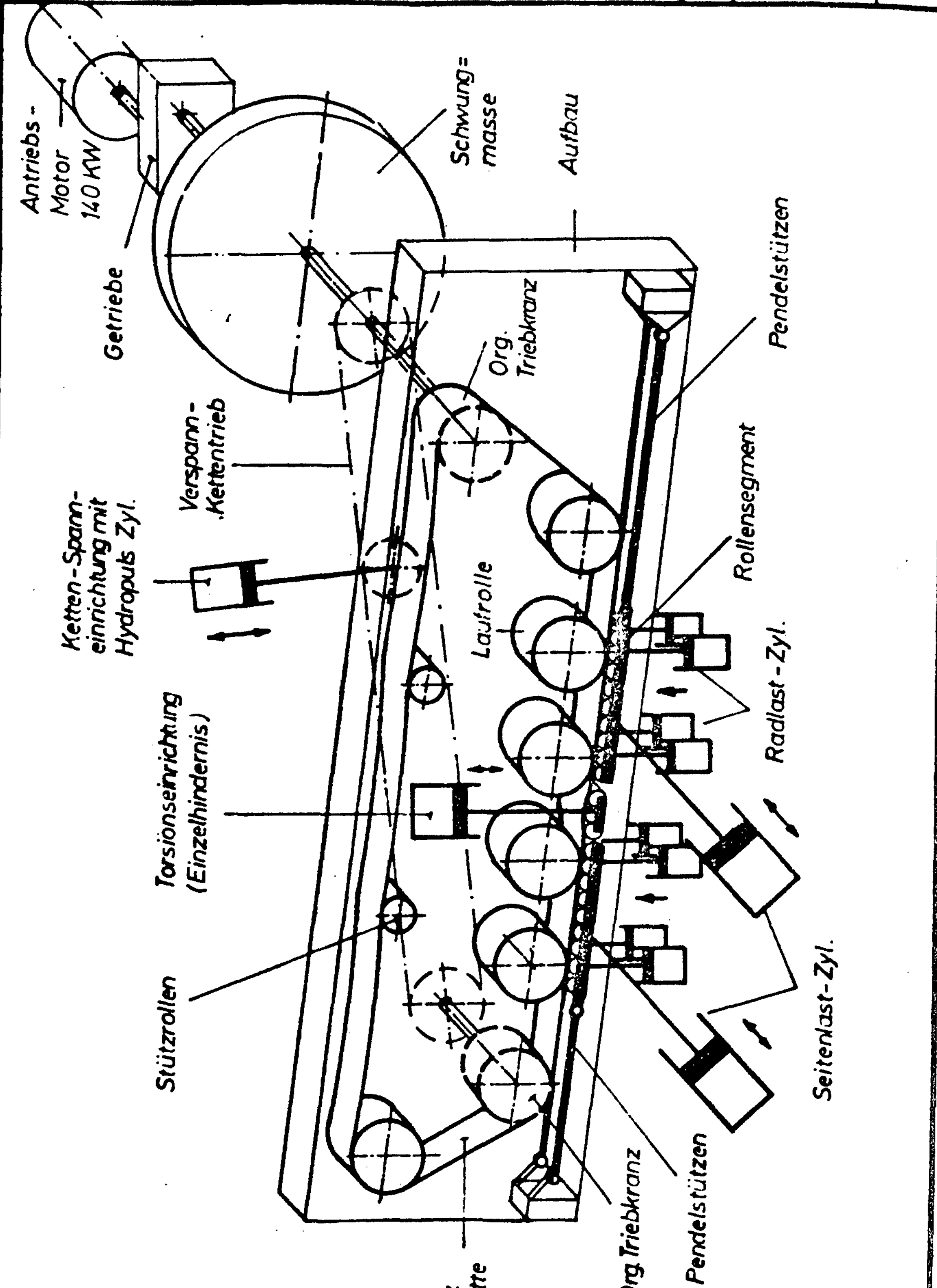


Fallbär
Lauftrommel
Lager
Versuchsaufbau zur Ermitt-
1:10
742 0002 01 - 01A





IABG Ottobrunn TFM	Prinzipskizze Gleiskettenprüfstand	Auftrag
		Nr.



20.92.1.1.

Bearbeiter Hammer	Datum 3.10.1974/na		

3. Investigations on Suspension and Stabilizer

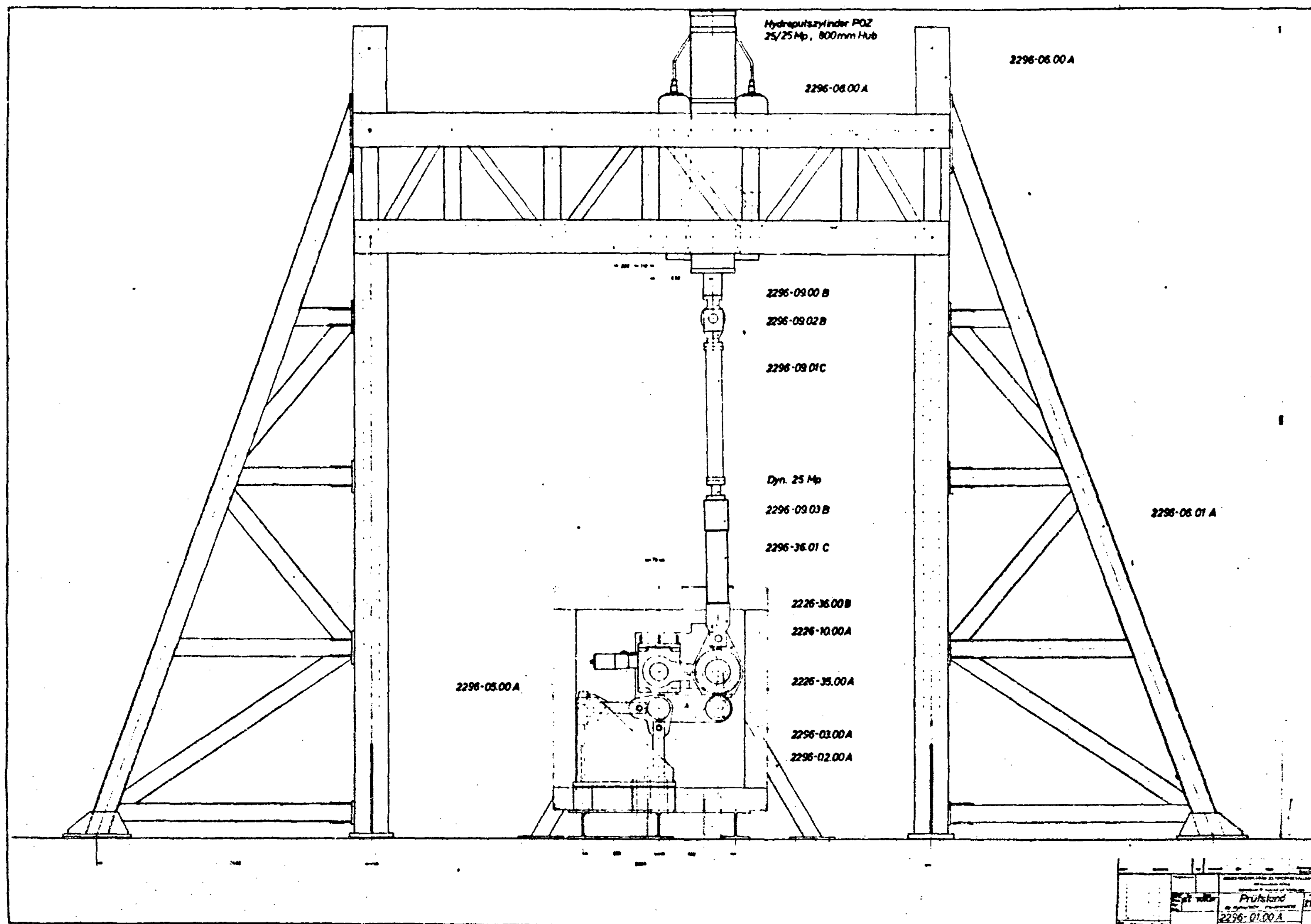
The most problems which exist at the running gear are those of suspension and stabilzer. The demands to the
as it was pointed out at increasing the terrain's
running gear rise with higher velocities roughness.
The main duty of the running gear is tc keep off the disturbance of the terrains roughness from the hull.

So you need a rather high performance of test plants when simulating the rating of the running gear in the laboratory (figure 6).

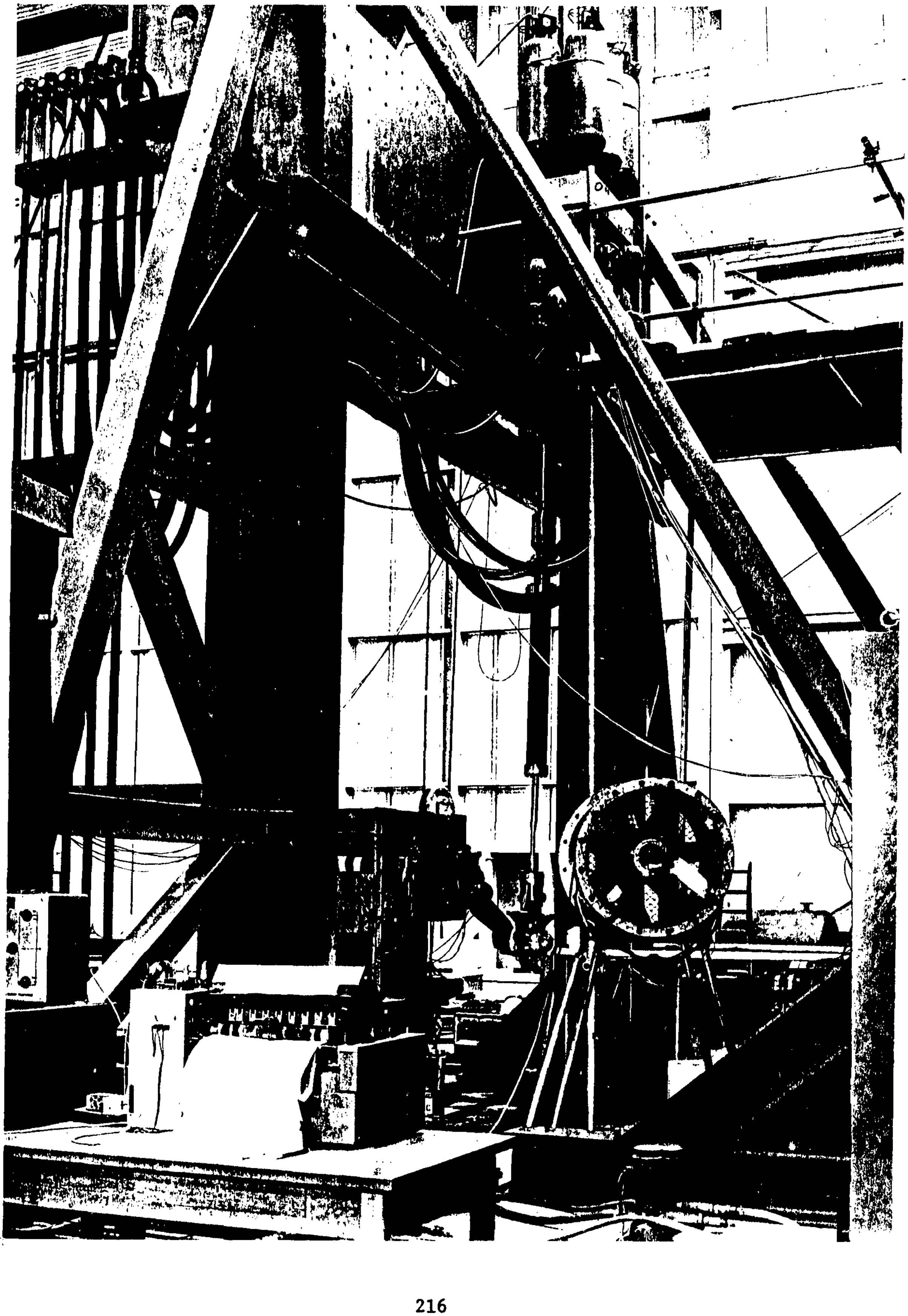
The center of our suspension test plant is a hydraulic actuator controled by a servo-valve. This actuator has a strike of 0.8 meter, a maximum load of 25.000 kp and a velocity of displacement of the piston of 10 m/s. The peak energy is restored in a hydraulic accumulator which is fixed directly to the actuator.

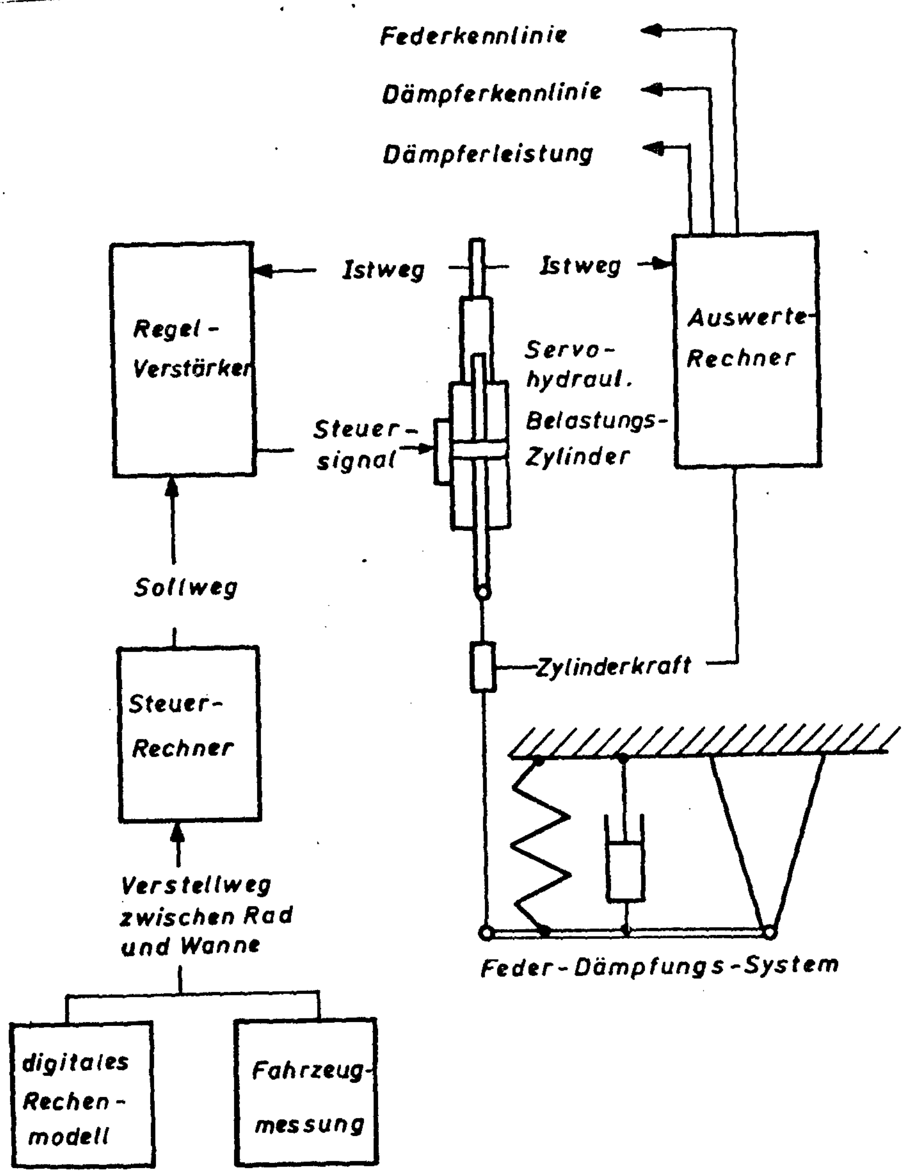
Figure 7 shows the view of the testplant with the actuator fixed at a loadframe and a hydropneumatic suspensionsystem on testing.
For realizing all demands to the testplant it is controled by a computer (figure 8). This computer can send random signals, as you get out of vehicle measurements, or synthetic signals as may be sine waves or rampfunctions to the controlamplifier of the servo-valve. It is also possible to compute control-datas out of measurements while running a test.
We can also change the test environment as you can see here, (fegure 9) where we run a test with deep an high temperatures.

Another possibility is to fix the test-suspensions system not rigid to the testframe but at a frame which is moveable in the vertical direction (figure 10) so that you have a simple on-mass-transient-analyzer of the vehicle.

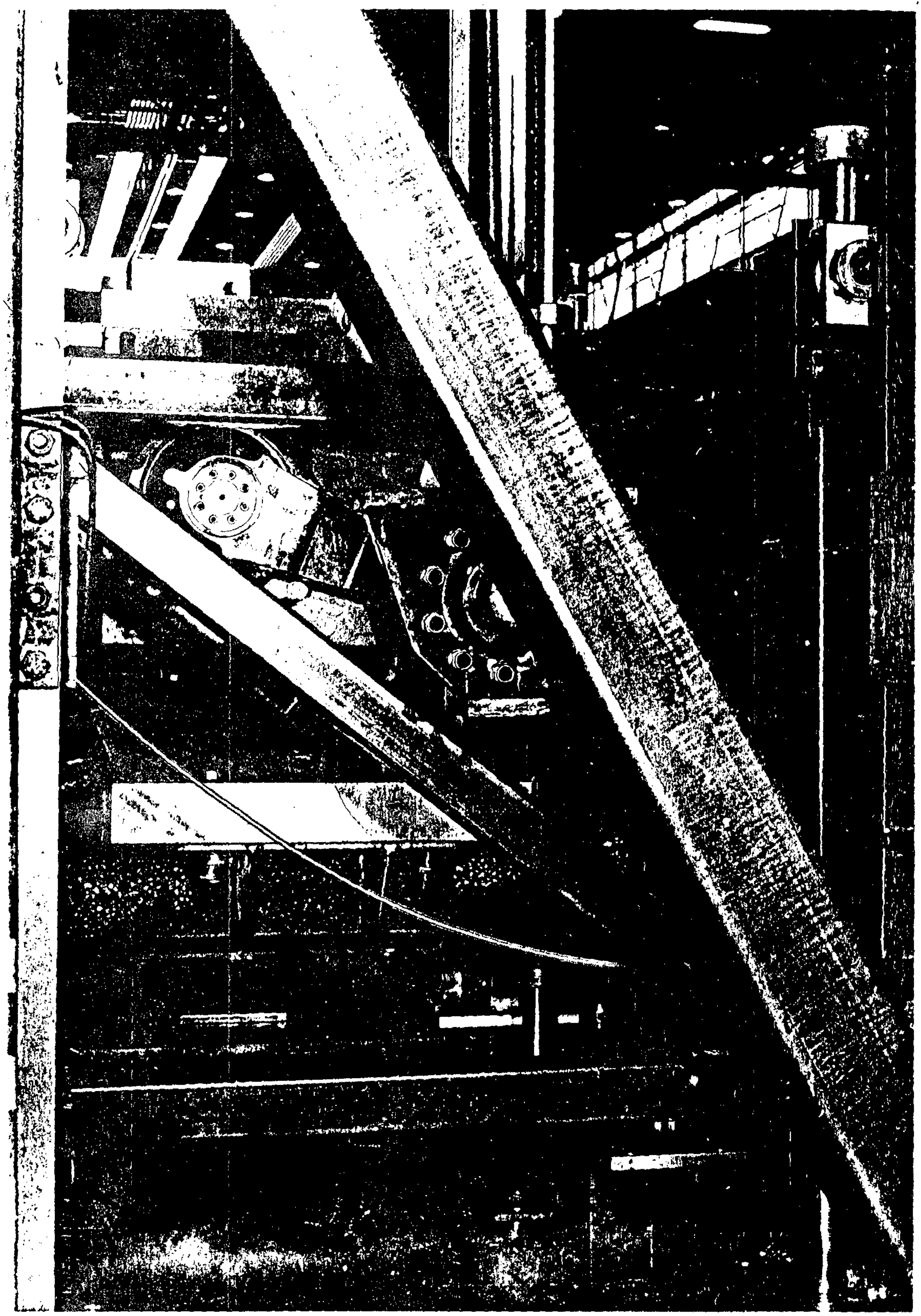


Hydrapulszylinder POZ
25/25 Mp, 800mm Hub
2296-06.00 A
2296-08.00 A
2296-09.00 B
2296-09.02 B
2296-09.01 C
Dyn. 25 Mp
2296-09.03 B
2296-36.01 C
2226-36.00 B
2226-10.00 A
2226-35.00 A
2296-03.00 A
2296-02.00 A
2296-05.00 A
2296-06.01 A
Prüfstand
2296-01.00 A
1:10





Schema der Steuerung des Schnellen Federprüfstandes

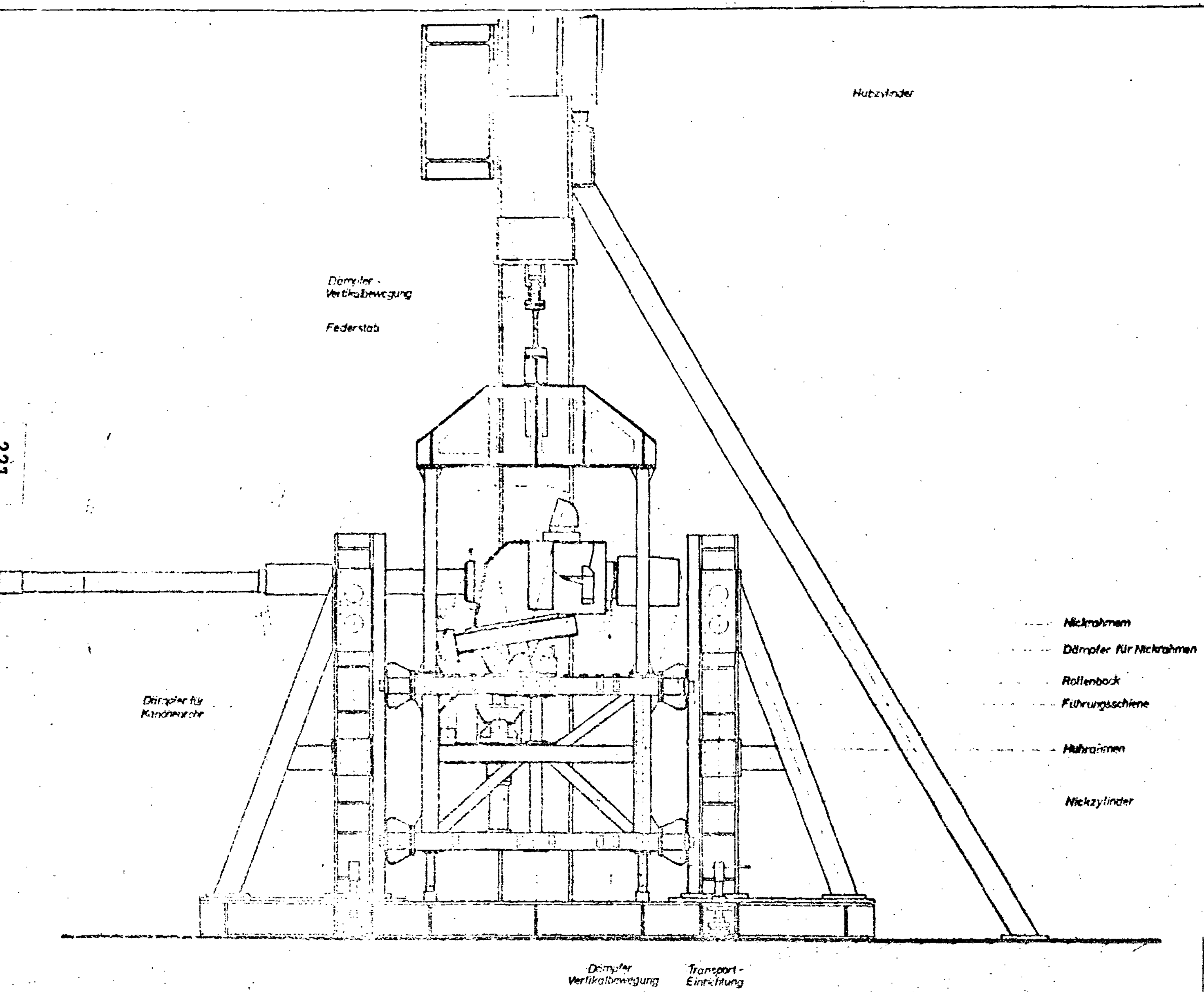


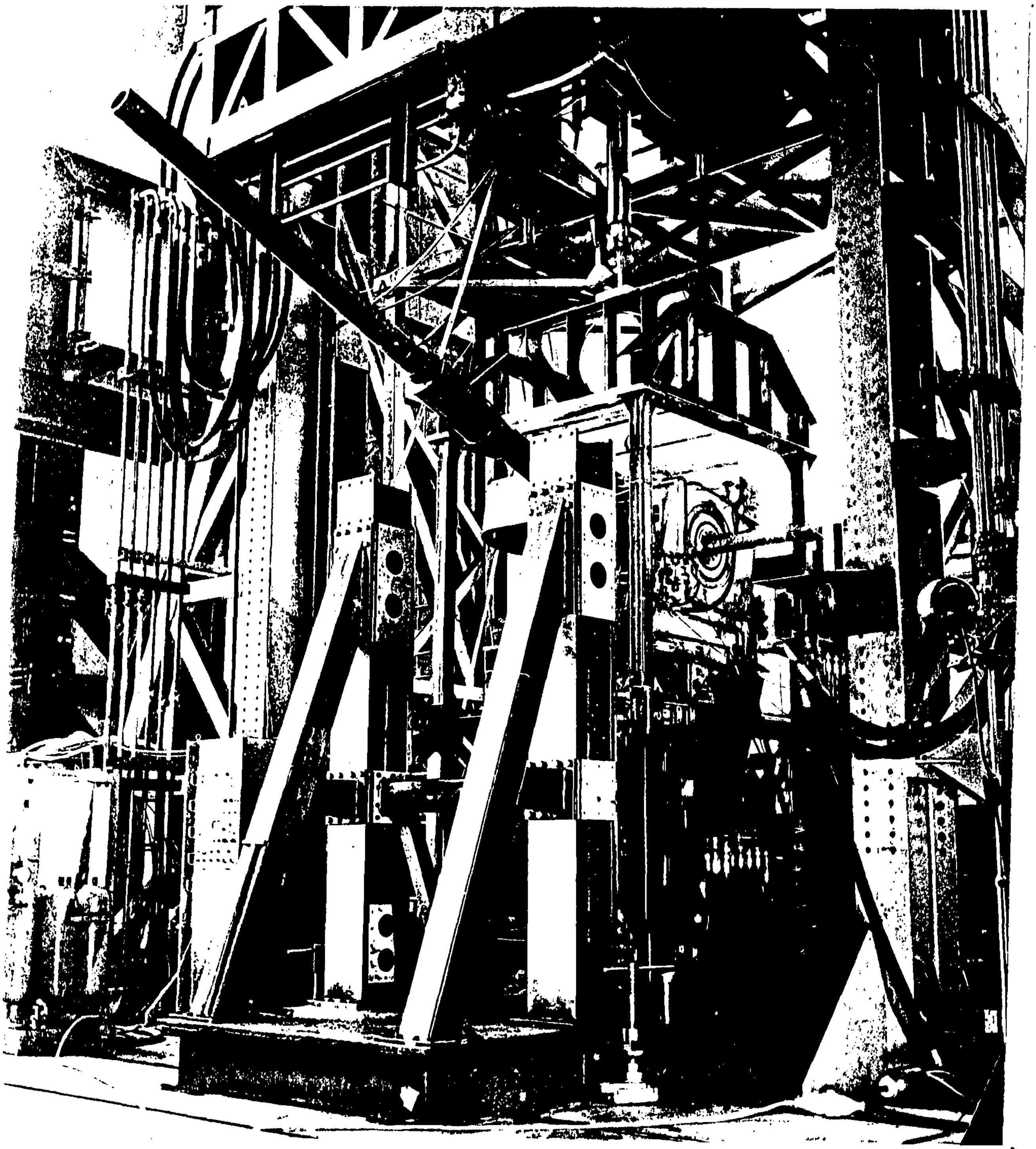
4. Simulation of the Movement of the Hull

One step further of the last discussed test facility is the facility to simulate the movement of the hull (figure 11).

This is necessary for instance to test weapons stabilization.

We have run such a test on a stabilization with one-rotating-degree of freedom, it means pitching and also the vertical displacement, for it acts disturbing to the weapons bearing. With this test facility we can also run test random controlled with control-signals measured out of vehicles or synthetic control which means sine-waves or ramp-functions. Figure 12 shows a view of the test plant with the weapons stabilisation of VT-1 on test.


Hubzylinder
Dämpfer - Vertikalbewegung
Federstab
Dämpfer für
Nickrahmen
Dämpfer für Nickrahmen
Rollenbock
Führungsschiene
Hubrahmen
Nickzylinder
Dämpfer Vertikalbewegung
Transport - Einrichtung



5. Final Remarks

It was the duty of this shortreview to point out in which way we are working at the simulation of vehicle's loads during missions, also if the problems are more complicate. I think the way to look for the several problems which come into existence during development by the matter of laboratory test will become more and more important in the future.

MOBILITY TESTS CONDUCTED WITH TEST RIGS

Presented By

D. HAUG
FRG

Ministry of Defence
Ru VII 6 - Az.: 90-25-50

Bonn, 1 April 1975
Ext. 4135

Subject: US/German Mobility Symposium to be Held from 8 to 11 April 1975

- Mobility Tests Conducted with Test Rigs -

Gentlemen,

Mobility (Fig. 1) is one of the four main elements that make up the combat value of a main battle tank. Fig. 2 shows the specific engine performance called for in order to achieve the level of mobility required by MBTs in the years 1930 to 1980. This figure reveals the unmistakable trend towards an increase in specific engine performance. OR studies have shown that, when sticking to the current operational tactics, no considerable increase in combat value can be achieved by this measure alone. Therefore, one of the objectives of our study, with which we in the Federal Republic of Germany have been dealing since early in the sixties, is to clarify the following problems: To what extent can the mobility of a land-bound tracked vehicle be increased, viewed from a technical aspect, and how can that mobility be optimally exploited on the battlefield with a view to significantly increasing the MBT's striking power when it has to engage enemy armor or antitank weapons?

Analytical mathematical studies have revealed that vehicles moving on straightline courses can be hit in open terrain with high probability by means of future fire control aids, such as trackers, lead computers etc., even when they are driving at high speed. These studies have also shown that a vehicle is apt to attain a high survivability when it moves more or less in the enemy direction on a slalom-type course (Fig. 3) with high lateral acceleration pursuing e.g. 60° curves. Sign and value of the target's angular velocity are bound to change rapidly and render it difficult for the enemy gunner to track his target and select the proper lead. When using gun-tube weapons, this will apply up to combat ranges of 1 second projectile flight time. Figure 4 shows hit probability as a function of target range. The lateral acceleration formed by the quotient of the squared vehicle speed and the mean curve radius has been plotted as a parameter. At a target range of 1,500 m, e.g. the following hit probabilities will result

$3 m/sec^2$	33%
$7 m/sec^2$	8%

Figure 5 shows hit probability curves for a conventional, a possible MBT 8o with about 4oHPpt and a highagile future tank with about 6oHPpt.

You will note that the lateral acceleration exerts a considerable influence on the hit probability. The influence exercised by speed is of secondary nature only.

These theoretical findings have been verified both with gun-tube weapons by simulation on the target gun camera range and by actual firings carried out with guided missiles of the first and second generation. Further tests have been planned.

- controllability;
- battlefield operations;
- feasibility of a highly mobile fighting vehicle with the high firepower of semi-fixed twin guns and effective front armor combined with optimum protection of the other vehicle surfaces.

These considerations and studies led to partly unconventional new solutions.

I would now like to present the various test rigs, with which the aforementioned studies and field tests were conducted or are planned to be conducted (Fig. 6). We make a difference between first and second generation vehicles. The five vehicles of the first generation are based on chassis of the US reconnaissance tank M 41, whereas the ten vehicles of the second generation are based equally on MBT 70 prototype and MBT LEOPARD 1 chassis. At present, we have 15 test rigs at our disposal. I would now like to give you a brief description of these vehicles:

1. The <u>LVT</u> (laser test rig) is used for proving the functionability of the coincidence firing method with laser and a semi-fixed weapon system.

2. The FVT (suspension and chassis test rig) is used for testing the first model of the torsion bar.

Having briefed you in telegram style on our fundamental considerations regarding the increase in mobility and the resulting effects on the vehicle's combat value, I would now like to pass on to the mobility tests carried out with test rigs.

As I have already mentioned, the theoretical studies revealed that effective protection by mobility may be achieved when the vehicle follows a slalom-type course at speeds ranging from 10 to 15 m/sec and at lateral accelerations ranging from 5 to 7 m/sec^2.

Tests made with the MBT LEOPARD 1 have shown that, with the LEOPARD power to weight ratio (approx. 20 hp/ton) a speed of only 7 to 8 m/sec at a lateral acceleration of 2.0 to 2.5 m/sec^2 can be attained; i.e. we were unable to conduct our tests with the equipment available and were thus forced to produce suitable test rigs. Since it was our objective to prove the feasibility of a fully functional, highly agile fighting vehicle, we could not restrict ourselves to mobility based on the power to weight ratio or suspension but had to extend our studies and test to other components and vehicle functions sensitive to mobility effects, such as

- the operability of the vehicle by its crew;
- quick-reacting and accurate firing, both statically and on the move;

energy storage suspension system and is at present being used for testing the coincidence firing method and the visibility concept.

3. The first mobility tests and guided missile firing trials as well as the smoke screening tests were conducted with the <u>RVT-1</u> (rocket-equipped test rig).

4. The mobility tests begun with the <u>RVT-1</u> are being continued with the <u>RVT-2</u> (missile test rig Nr. 2). This vehicle has a 1,800 hp power pack. I shall deal with the ride-dynamical tests made with this vehicle in more detail in the course of my presentation.

   ARPA has requested to use this vehicle on a loan basis. This test rig will probably be available in the United States for demonstration purposes in the 2nd half of this year.

   To continue the ride-dynamical tests within the engine performance range up to 2,300 hp, another test rig built on the basis of a modified MBT LEOPARD 1 hull - the so-called mobility test rig - is in the planning stage.

   The RVT-2 will then be more or less exclusively available for further firing trials.

5. A combined gas turbine/Diesel engine power plant was envisaged for the KVT (combined power plant test rig) or HVT (auxiliary engine test rig). At present, a liquid air combustion engine operating independently of outside air is installed in this test rig.

6. The highly mobile vehicle concept featuring two semi-fixed 105 mm weapon systems has been realized in the VT 1-1 (test rig 1-1).

7. The ET 703 is used in its initial stage for carrying out agility tests and has been converted to the VT 1-2 (test rig 1-2) and has been equipped with two 120 mm guns (smooth bore) and with an automatic loading device.

8. The first coincidence firing tests were conducted with the ET 706 with its turret locked.

9. The ET 707 has been equipped with the Hydropneumatic energy storage suspension system. Comparative mobility and suspension tests were made with this vehicle.

10. The torsion bar energy storage suspension system was installed in the ET 708. This vehicle is also used for conducting comparative mobility tests.

In my previous presentation, I already reported on the results obtained with the suspensions of the ET 703, 707, and 708.

11. The concept of the five vehicles GVT 01 to GVT 05 is very similar to that of the VT 1-1 and VT 1-2. These vehicles are used for conducting combat trials with the objective of exercising operational tactics and testing the controllability of highly mobile fighting vehicles on the battlefield.

After giving you a survey of the test rigs, I would now like to give you more details of the mobility tests made with the RVT-2. In view of the fact that neither reliable theoretical nor practical findings were available, we were forced to enter unfamiliar ground. We were interested in clarifying the following two main issues:

a. Up to what limit can lateral forces be supported by the tracks on various earth formations, i.e., what maximum lateral acceleration can be achieved.

b. The performance required at the sprockets in various vehicle configurations.

The modified hull of a US reconnaissance tank M41, equipped with a 1,800 hp power pack, was used as a basic vehicle.

To test (Figure 7) the influence exerted by the steering ratio on the performance requirements at a constant ground pressure of 0.8 kg/cm^2, the vehicle may be supported on 4, 5 or 6 roadwheels on each side. This would produce the following three steering ratios: 0.93; 1.24 and 1.55.

The technical data of these three vehicle versions are shown in Figure 8. At an engine performance of 1,800 hp and a ground pressure of 0.8 kg/cm^2, specific outputs of 87.4 hp/ton, 65.5 hp/ton and 52.4 hp/ton will be achieved. No changes were made to the suspension components, such as the tracks, roadwheels, track adjusters and suspension. Merely the shock absorbers mounted to the last roadwheel pair were replaced by the more efficient hydraulic shock absorbers used on the MICV MARDER.

Due to the obsolete suspension geometry (160 mm positive) spring deflection with 250 mm overall spring deflection) and a few obsolete components, the test results obtained with this vehicle cannot be applied directly to tracked vehicles at present in the planning stage or under construction.

By variable configuration of the vehicle and its various components, the following parameters can be varied in the test series:

a. Three different steering ratios 0.93; 1.24; 1.55.

b. Engine performance infinitely variable up to 1,800 hp.

c. Two different final vehicle speeds 59 km/h (37 mph) and 69 km/h (43 mph).

d. Five different fixed radii in the mechanical split-power gear train.

e. Variation of the static track tension.

f. Specific ground pressure of 0.75 to 1.15 kg/cm^2.

g. Soil properties: loamy and sandy soil of various moisture degrees and vegetation types, compacted ground (concrete, composite paving).

Under variation of the various parameters, the trials program (Figure 9) comprises the following individual tests:

- coasting tests
- straightline runs at constant speed
- straightline acceleration runs
- pivot turning
- arc-type runs
- slalom-type runs
- applied load factor during simulated combat operations.

To acquire the measuring data during the various tests, a comprehensive EDP system was established, comprising the following main measuring points:

- torque and number of revolutions achieved by the main transmission and the final drives
- distance speeds and accelerations achieved in the longitudinal and lateral directions
- roll, pitch and yaw angular velocities

Data (Figure 10) was transmitted by telemetry from the test rig to the mobile receiving station.

The aforementioned tests produced comprehensive data material which is at present being evaluated. The complete test results can therefore not be presented at the present moment. However, the following statements can be made on the basis of the evaluated results now available and on the basis of the experience made during the tests:

- Slalom-type runs with lateral accelerations ranging from 6 to 7 m/sec^2 can be performed both on cohesive sandy soils and on loamy soils.
- To this end, specific sprocket performances ranging from 40 to 45 hp/ton at vehicle speeds up to 40 km/h (25 mph) are called for. Herr Merklinghaus will give you detailed information on this in his presentation.

- In comparison with the original M 41, the test rig withstood the test runs during which more than 1000 km (600 miles) were covered, under relatively excessive loading without sustaining considerable damage.
- The test drivers operating the vehicles from a sprung truck seat equipped with safety belts, withstood these runs without sustaining bodily harm.
- Straight line acceleration capability-highly agile tracked vehicles can be increased considerably also off road courses.

With the film to be shown after my presentation, I would like to give you an optical impression of the riding qualities of the RVT-2 test rig.

At the beginning of my presentation on the tests made with the RVT-2, I mentioned that the test results served to establish general trends and procedures and cannot be directly applied to tracked vehicles in the planning stage or at present under construction. The bulk of the data established with the RVT-2 is maximum values that can be considerably improved in vehicles built with more advanced components. Here are some of the possible improvements:

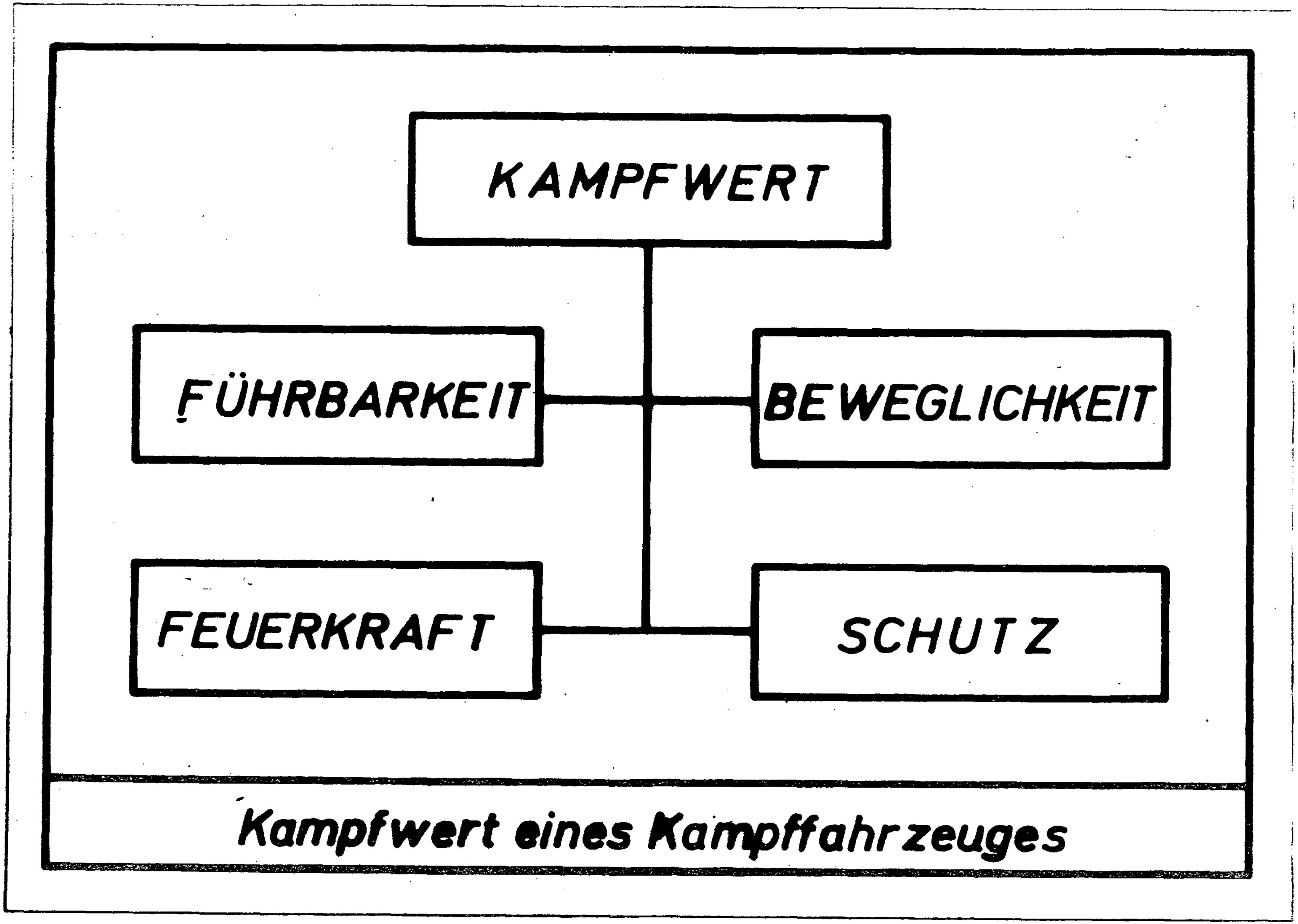
- Suspension with increased spring deflection, improved absorption of vibrations and lower losses of performance by using the energy storage suspension system.

- Reduction of the performance losses caused when driving in curves by reducing the lateral friction between the track center guides and the roadwheel leading flanks.
- Reduction of the static track tension and improvement of the dynamic track tension by using semi-active track adjusters.
- Optimization of the tracks with a view to transmitting high lateral guiding forces on the ground and lower losses of performance when driving in curves.
- Optimization of the vehicle steering gear to enable infinitely variable small radii so that the minimum curve radius possible required to achieve the maximum lateral acceleration possible can be attained at any vehicle speed.

And now allow me, please, at the end of my presentation, to draw your attention to a final point.

Depending on the tactical situation, the time required for highly mobile tactical maneuvers on the battlefield is estimated to reach a maximum of 5% of the entire mileage covered by a future fighting vehicle so that the high specific driving power called for should be merely seen as a short-term peak performance to be produced by the power pack and the other related components. During the remaining time, the vehicle will be operating under a considerably lower load, which will have a favorable effect on its durability.

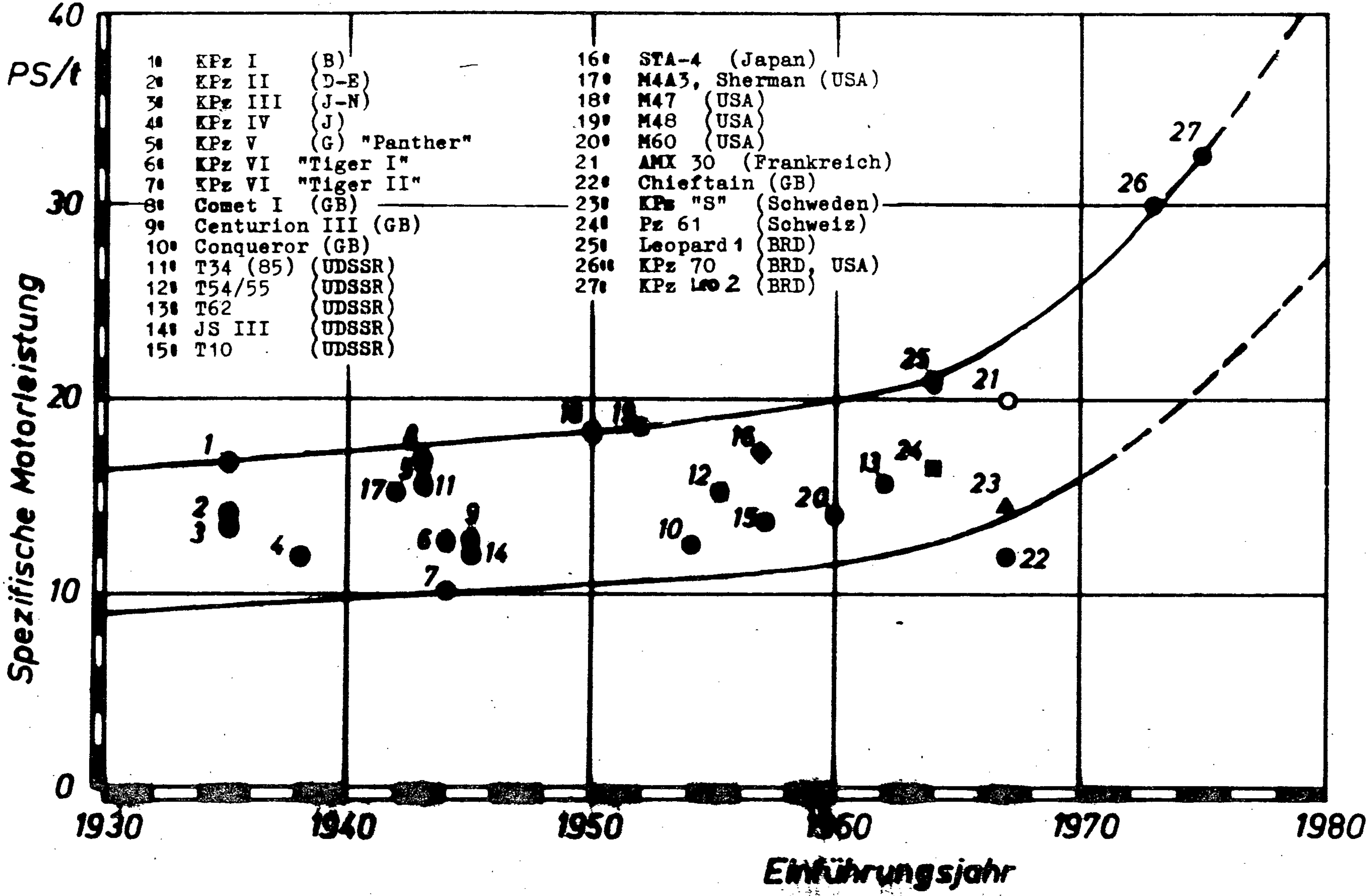
I thank you for your kind attention.



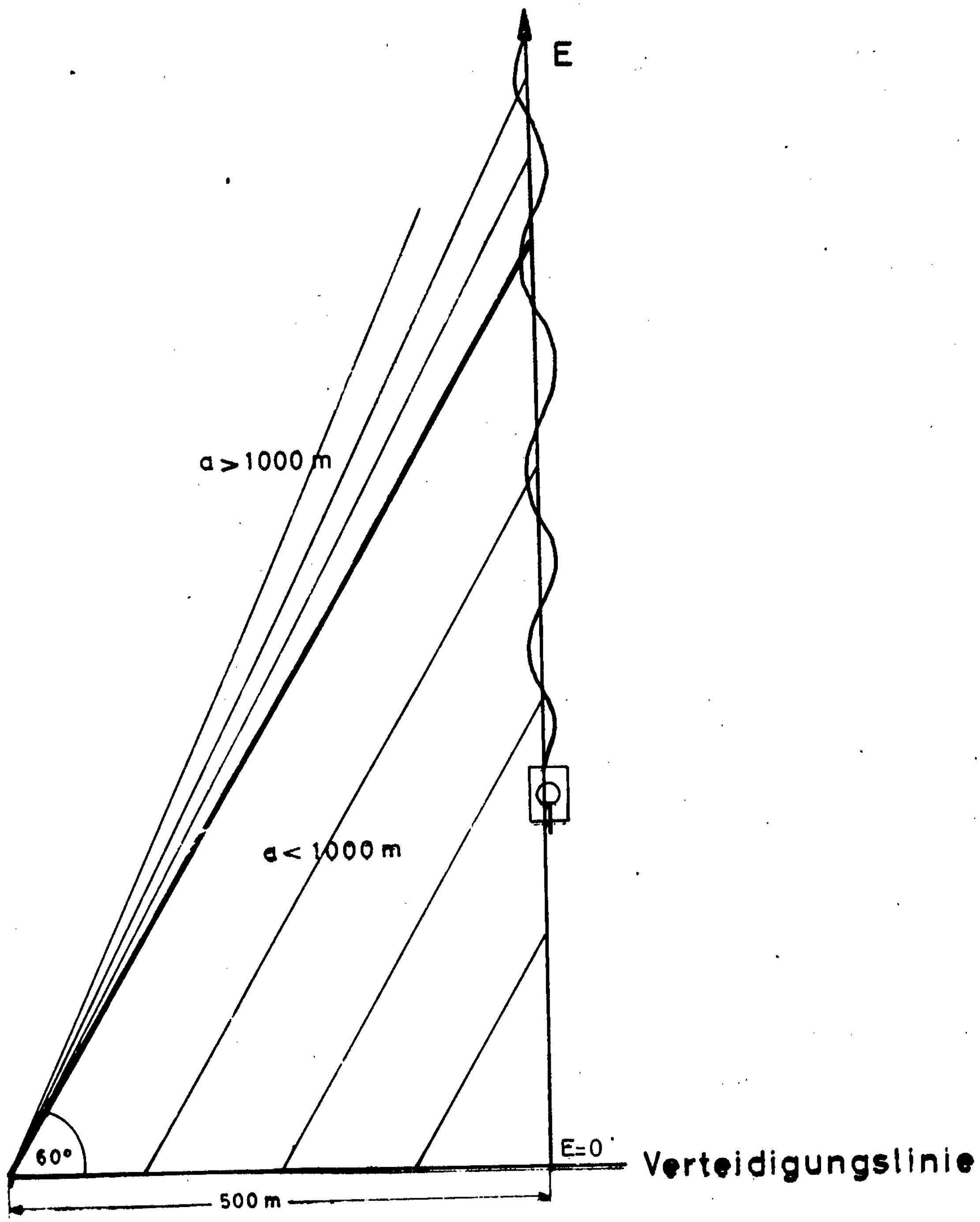
Kampfwert eines Kampffahrzeuges

Fig. 1

Fig. 2



Spezifische Motorleistung von Kampfpanzern



Strategie von Angreifer und Verteidiger

Fig. 3

Variation der Radialbeschleunigung

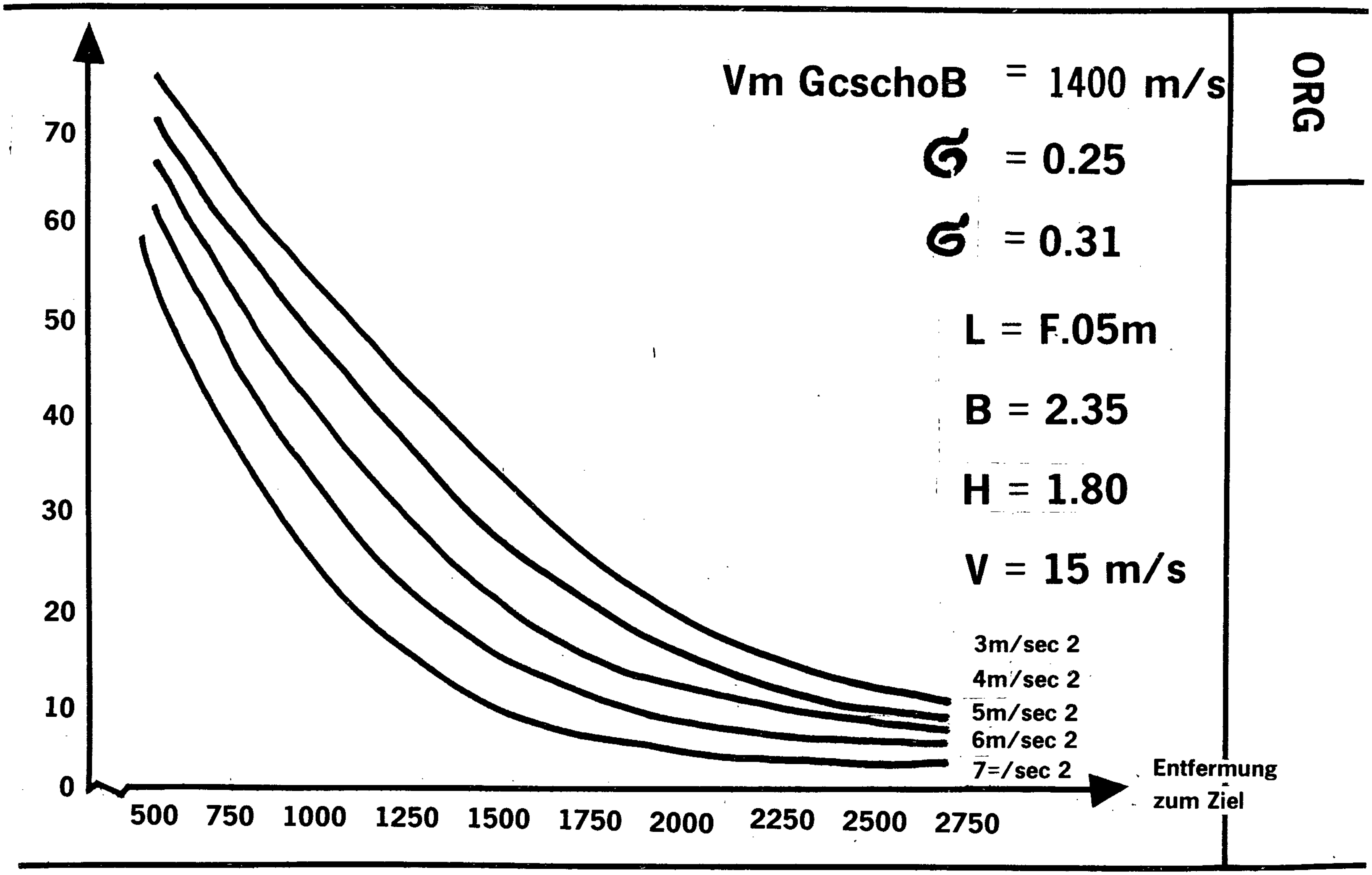


Fig. 5

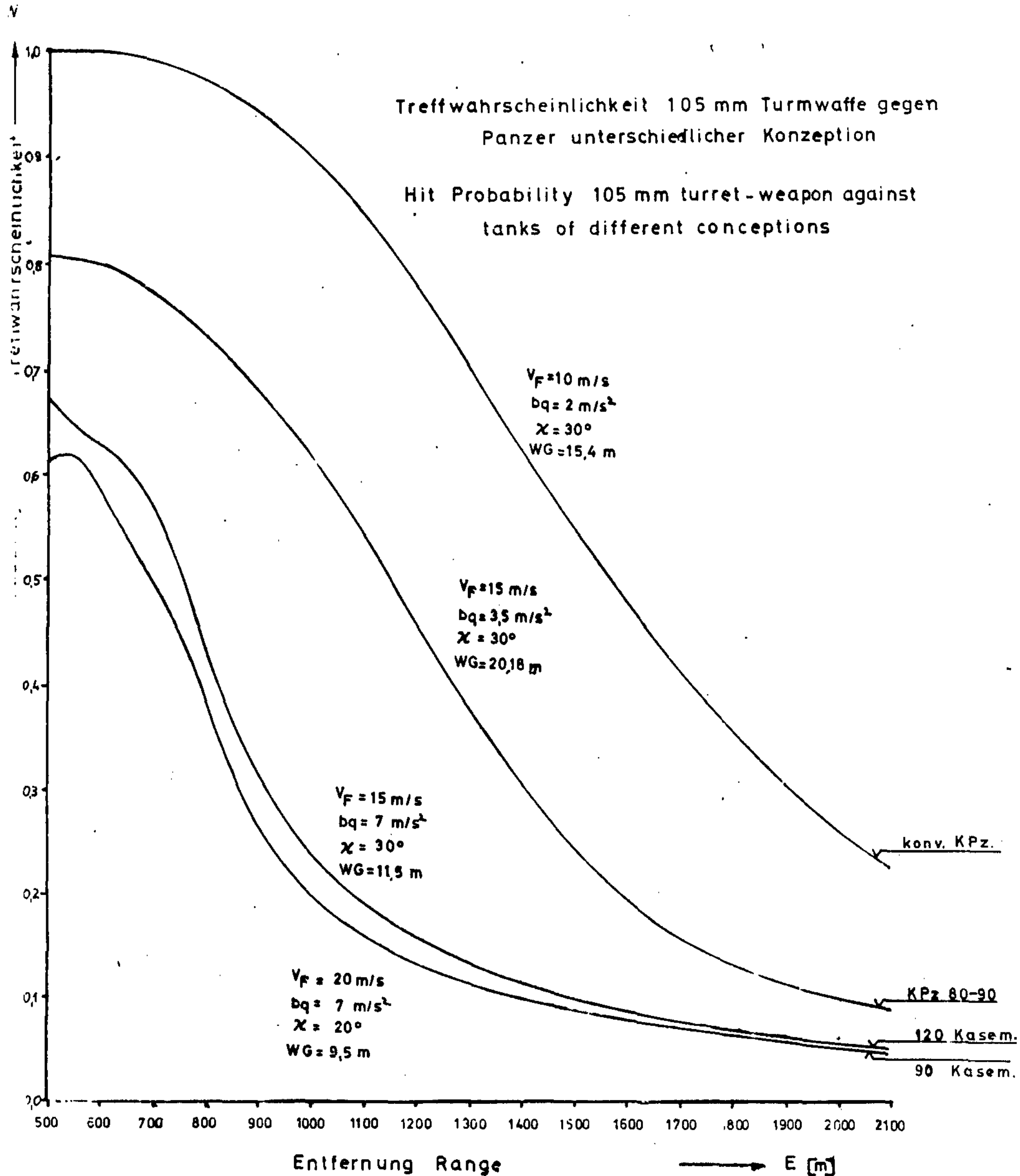

Treffwahrscheinlichkeit 105 mm Turmwaffe gegen Panzer unterschiedlicher Konzeption
Hit Probability 105 mm turret-weapon against tanks of different conceptions
V_F = 10 m/s
bq = 2 m/s²
χ = 30°
WG = 15,4 m
V_F = 15 m/s
bq = 3,5 m/s²
χ = 30°
WG = 20,18 m
V_F = 15 m/s
bq = 7 m/s²
χ = 30°
WG = 11,5 m
V_F = 20 m/s
bq = 7 m/s²
χ = 20°
WG = 9,5 m
konv. KPz.
KPz 80-90
120 Kasem.
90 Kasem.
1,0
0,9
0,8
0,7
0,6
0,5
0,4
0,3
0,2
0,1
0,0
500 600 700 800 900 1000 1100 1200 1300 1400 1500 1600 1700 1800 1900 2000 2100
Entfernung Range → E [m]

Fig. 6

Test vehicles for mobility-trials

1. Generation (M41-chassis)

LVT	FVT	RVT-1	RVT-2	KVT /HVT
Coincidence-firing with Laser	Energy storage suspension system Coincidence-firing with Laser	Mobility-trials; smoke screening	Mobility-trials	Hybrid power pack

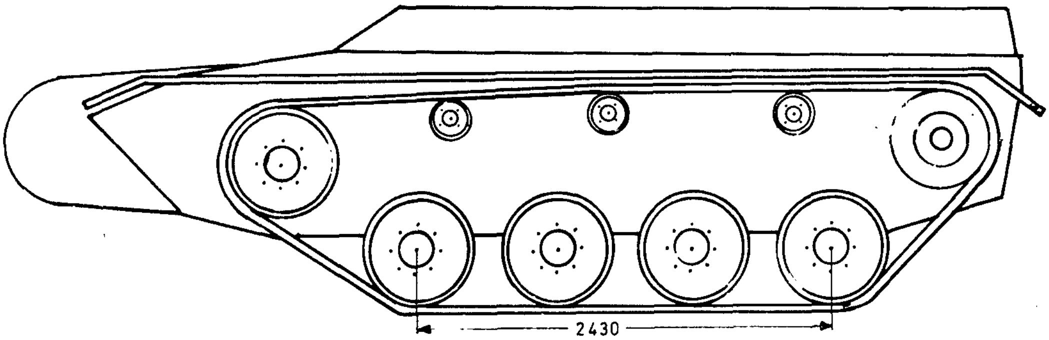
2. Generation (MBT70-chassis)

VT1-1	ET703 /VT1-2	ET706	ET707	ET708
Total concept with 2 semirigid mounted guns (105 mm)	Mobility-, tracks and suspensions trials; total concept as VT1-1 but with guns 2x120mm	Coincidence-firing trials; mobility-trials	Mobility-, tracks and suspensions trials	Mobility-, tracks and suspensions trials

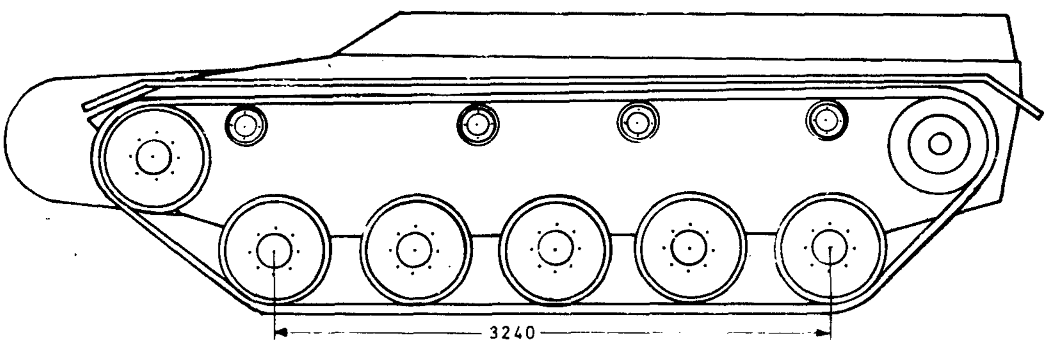
MBT Leo1-chassis

GVT01	GVT02	GVT03	GVT04	GVT05
Combat trials	Combat trials	Combat trials	Combat trials	Combat trials

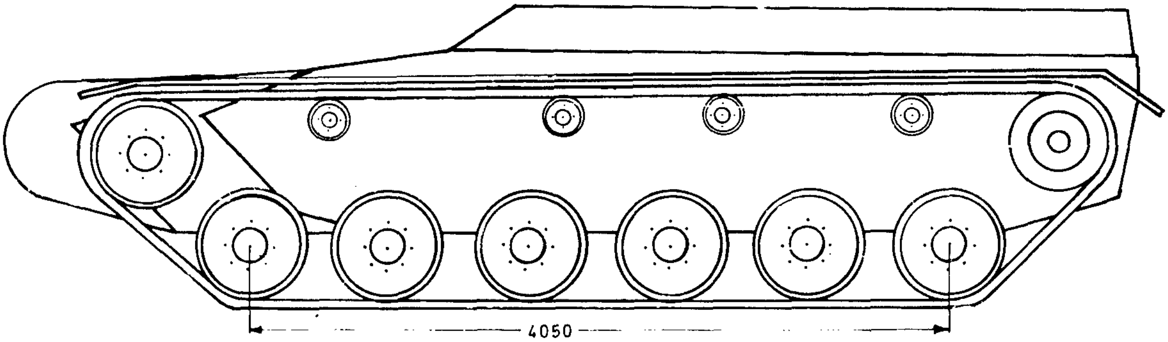
IABG TV	RVT 2	



4 Laufrollen-Ausführung Lenkverhältnis: 0,93



5 Laufrollen-Ausführung Lenkverhältnis 1,24



6 Laufrollen-Ausführüng Lenkverhältnis: 1,55

Fig. 7

Technische Daten des Versuchsfahrzeuges

Abmessungen:	Länge:	6425 mm/7235 mm bei 6 Laufr
	Breite:	3148 mm
	Höhe: ca	1800 mm
	Spurweite S:	2605,4 mm
	Kettenbreite:	530,0 mm
	Kettenteilung:	152,4 mm

	4 LR-Ausführung gefahren	in Vorber.		5 LR-Ausführung gefahren			6 LR-Ausführung gefahren
Kettenaufstandslänge	2430 mm			3240mm			4050 mm
Lenkverhältnis	0,93			1,24			1,55
Gewicht mit * Gummipads	22,38	20,9	29,62	26,1	27,85	29,44	34,77
Gewicht mit Kampfkette	22,08	20,6	29,32	25,73	27,48	29,07	34,34
rechn. spez. Bodendr. mit Kampfkette	0,868	0,8	1,15	0,75	0,8	0,857	0,8
Leistungsgew. mit Kampfkette							
a) 1800 PS-Motor	81,5	87,5	61,4	70	65,5	61,9	52,4
b) 2300 PS-Motor	104	111,7	78,4	89,4	83,7	79,1	67,0

* Ausgleich über Verminderung der Zusatzgewichte vorgesehen.

Fahrzeuggeschwindigkeiten bei n_{Motor} = 2600 U/min

(i = 4,44)			(i = 3,809)		
1. Gang	12.9	km/h	1.Gg.	15.04	km/h
2. Gang	23.02	km/h	2.Gg.	26.83	km/h
3. Gang	35.7	km/h	3.Gg.	41.61	km/h
4. Gang	59.41	km/h	4.Gg.	69.25	km/h

i = Vorgelege

Fig. 8

Fig. 9

IABG TV	Meßstellenplan für fahrdynamische Untersuchungen am RVT 2	

Meßstelle	Beschleunigungsfahrt geradeaus	Beharrungsfahrt geradeaus	Ausrollversuch	Kreisbogenfahrt	Wenden u. Hochachse	Wedelfahrt	Ermittlung rotat. Massenkräfte	statistische Fahrten
Md_{Turb}	X	X		X		X	X	X
Md_{Lenk}				X	X	X	X	X
Md_{Keli}	X	X	X	X	X	X	X	X
Md_{Kere}	X	X	X	X	X	X	X	X
n_{Mot}	X			X	X	X	X	X
n_{Turb}	X	X		X		X	X	X
n_{Keli}	X	X	X	X	X	X	X	X
n_{Kere}	X	X	X	X	X	X	X	X
b längs	X		X	X		X		
b quer				X		X		X
V tats. läng	X	X	X					
S tats. läng	X		X					
Fahrkurs				X		X		
Gierwinkel				X	X			
Giergeschwind.				X	X	X		
Lenkdrücke kl. Radius li. re.				X	X	X	X	X
Lenkdrücke gr. Radius li. re.				X	X	X		X
Wandler Kuppl.	X			X		X		
1÷4 Ganglage	X						X	
1÷4 Schaltdrücke	X						X	
Nickwinkel						X		
Schwimmwinkel				X		X		
Wankwinkel						X		

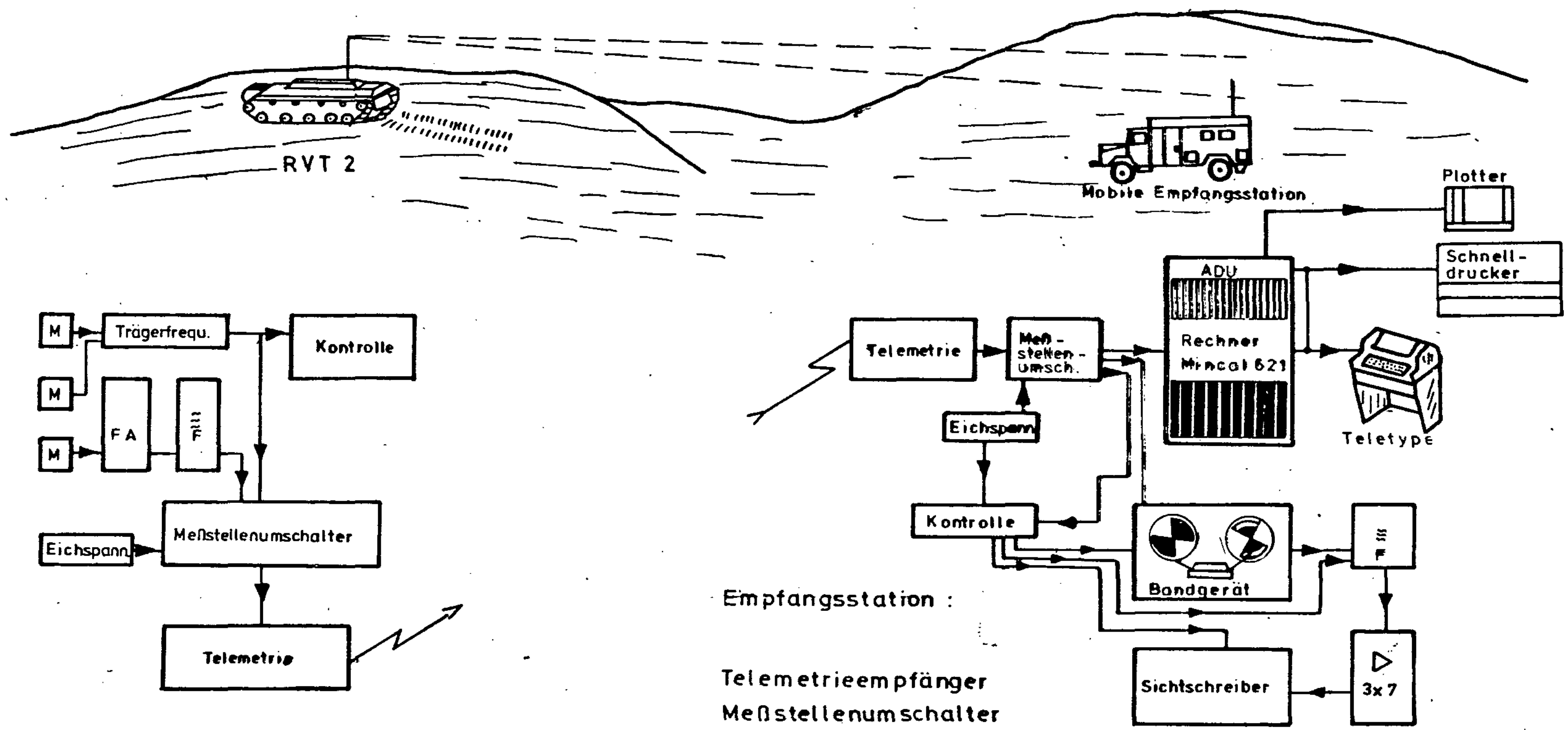
Plattform

Anmerkung:
Eine Reihe von Meßstellen sind ineinander umrechenbar. Sie sind als Kontrolle bzw. Ersatz gedacht.

versuchsbegleitende Bodenuntersuchungen:
- Penetrometerversuche
- Bodenfeuchtigkeitsbestimmung

IABG
TV

Fig. 1o



Kettenfahrzeug:

Meßstellen 1 ÷ 14
KWS
Filter
Meßstellenumschalter
Kontrolleinheit
Eichspannung
FA Wandler
Telemetriesender

Empfangsstation :

Telemetrieempfänger
Meßstellenumschalter
Rechner
Filter
Verstärker für Registr. Schreiber
Kontrolleinheit
Eichspannung
Analogbandgerät

Erweiterung:
1. unbem. Fernbedienung
2. Steuerung des Versuchsträgers über Rechner

TEST METHODOLOGY

Presented By

A. COMITO
U.S.

Ron Beck
Computer Simul.

THIS BRIEFING DESCRIBES OUR EFFORTS IN THE AREA OF TEST METHODOLOGY DEVELOPMENT AS APPLIED TO OUR MILITARY VEHICLES - PRIMARILY IN THE AREAS OF VIBRATION AND DURABILITY TESTING.

VG# 1 TEST METHODOLOGY IS DEFINED AS THE APPLICATION OF SCIENTIFIC DISCIPLINE TO THE TESTING OF VEHICLES UNDER THEIR SIMULATED OR RE-CREATED OPERATIONAL ENVIRONMENT IN THE LABORATORY.

VG# 2 THE TEST METHODOLOGY GROUP IS AN ORGANIZATIONAL ELEMENT UNDER THE ENGINEERING SCIENCE DIVISION.

VG# 3 THE PRIME OBJECTIVE OF TEST METHODOLOGY IS TO IMPROVE RELIABILITY OF FIELDED VEHICLES BY DECREASING DEFICIENCIES.

MANY YEARS ARE REQUIRED TO FIELD A NEW VEHICLE. A SUBSTANTIAL PORTION OF THIS TIME IS USED IN CORRECTING TECHNICAL PROBLEMS DURING THE DEVELOPMENT, ADVANCED PRODUCTION ENGINEERING AND IN EARLY PRODUCTION PHASES. THE COST AND TIME REQUIRED TO CORRECT DEFICIENCIES INCREASES DRASTICALLY AS WE MOVE FROM CONCEPT TO PRODUCTION. THIS VIEWGRAPH SHOWS THE
VG# 4 RELATIVE COST OF CORRECTING DEFICIENCIES THROUGHOUT A VEHICLE'S LIFE CYCLE. AS CAN BE SEEN, THERE IS A BIG PAYOFF IN ELIMINATING DEFICIENCIES EARLY IN THE DEVELOPMENT CYCLE.

IN ORDER TO ACHIEVE THIS OBJECTIVE, TEST METHODOLOGY PERFORMS THREE BASIC FUNCTIONS: IT MEASURES, SIMULATES AND VALIDATES.

THE FIRST TASK IS TO OBTAIN DATA ON THE ENVIRONMENT IN WHICH THE VEHICLE, ITS SUB-SYSTEMS AND COMPONENTS MUST LIVE. THE TEST METHODOLOGY SUB-FUNCTION DIRECTS AND MANAGES FIELD TESTS TO MEASURE THIS DATA. THIS DATA BECOMES PART OF A PERMANENT DATA BANK WHICH CAN BE USED FOR DESIGN OF COMPONENTS AND THEIR INTEGRATION INTO A VEHICLE SYSTEM. TOGETHER WITH OUTPUTS FROM MATHEMATICAL MODEL SIMULATIONS, THIS DATA BANK CAN BE APPLIED TO VEHICLE DESIGN IN THE CONCEPT STAGE. THE VEHICLE DESIGN CAN BE VERIFIED LATER WITH MEASUREMENTS OFF A PROTOTYPE.

VG# 5 THE METHOD OF MEASURING DATA IS AS FOLLOWS:

A VEHICLE IS INSTRUMENTED BY PLACING SENSORS ON SELECTED COMPONENTS OF A SYSTEM SUCH AS THE SUSPENSION OR DRIVE TRAIN. THE VEHICLE IS THEN RUN OVER SELECTED FIELD COURSES TO MEASURE ACCELERATION, FORCE, TORQUE, PRESSURE, POSITION, ETC. THE DATA IS RECORDED ON MAGNETIC TAPE.

VG# 6 TO HELP IN RECORDING THIS DATA MORE EFFICIENTLY, A MOBILE INSTRUMENT VAN WAS BUILT AND OUTFITTED WITH AN FM TELEMETRY STATION.

VG# 7 THIS PERMITS MORE FLEXIBILITY IN OUR TEST OPERATIONS AND REMOVES THE RECORDING EQUIPMENT
VG# 8 FROM THE TEST VEHICLE. THE BIGGEST ADVANTAGE, HOWEVER, IS HAVING THE CAPABILITY FOR INSTANT READ-OUT OF THE DATA.

AFTER THE DATA IS RECORDED THE MAJOR TASK OF REDUCING THAT DATA FOR DESIGN ANALYSIS AND/OR LABORATORY SIMULATION IS PERFORMED.

VG# 9 THE TAPE MUST BE EDITED AND PUT INTO A USABLE FORMAT.

VG# 10 DATA REDUCTION IS PERFORMED BY COMPUTER AND THE PROCESS IS ILLUSTRATED IN THE VIEWGRAPH. THIS PROCESS WAS USED TO REDUCE VIBRATION DATA ACQUIRED ON COMPONENTS OF THE M813 5-TON CARGO TRUCK.

VG# 11 12 13 SAMPLES OF THE FINAL PRODUCT IS IN THE FORMAT SHOWN IN THE FOLLOWING VIEWGRAPHS:

13-2 WE HAVE DATA SHEETS LIKE THESE ON MAJOR MECH. AND ELECT. COMP. OF THE M813 VEHICLE TESTED OVER THE MUNSON & PERRYMAN COURSES AT APG.

TO DATE, FIELD TEST DATA HAS BEEN ACQUIRED ON THE FOLLOWING:

VG# 14

- M813 5-TON CARGO TRUCK, 6 X 6
- M151 1/4-TON TRUCK
- XM705 1 1/4-TON TRUCK
- SAM-D TRAILER
- GOER 8-TON CARGO
- M656 5-TON TRUCK, 8 X 8

AFTER THE DATA IS ACQUIRED, METHODS ARE APPLIED TO RE-CREATE THE OPERATIONAL ENVIRONMENT IN THE LABORATORIES. HERE WE WORK CLOSELY WITH PERSONNEL IN THE ARMOR, AND COMPONENTS DIVISION OF THE MOBILITY LABORATORY AS WELL AS THE PROPULSION SYSTEMS LABORATORY. MUCH OF OUR TESTING TO-DATE HAS BEEN IN THE AREA OF VIBRATION - OF COMPLETE VEHICLES AS WELL

AS INDIVIDUAL COMPONENTS. OUR APPROACH IN THIS AREA HAS BEEN TO GENERATE A SIGNAL TO
DRIVE THE ACTUATORS IN THE SAME MANNER AS IF IT WERE IN THE FIELD. THIS HAS BEEN DONE
VG# 15 WITH TAPED DATA OR SHAPED RANDOM NOISE.

FOR PURPOSES OF CORRELATING THE FIELD DATA TO THE LABORATORY SIMULATION DATA, WE EXAMINE
VG# 16 BOTH FIELD AND LABORATORY WAVEFORM CHARACTERISTICS OF ACCELERATION.

BY DUPLICATING THESE CHARACTERISTICS WE HOPE TO PRODUCE THE SAME EFFECT ON THE VEHICLE
SYSTEM OR COMPONENT IN THE LABORATORY THAT IT SEES IN FIELD OPERATION.

OVER THE PAST FEW YEARS, WE HAVE LOOKED AT THREE OF THE BASIC TECHNIQUES FOR DYNAMIC
VG# 17 SIMULATION. THESE ARE AS SHOWN IN THE VIEWGRAPH. AS PART OF OUR METHODOLOGY DEVELOPMENT,
WE ARE NOW ATTEMPTING TO USE ALL THREE TECHNIQUES ON THE SAME VEHICLE FOR THE PURPOSE OF
RANKING THE DEGREE OF CORRELATION. THE TEST BED USED FOR OUR EXPERIMENT IS THE M813
VG# 18 5-TON CARGO TRUCK SHOWN HERE MOUNTED ON HYDRAULIC ACTUATORS. THE ACTUATORS SHOWN WERE
PURCHASED FOR THIS PURPOSE AND, TOGETHER WITH THE MOUNTS TO THE AXLE, WERE DESIGNED TO
BE UNIVERSAL FOR WHEELED VEHICLE APPLICATIONS. WE HAVE RUN THIS SIMULATION DIRECTLY FROM
OUR TAPED DATA WITH REASONABLY GOOD RESULTS. WE PLAN TO USE THE OUTPUT FROM A MATHEMATICAL

MODEL AS WELL AS SHAPED RANDOM NOISE FOR ACTUATOR EXCITATION. MR. GRANT HAS DEVELOPED A MATHEMATICAL MODEL FOR THIS VEHICLE WHICH CAN READILY BE PROGRAMMED ON THE ANALOG COMPUTER. THE COMPUTER SIMULATION WILL USE A SIMILAR TERRAIN PROFILE AS THAT OVER WHICH THE M813 WAS TESTED. THE OUTPUT WILL BE A DISPLACEMENT SIGNAL TO THE ACTUATORS. THE SHAPED RANDOM NOISE EXCITATION WILL HAVE THE SAME POWER SPECTRAL DENSITY AS THAT FROM THE FIELD DATA. THIS BRIEFLY DESCRIBES OUR ATTEMPTS IN THE AREA OF VIBRATION. OTHER TESTS HAVE BEEN PERFORMED BESIDES VIBRATION TESTS WHICH INCLUDE THE FOLLOWING:

VG# 19

M813 BRAKE SIMULATION TEST

M817 COOLING TEST

XM705 1 1/4-TON TRUCK - POWER TRAIN SIMULATION

M151 1/4-TON TRUCK - ENGINE FIELD SIMULATION ENDURANCE TEST

GOER ENGINE/TRANSMISSION FIELD SIMULATION TEST 8-TON, 6 X 6

THE PROCEDURES FOLLOWED IN THESE TESTS WERE TO DUPLICATE THE CYCLES AND CONDITIONS RECORDED ON TAPE IN THE LABORATORY.

THE TEST OF THE GOER TRANSMISSION WAS PARTICULARLY SUCCESSFUL IN THAT IT DEMONSTRATED HOW A DEFICIENCY WAS FOUND AND CORRECTED PRIOR TO PRODUCTION WHICH RESULTED IN OVER A MILLION

DOLLAR COST SAVINGS TO THE GOVERNMENT. USING TEST DATA PREVIOUSLY MEASURED IN THE FIELD, A NEW ENGINE/TRANS MADE BY A DIFFERENT MANUFACTURER SHOWED A COOLING DEFICIENCY IN THE LABORATORY TEST THAT WAS NOT RECOGNIZED IN FIELD TESTS OF THE PROTOTYPE. AT THAT TIME A NEW CONTRACT HAD ALREADY BEEN SIGNED CALLING FOR APPROXIMATELY 1300 VEHICLES.

VG# 20 IN SUMMARY, TEST METHODOLOGY POINTS THE WAY TO IMPROVED TESTING IN THE LABORATORY. IN GENERAL, THESE ARE THE STEPS FOLLOWED. THE RESULTS OF OUR EFFORT WILL BE THE FOLLOWING BENEFITS:

VG# 21

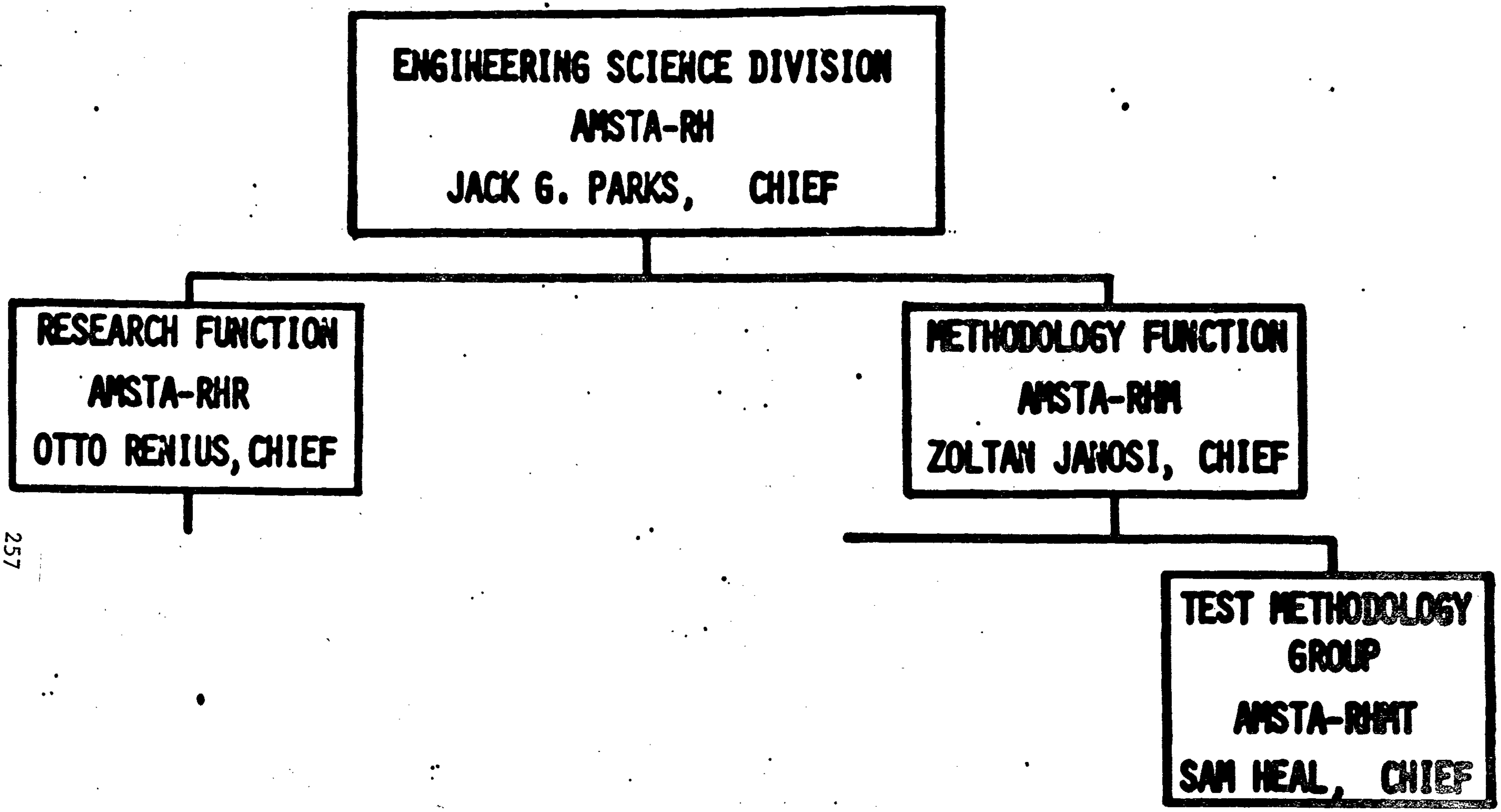
- IMPROVED RELIABILITY, AVAILABILITY, MAINTAINABILITY, AND DURABILITY OF FUTURE AND FIELDED VEHICLES.
- COST SAVINGS AND SHORTENING OF THE MATERIAL ACQUISITION PROCESS.
- DEFINITIVE AND REALISTIC SPECIFICATIONS FOR PROCUREMENT OF VEHICLE COMPONENTS.
- GENERATION OF DUTY CYCLE DATA BANKS FOR DESIGN APPLICATION AND SOLUTION OF FIELDED VEHICLE PROBLEMS.

THIS CONCLUDES MY PRESENTATION. MR. GRANT WILL FURTHER DISCUSS SOME OF THE VIBRATION TECHNIQUES MENTIONED EARLIER ALONG WITH HIS RESULTS.

TEST METHODOLOGY

DEFINITION:

APPLICATION OF SCIENTIFIC DISCIPLINE
TO TESTING OF VEHICLE SYSTEM AND COMPONENTS
UNDER SIMULATED OPERATIONAL ENVIRONMENT

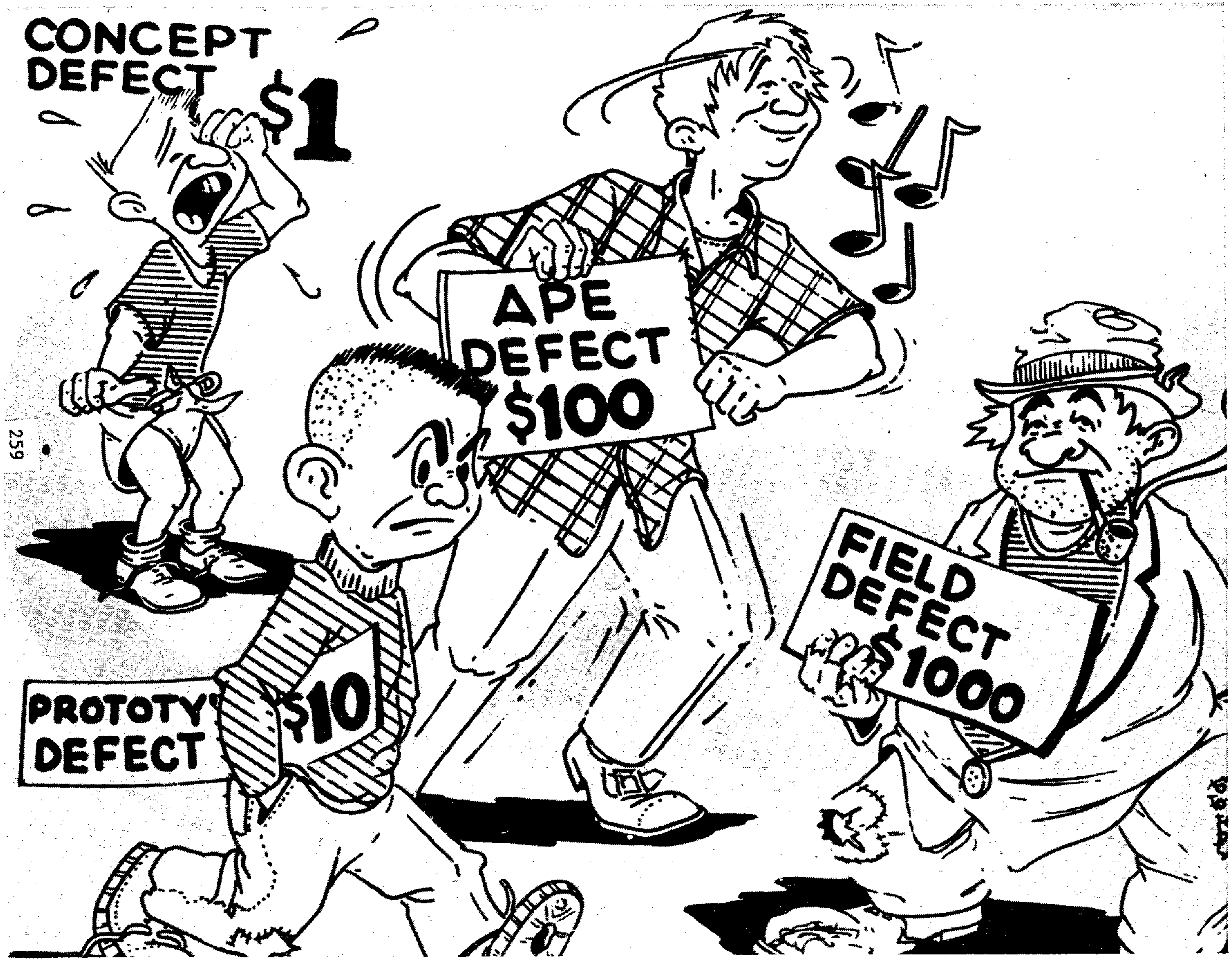

ENGINEERING SCIENCE DIVISION
AMSTA-RH
JACK G. PARKS, CHIEF
RESEARCH FUNCTION
AMSTA-RHR
OTTO RENIUS, CHIEF
METHODOLOGY FUNCTION
AMSTA-RHM
ZOLTAN JANOSI, CHIEF
TEST METHODOLOGY GROUP
AMSTA-RHMT
SAM HEAL, CHIEF

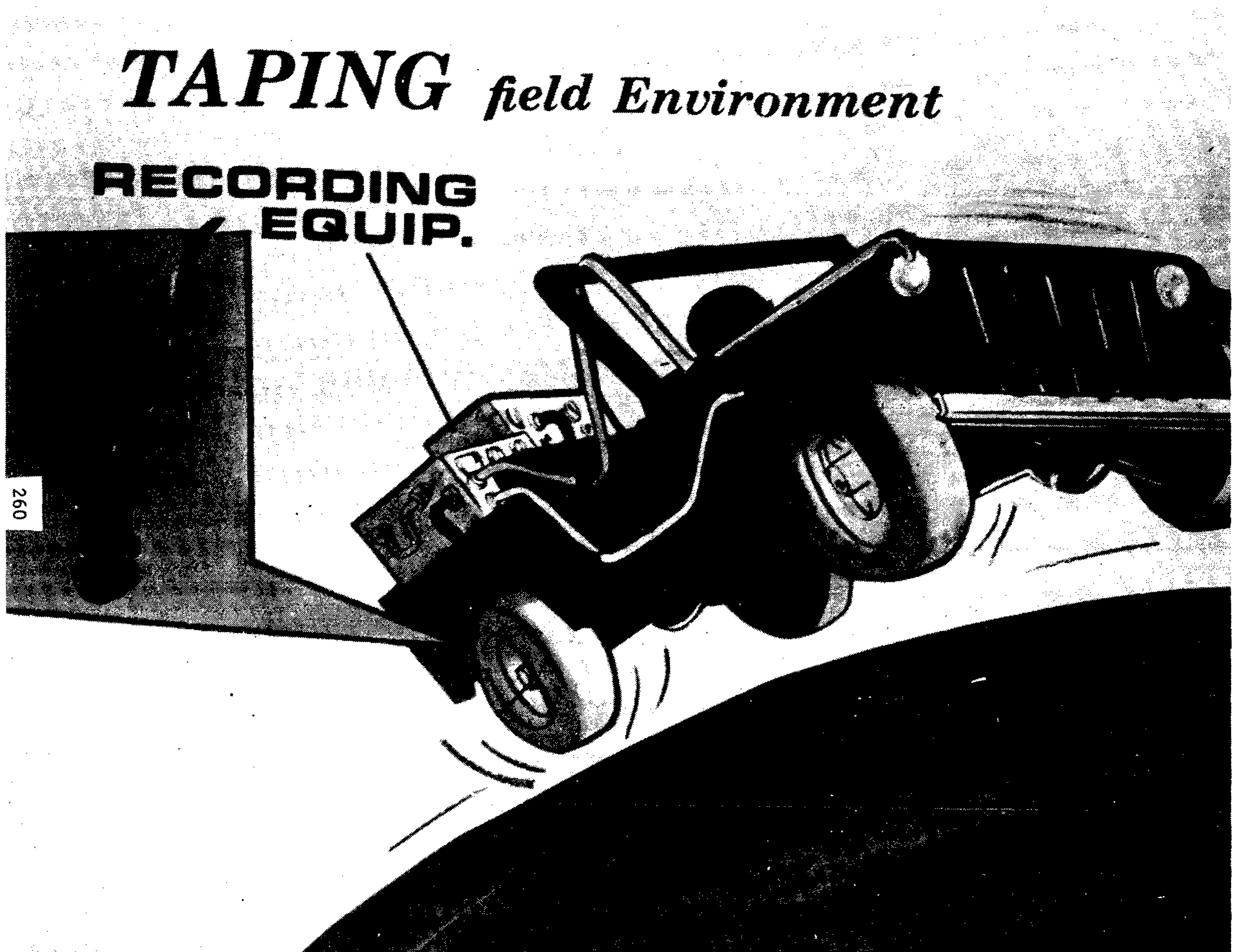
TEST METHODOLOGY

OBJECTIVE:

TO IMPROVE VEHICLE RELIABILITY BY DECREASING DEFICIENCIES

U.S. ARMY TANK AUTOMOTIVE COMMAND


CONCEPT
DEFECT
$1
APE
DEFECT
$100
PROTOTY
DEFECT
$10
FIELD
DEFECT
$1000

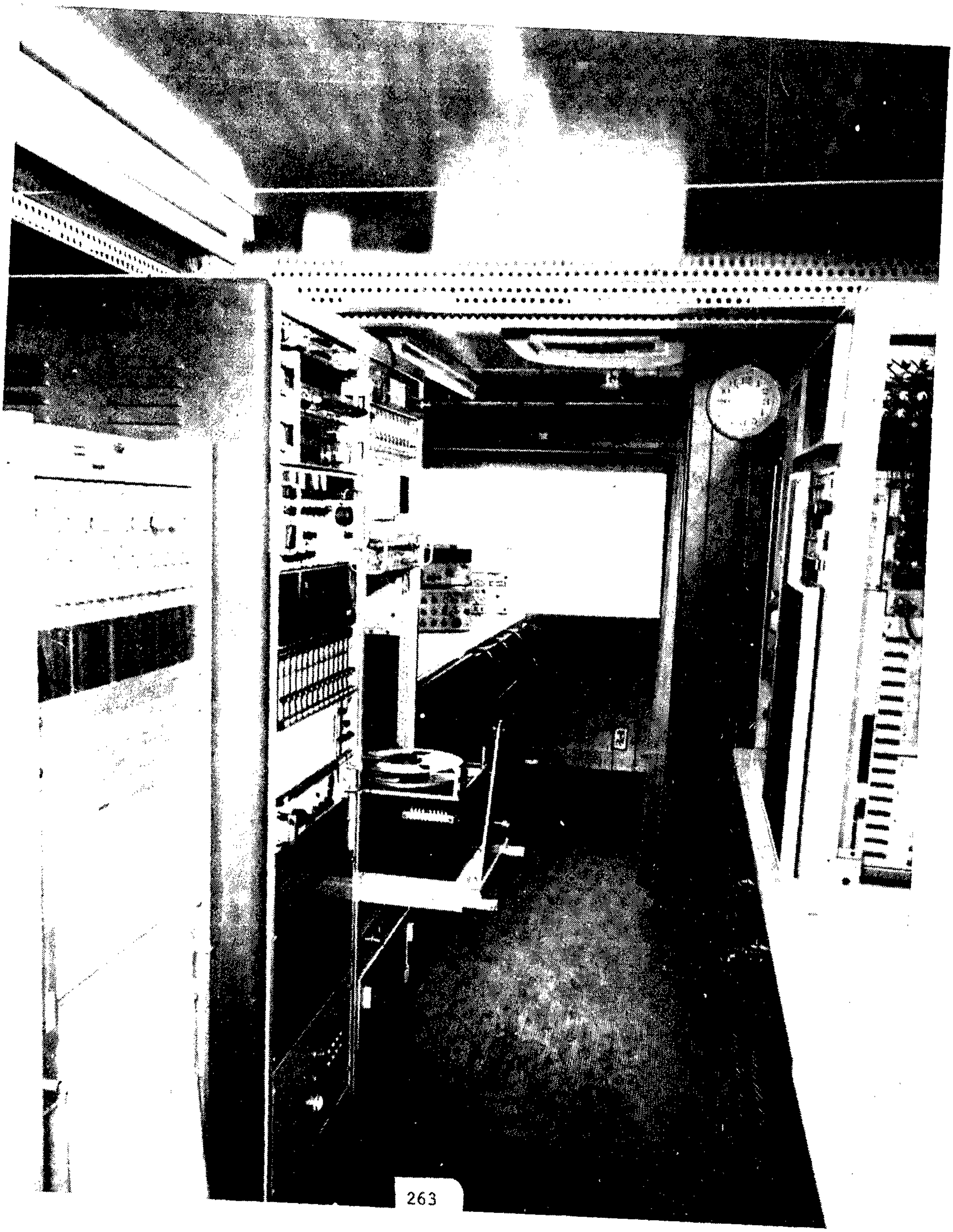

TAPING field Environment
RECORDING EQUIP.

MOBILE INSTRUMENT VAN IS AVAILABLE.

TWO 14 CHANNEL MAGNETIC TAPE RECORDERS
ON BOARD

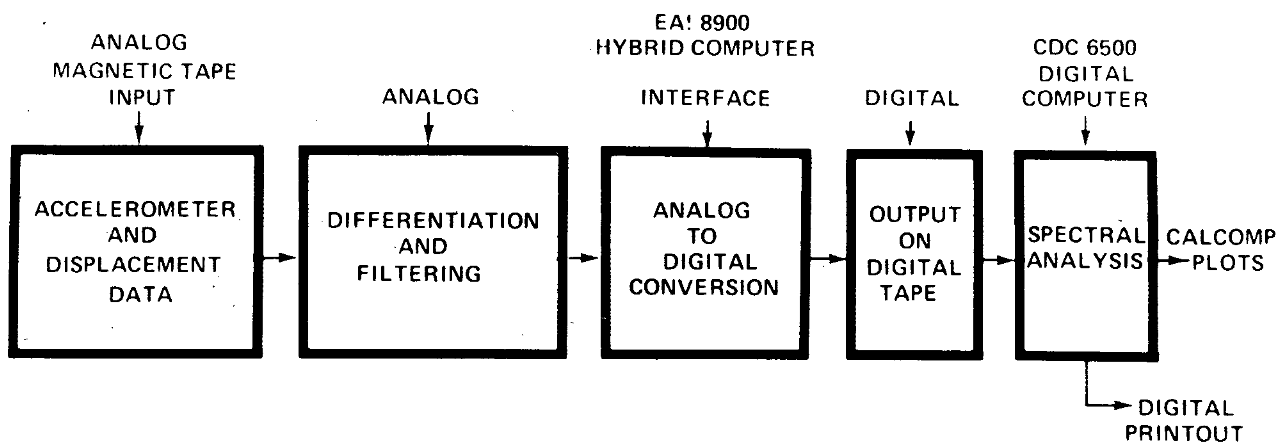
DATA TRANSMISSION: TELEMETRY

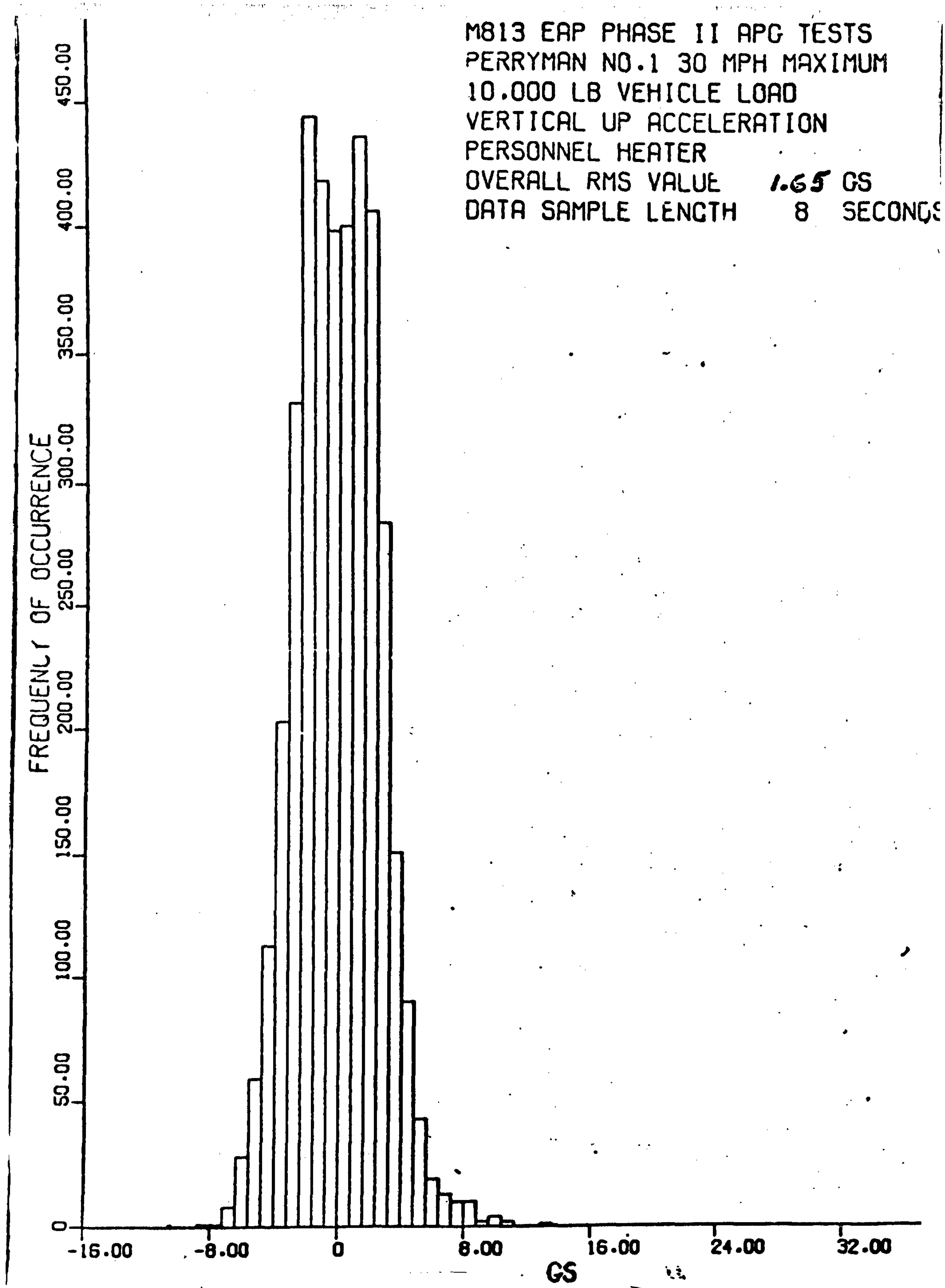

U.S.ARMY TANK-AUTOMOTIVE COMMAND
MOBILE LABORATORY
ENGINEERING AUDIT

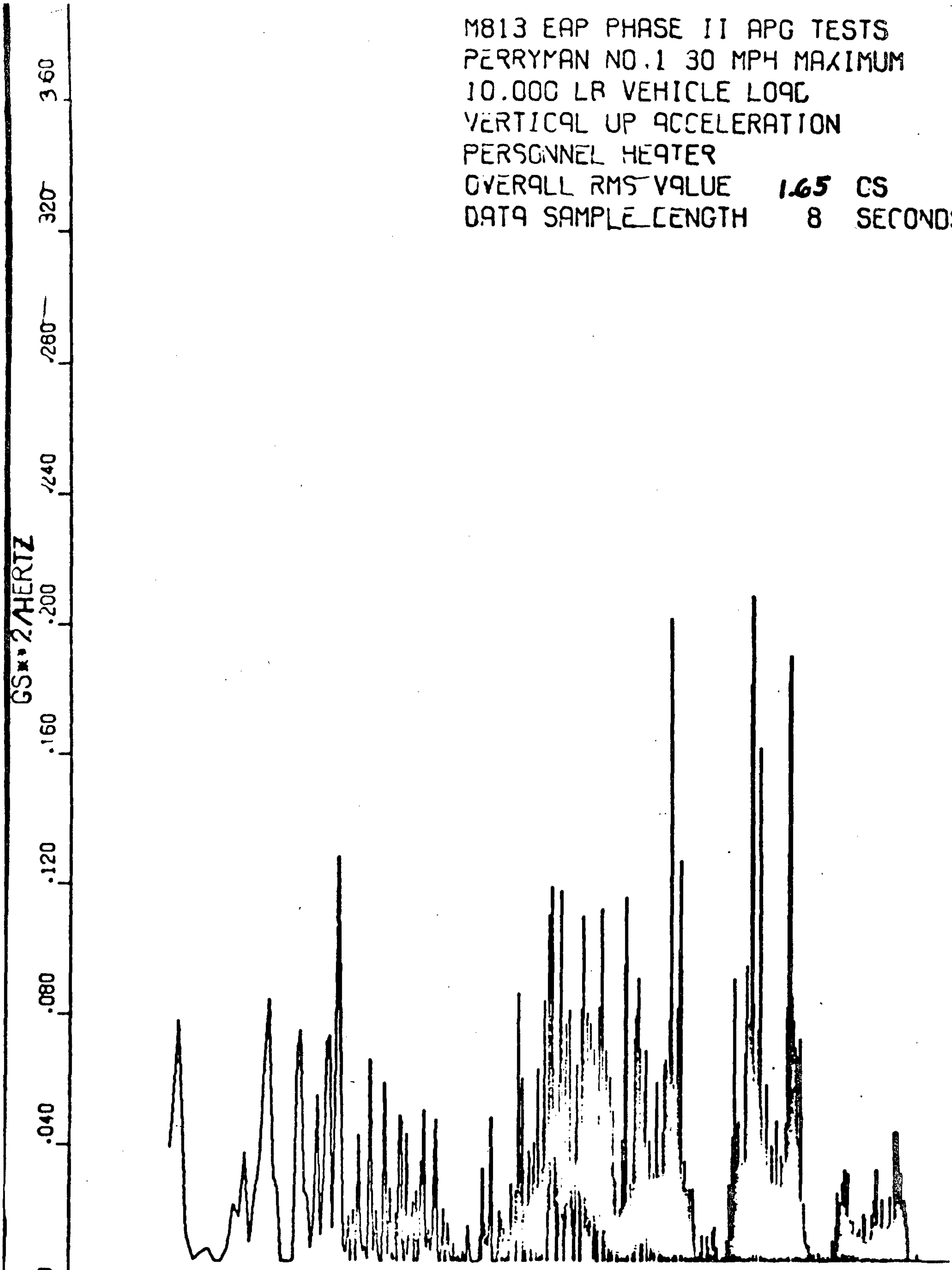


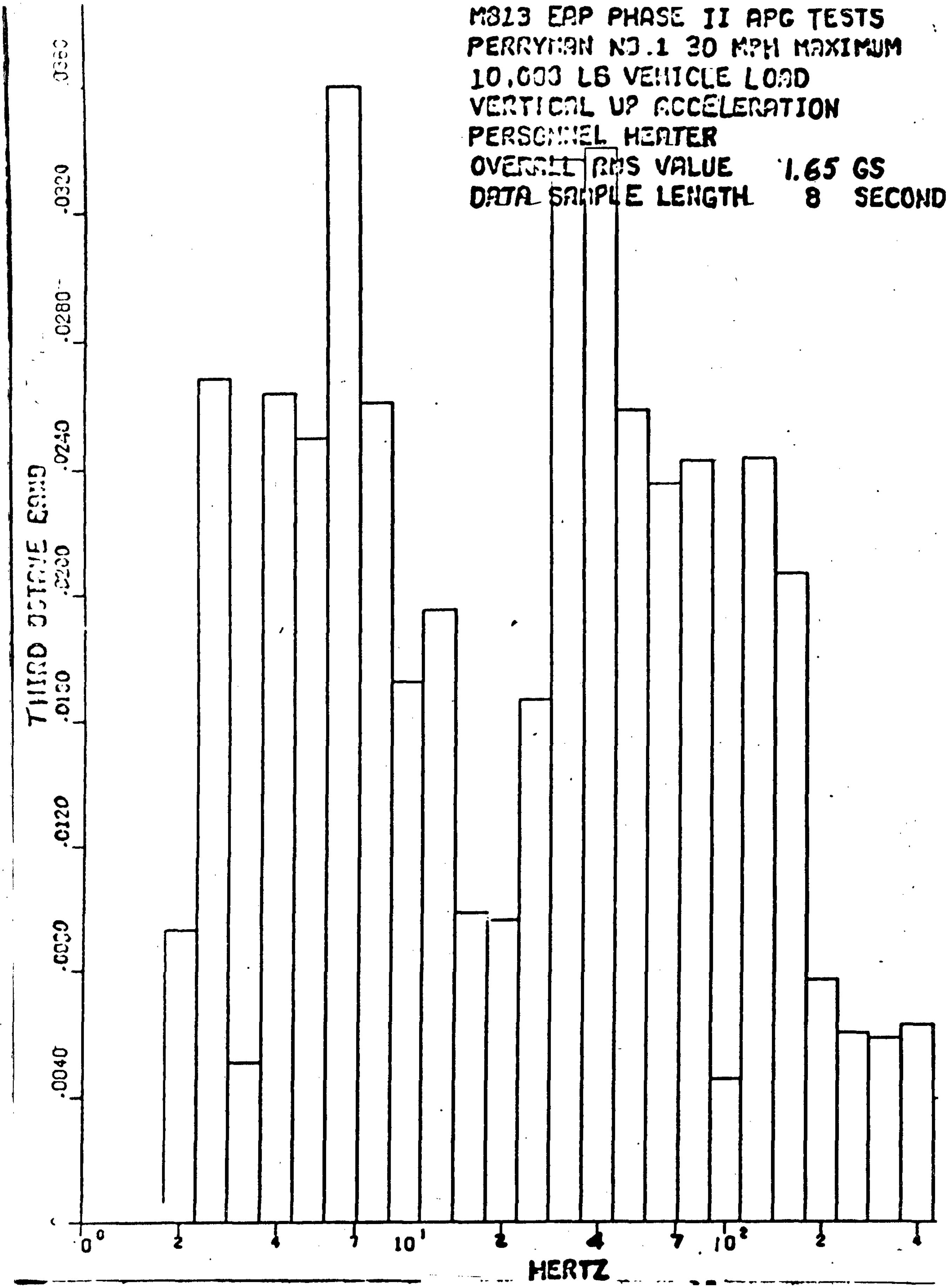

Editing
FIELD
ENVIRONMENT
TAPE

HYBRID SPECTRAL DATA REDUCTION

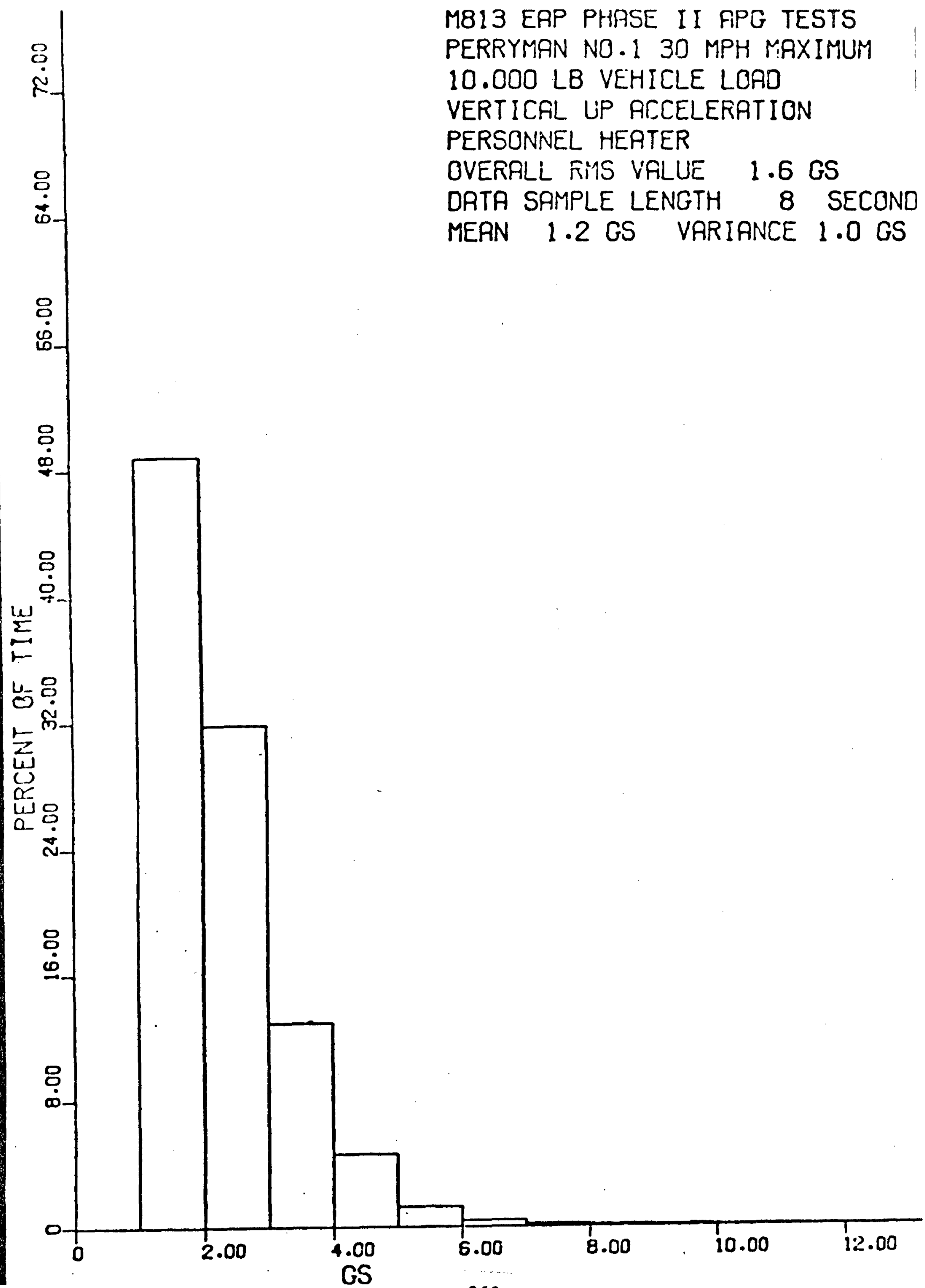



M813 EAP PHASE II APG TESTS
PERRYMAN NO.1 30 MPH MAXIMUM
10.000 LB VEHICLE LOAD
VERTICAL UP ACCELERATION
PERSONNEL HEATER
OVERALL RMS VALUE 1.65 GS
DATA SAMPLE LENGTH 8 SECONDS
FREQUENCY OF OCCURRENCE
450.00
400.00
350.00
300.00
250.00
200.00
150.00
100.00
50.00
0
-16.00
-8.00
0
8.00
16.00
24.00
32.00
GS


M813 EAP PHASE II APG TESTS
PERRYMAN NO.1 30 MPH MAXIMUM
10.000 LB VEHICLE LOAD
VERTICAL UP ACCELERATION
PERSONNEL HEATER
OVERALL RMS VALUE 1.65 GS
DATA SAMPLE LENGTH 8 SECONDS
GS**2/HERTZ
0
.040
.080
.120
.160
.200
.240
.280
.320
.360
10^0
2
4
7
10^1
2
4
7
10^2
2
4
HERTZ



M813 EAP PHASE II APG TESTS
PERRYMAN NO.1 30 MPH MAXIMUM
10,000 LB VEHICLE LOAD
VERTICAL UP ACCELERATION
PERSONNEL HEATER
OVERALL RMS VALUE 1.65 GS
DATA SAMPLE LENGTH 8 SECOND
THIRD OCTAVE BAND
.0040
.0120
.0160
.0240
.0280
.0320
HERTZ
10^0
2
4
7
10^1
2
4
7
10^2
2
4


M813 EAP PHASE II APG TESTS
PERRYMAN NO.1 30 MPH MAXIMUM
10.000 LB VEHICLE LOAD
VERTICAL UP ACCELERATION
PERSONNEL HEATER
OVERALL RMS VALUE 1.6 GS
DATA SAMPLE LENGTH 8 SECOND
MEAN 1.2 GS VARIANCE 1.0 GS
PERCENT OF TIME
0
8.00
16.00
24.00
32.00
40.00
48.00
56.00
64.00
72.00
0
2.00
4.00
6.00
8.00
10.00
12.00
GS

FIELD DATA ACQUIRED FOR:

- M813 5-TON CARGO TRUCK, 6 X 6
- M151 1/4-TON TRUCK
- XM705 1 1/4-TON TRUCK
- SAM-D TRAILER
- GOER 8-TON CARGO
- M656 5-TON TRUCK, 8 X 8

SHAKER EXCITATION

1. THE RECORDED TIME HISTORY CAN BE REPRODUCED AND USED TO DRIVE THE SHAKER.
2. A RANDOM NOISE SIGNAL WITH GAUSSIAN OR NORMALLY DISTRIBUTED AMPLITUDES AND APPROPRIATE SPECTRAL SHAPING CAN BE USED TO GENERALLY PRODUCE THE ESSENTIAL CHARACTERISTICS OF THE SERVICE ENVIRONMENT.

EXAMINE FIELD AND LABORATORY ACCELERATION WAVEFORMS WITH REGARD TO:

1. THE SPECTRAL CHARACTERISTICS, I.E., THE VARIATION OF INTENSITY WITH FREQUENCY.

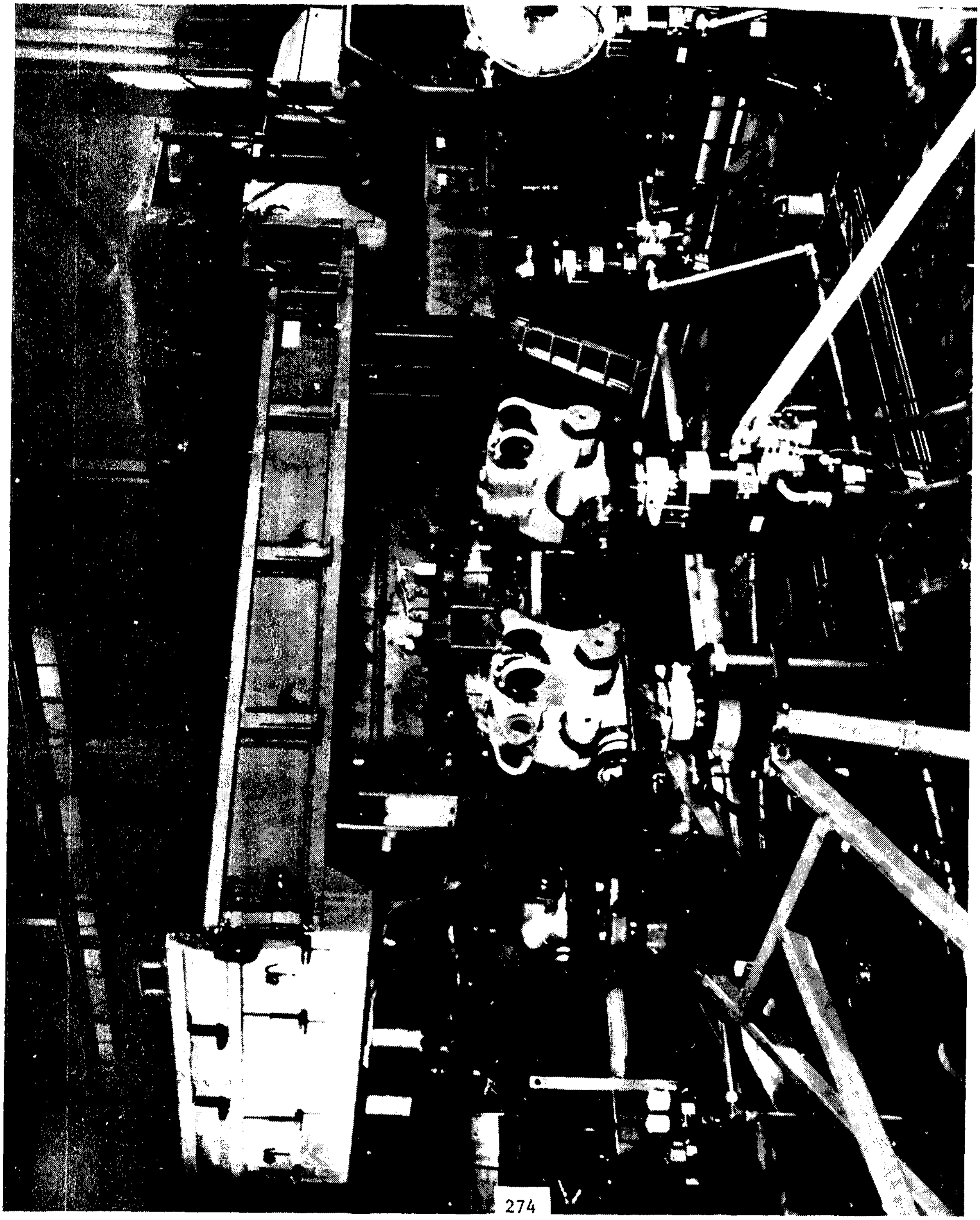
2. THE STATISTICAL CHARACTERISTICS OF EITHER THE INSTANTANEOUS OR PEAK VALUES OF THE WAVEFORM IN TERMS OF THE APPROPRIATE PROBABILITY DENSITY FUNCTIONS OR CORRELATION FUNCTIONS.

LOAD SPECTRUM TESTING TECHNIQUES

DUPLICATION - CONTROL OF SHAKER BY PLAY-BACK OF FIELD LOAD DATA TAPE RECORDING

SIMULATION - SHAKER EXCITATION SIGNAL GENERATED FROM PREDETERMINED MATHEMATICAL MODEL SIMULATION ON THE ANALOG COMPUTER USING MEASURED TERRAIN PROFILES.

SYNTHESIS - ARTIFICIALLY CREATED VIBRATION TO RECREATE THE FIELD LOAD HISTORY



M813 BRAKE SIMULATION TEST

M817 COOLING TEST

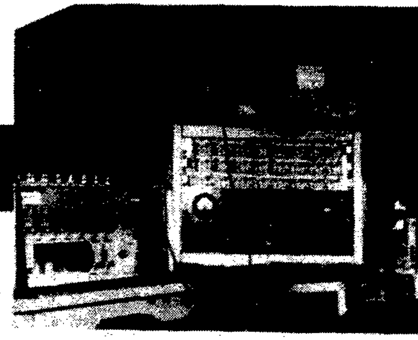
XM705 1 1/4-TON TRUCK - POWER TRAIN SIMULATION

M151 1/4-TON TRUCK - ENGINE FIELD SIMULATION ENDURANCE TEST

GOER ENGINE/TRANSMISSION FIELD SIMULATION TEST

TEST METHODOLOGY
The Process

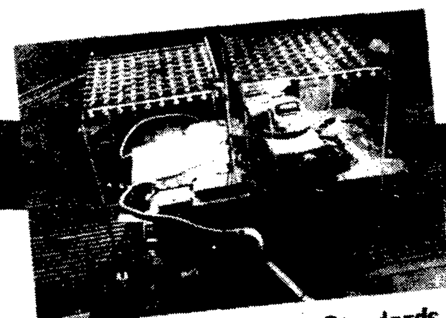
STEP 1.- Instrument Vehicle to Obtain Duty Cycle Data.



STEP 2.- Operate Vehicle Over Test Courses To Obtain Data On Tapes.

STEP 3.- Develop Laboratory Tapes From Duty Cycle Tapes for Controlling Laboratory Simulation Tests.

STEP 4.- Perform Laboratory Simulation Tests of Vehicle Systems and Sub-Systems or Components.



STEP 5.- Develop Test Standards and Specifications.

BENEFITS OF TEST METHODOLOGY DEVELOPMENT

- IMPROVED RELIABILITY, AVAILABILITY, MAINTAINABILITY, AND DURABILITY OF FUTURE AND FIELDED VEHICLES.
- COST SAVINGS AND SHORTENING OF THE MATERIAL ACQUISITION PROCESS.
- DEFINITIVE AND REALISTIC SPECIFICATIONS FOR PROCUREMENT OF VEHICLE COMPONENTS
- GENERATION OF DUTY CYCLE DATA BANKS FOR DESIGN APPLICATION AND SOLUTION OF FIELDED VEHICLE PROBLEMS.

COMPUTER MODEL OF TRACK AND SUSPENSION
CONSISTING OF TRACK AND SUSPENSION
PROPULSION AND MOBILITY MODELS

Presented By

W. MERKLINGHAUS
FRG

Vehicle Ride Computer Modell

For almost 3 years we work on a computer program for simulating the oszillations of the hull and of the bogies.

Our program is applied to

1) Suspension characteristics optimization (optimization of spring and damping rate)

2) ride comfort

3) computation of the thermal load of the damping elements

4) determining of maximum speed under given conditions (maximum acceleration at driver's seat)

5) motion of the hull (with respect to the stabilization of the weapon)

6) comparison of different projected vehicles to find the best layout

The first program used a simple plane model with only two degrees of freedom for the hull and one additional degree for each bogie. The spring and damper characteristics may be arbitrary non linear.

The programming languages used are FORTRAN and MIMIC (for CDC 6500). In most cases MIMIC-programs take less time in programming, but FORTRAN programs take not so much computing time. Therfore we prefer MIMIC only, if we have to alter the program very often. In the future we shall also have a test with the programming language CSSL on our CDC 6500.

The integration algorithm employed is a fourth-order Runge-Kutta method. We also tested some other methods, but we were not successful in finding an algorithm, which would reduce the computing time without increasing the numerical error. This point seems to be a real difficulty, for at present the computing time in many cases is to long.

By comparing the computed results with measured data, we did not find sufficient accuracy. Therefore we decided to improve our model with regard to the following aspects:

1) Consideration of the forces of the track on the bogies

2) Extension to three degrees of freedom for the hull (oscillation in the direction of motion of the vehicle)

By this modification we had a better conformity with measured data.

We have several kinds of terrains in our computer model

1) single obstacles of different forms (ramps, steps, ditches)

2) artificial terrains as APG proving ground or sine-wave road

3) measured terrains

4) random profils from a random generator with a given power spectral density and normally distributed amplitudes

The velocity of the vehicle may be given as a constant value or by list.

At present we simulate the vehicles of the new battle tank KPz 3 by running it over steps, a sin-wave road and a measured terrain.

Investigation of Tracks

1. Description of Problems

Basing on the fact, that the tracks which are used at the FRG's tanks have been found out as rather reliable the demand was to find out which way was to go to develop a track with higher specific performance, which means especially a track which has less weight versus performance but the same reliability.

To get a uniform basis there was made an investigation of the material stress of the tracks of Leopard I, M 48 and M 41.

For this reason there were done breaking tests (figure 1) as you see here as an example with track of Leopard one where as the whole track assingle parts of the track were disrupted.

2. Tension Analysis

As another step there was done a stress analysis by the matter of strain gauge, that means strain gauges were fixed on those places which were found out as places of maximum stress by computation, by crackle lacquer and by breaking tests (figure 2).

With an suitable loading equipment there were simulated all essential loads

- the traction
- the transverse force
- the torsion
- the bending versus axle
- the compressive force

Figure 4 shows the measured tensions at chosen measuring points.

3. Fatigue Tests

The dynamic operating loads may cause a limiting of endurance as a matter of state of stress, of the construction, of the load cycles and of corrosive influence.

As the real load cycles for the track are unknown as for as I know, we will try to run measurements on vehicles. But at the other hand we need equivalent values so we started with onestep-fatigue-tests on part of tracks.

Figure 5 shows a woehler-line of pulsation stress of traction for the three track which were tested.

5. Stress Computation

Beside this experimental evaluations which shall bringout basis datas for new design we also are engaged in doing stress analysis with the wellknown method of finite-elements.

By this way there will be changed different parameters to got the optimum of material economy.

Figure 6 points out the influence of the broads of the butt joint of a connector-track to the bending of the bolt. Here it is assumed that the bolt is pivoted in rubber as it is done at the Leopard-track.

It can be pointed out that in the fact of the variations the place of maximum stress may be moved from the middle of the bolt to the position between butt joint and middle part of the track.

The greatest influence to the stress of the bolt is due to its diameter (figure 7). With the variation of this diameter we get quantitative values for the design, which is not evident for the statically indeterminated bearing.

Figure 8 points out the influence of the design of the elements of track. Here we have broken down the track shoe in several parts and connected them with additional bolts. The result is a declining of the maximum tension of the main bolt.

If we got four track shoes at one track-element where at the elements are connected with five connectors we get also a declining of the stress. The maximum strain is also changed by modifying the width of the connectors (figure 9) where at you get the optimum with equally broad connectors.

The very high torsion in the pin is expecially due to the rubbor bearing of the pin in the track shoe. Therefore we thought about getting a stress-reduction by non-elastic supporting points.

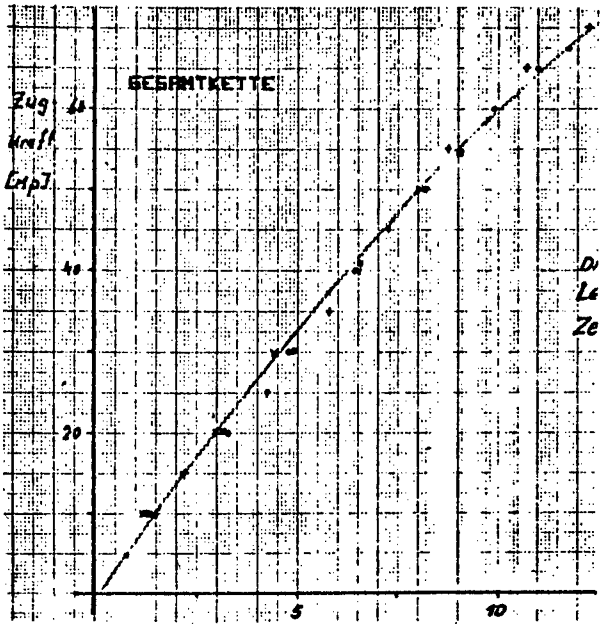
Figure 10 points out the comparsion of the mere rubber bearing with different situated supporting points. It is evidient, that it is possible here by to reduce the maximum bending-torsion.

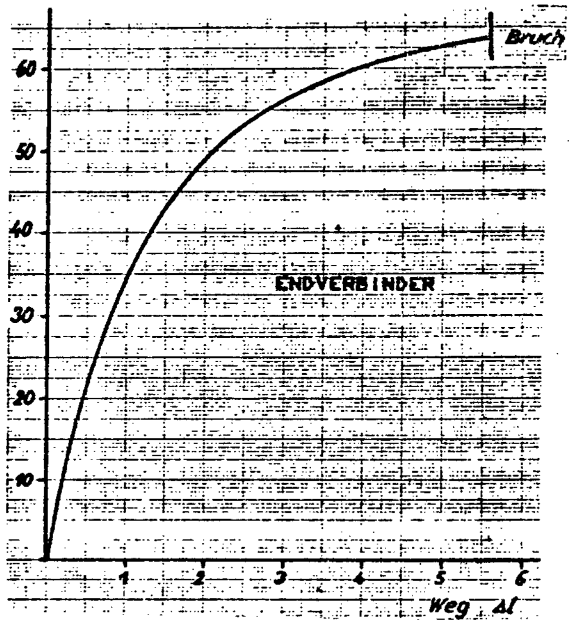
Figure 11 und 12 points out how to reduce the bending-stress at the pin if the sprocket does not handle at the butt joints but at connectors which are situated more to the middle of the track's element. Here is also evaluated the influence of the position of the track element to the sprocket. The stress of the pin will be as more equal as farer the element is situated from the sprocket.

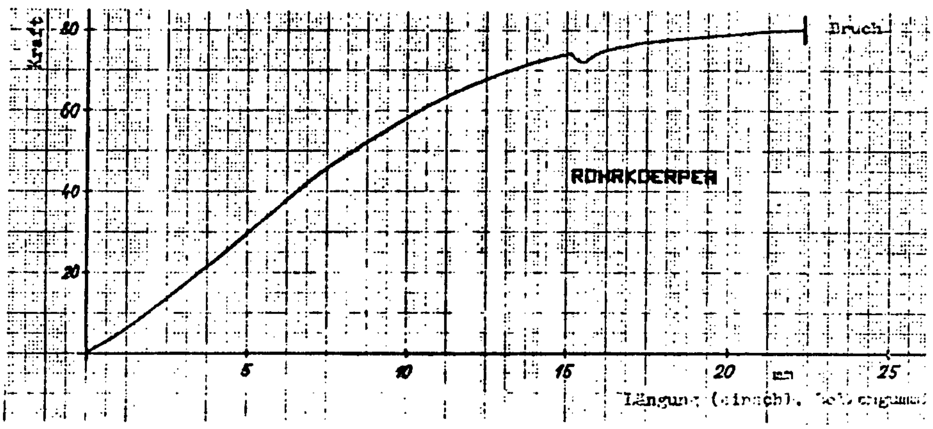
A last example shows in figure 13 and 14 the influence of a cross piece at the track shoe. The rather small influence could be removed by a small thickening of the tube walls where at you got a loss weight never the less.

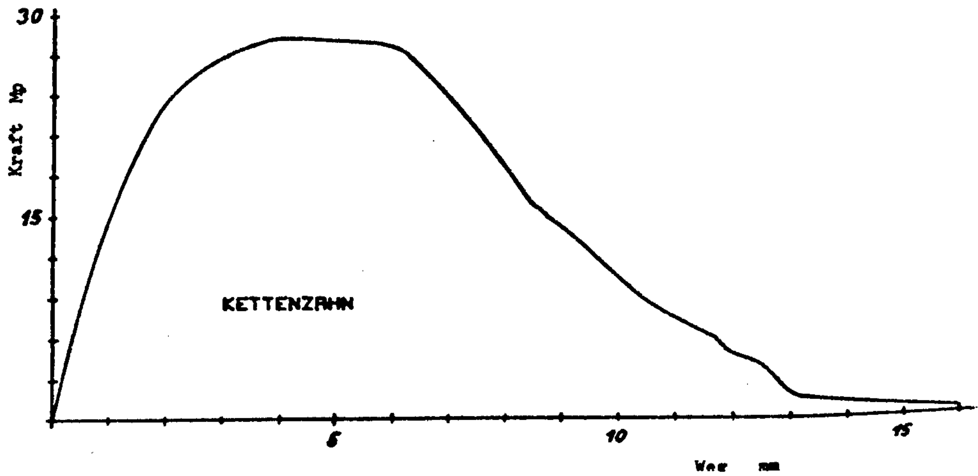
The discussed examples were not a part of a final result, we are working at this problem till jet. But it is pointed out that it is possible to get a higher performance by varying the design in the discussed way. It is pointed out that especially the track with several track shoes in cross direction brings out benefit in reducing the stress. Another benefit is the possibility to get a small and a broad track by adding and taking off addinional track shoes.

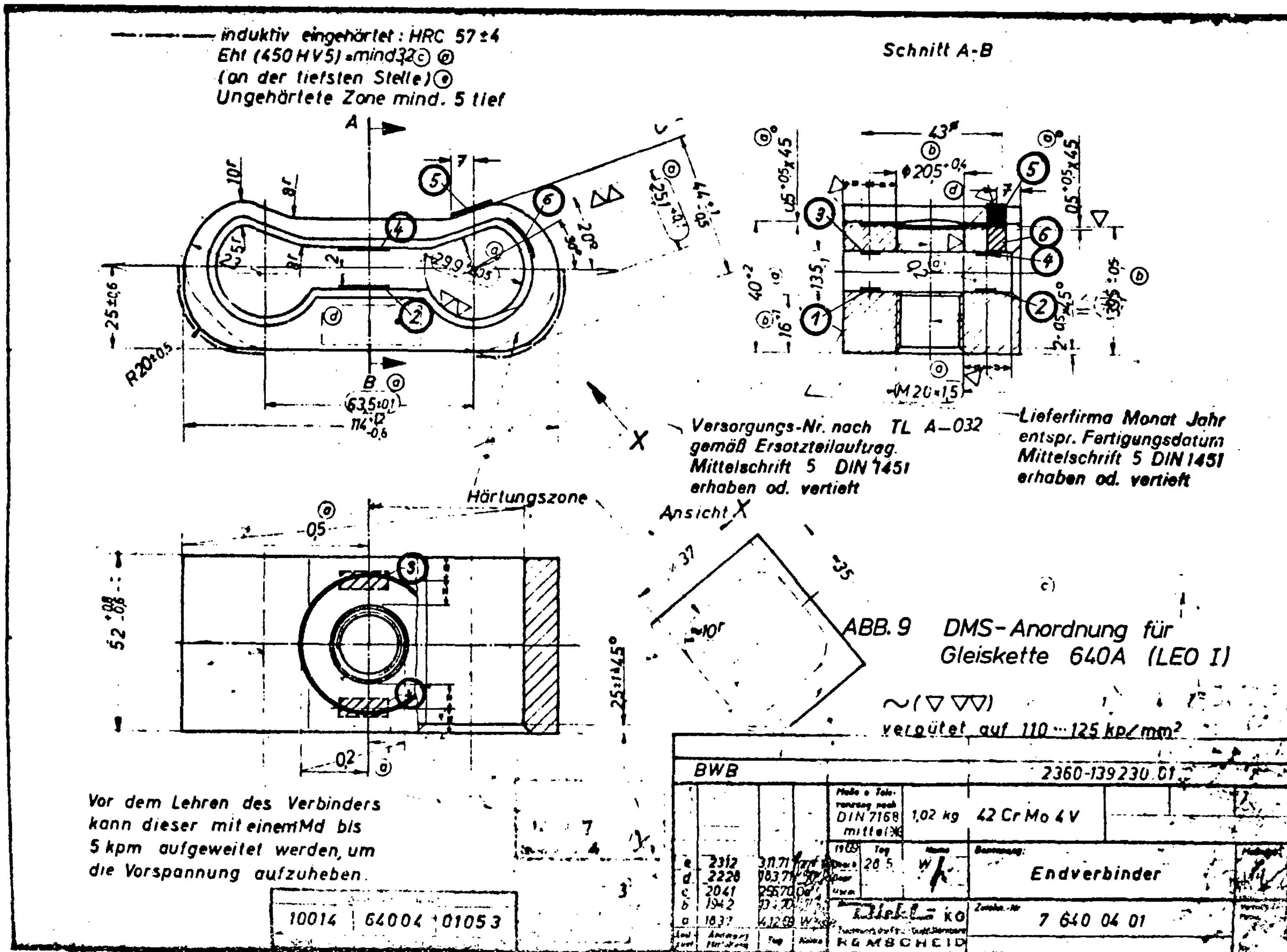
ZERREISSVERSUCH AN KETTE 640 A










induktiv eingehärtet: HRC 57±4
Eht (450 HV5) = mind 3,2 ⓒ ⓐ
(an der tiefsten Stelle) ⓐ
Ungehärtete Zone mind. 5 tief
Schnitt A-B
Versorgungs-Nr. nach TL A-032
gemäß Ersatzteilauftrag.
Mittelschrift 5 DIN 1451
erhaben od. vertieft
Lieferfirma Monat Jahr
entspr. Fertigungsdatum
Mittelschrift 5 DIN 1451
erhaben od. vertieft
Härtungszone
Ansicht X
ABB. 9 DMS-Anordnung für
Gleiskette 640A (LEO I)
vergütet auf 110 ··· 125 kp/mm²
Vor dem Lehren des Verbinders
kann dieser mit einem Md bis
5 kpm aufgeweitet werden, um
die Vorspannung aufzuheben.
10014 64004 01053
BWB
2360-139230 01
DIN 7168
mittel
1,02 kg
42 Cr Mo 4 V
Endverbinder
7 640 04 01

Spannungen $[Kp/mm^2]$

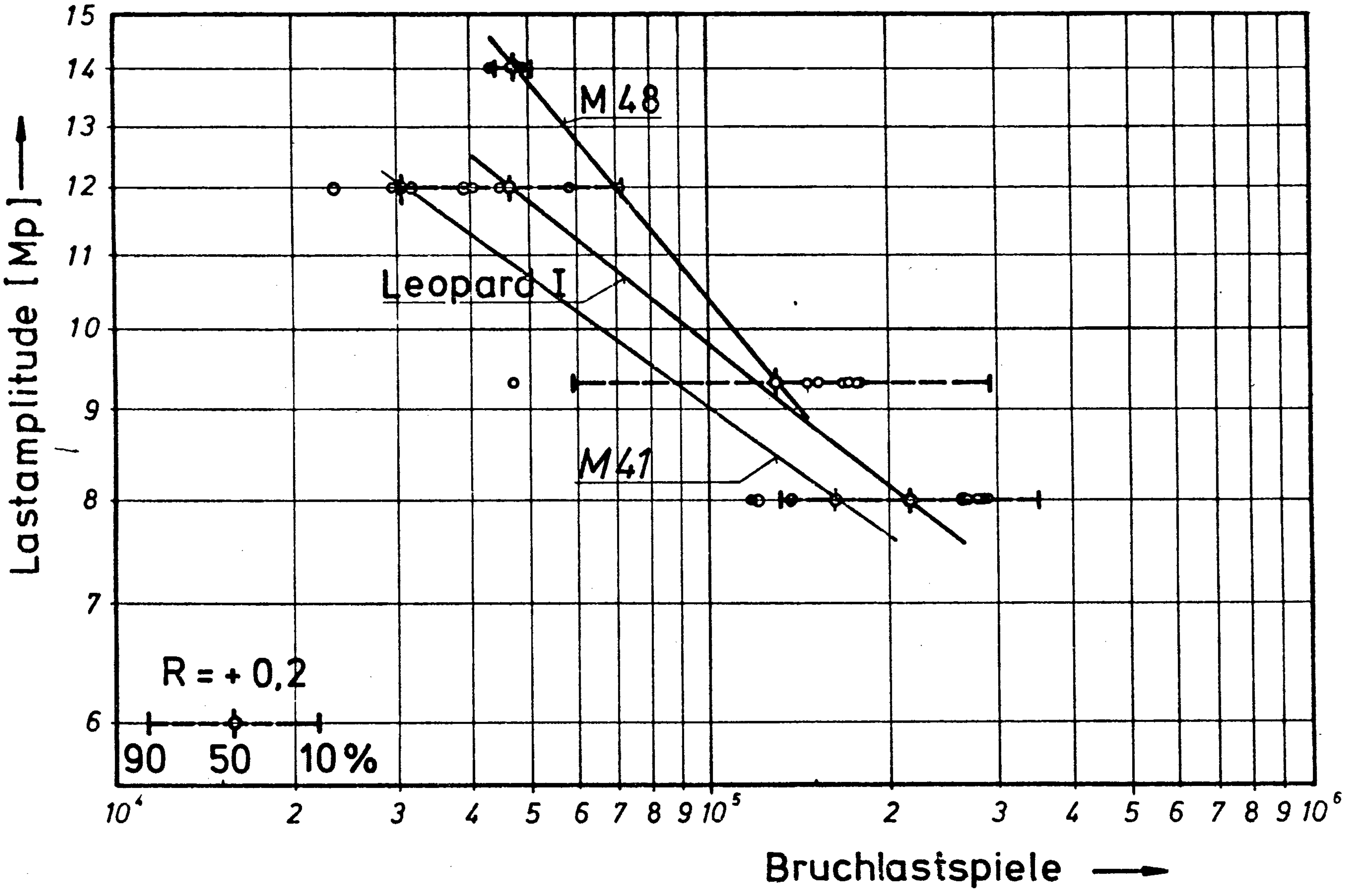
	Leo I			M 48			M 41		
Lastfallstelle	I	II	III	I	II	III	I*	II	III
Längszug 35 Mp	+ 71	+ 56	+ 8	+95	+ 22	+ 32	+ 20	+ 51	- 4
Längszug 15 Mp	+ 31	+ 19	+ 2	+34	+ 9	+ 15	+ 8	+ 12	- 1
Querkraft 6 Mp mit Längszug 15 Mp	+ 45	+ 30	- 13	-51	- 12	- 23	+ 11	+ 23	- 41
Torsion $4,3^{o}$ mit Längszug 15 Mp	+ 31	+ 22	+ 0,4	+ 8	+17	- 6	+ 13	- 0,5	- 0,5

I - Endverbinder II - Rohrkörper III - Kettenzahn

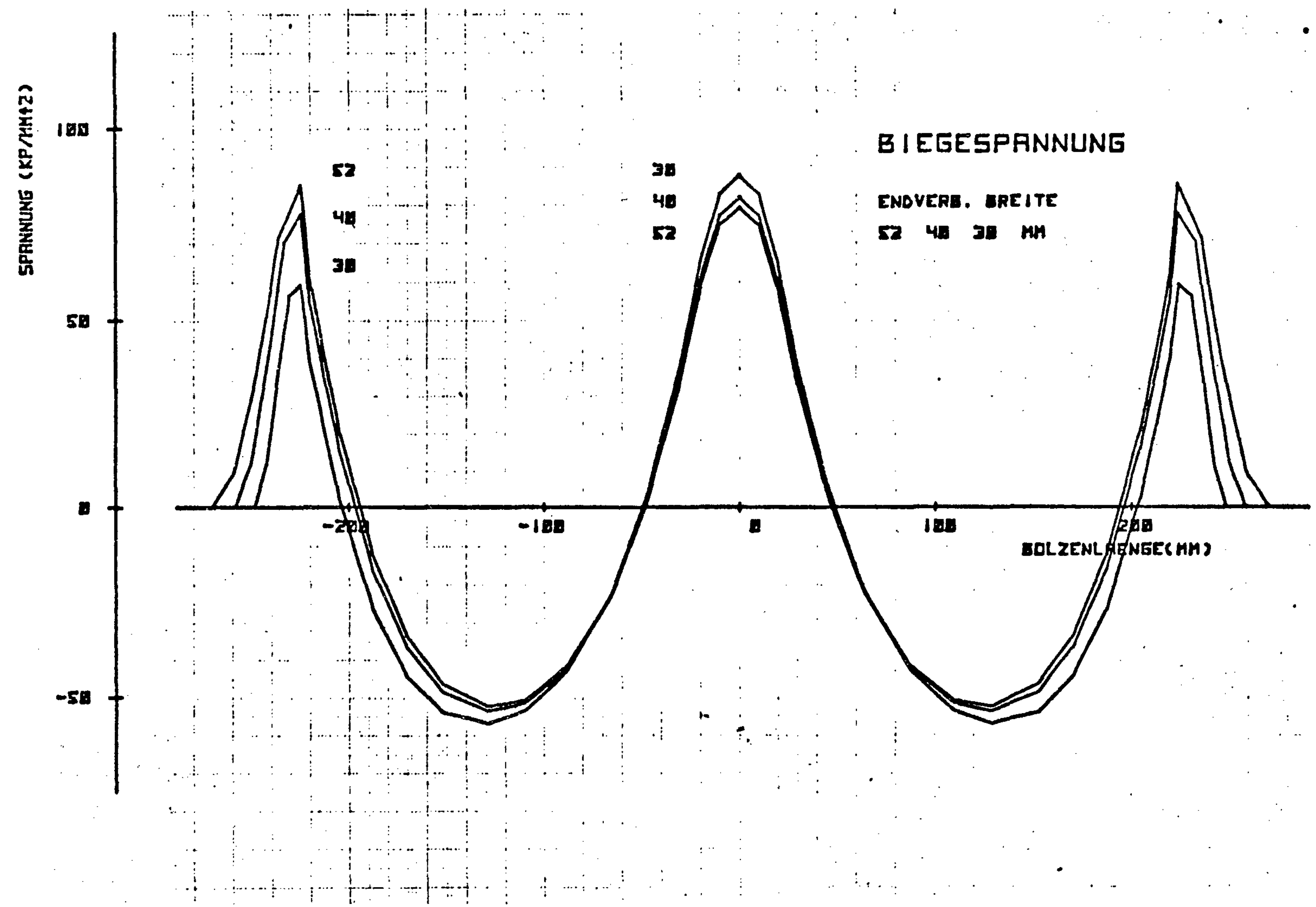
II* - Krafteinleitungsbereich

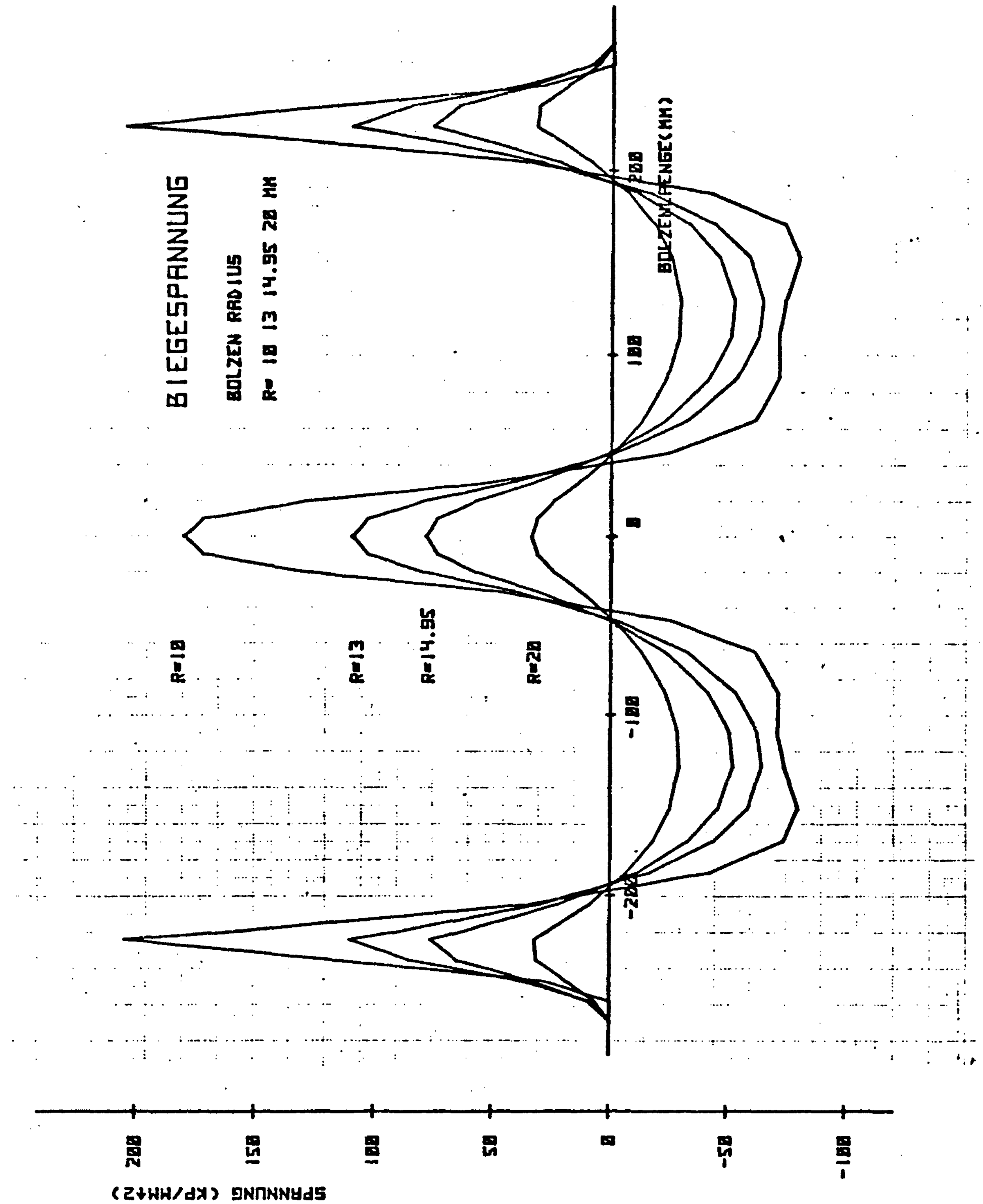
Spannungsvergleich bei stat. Last

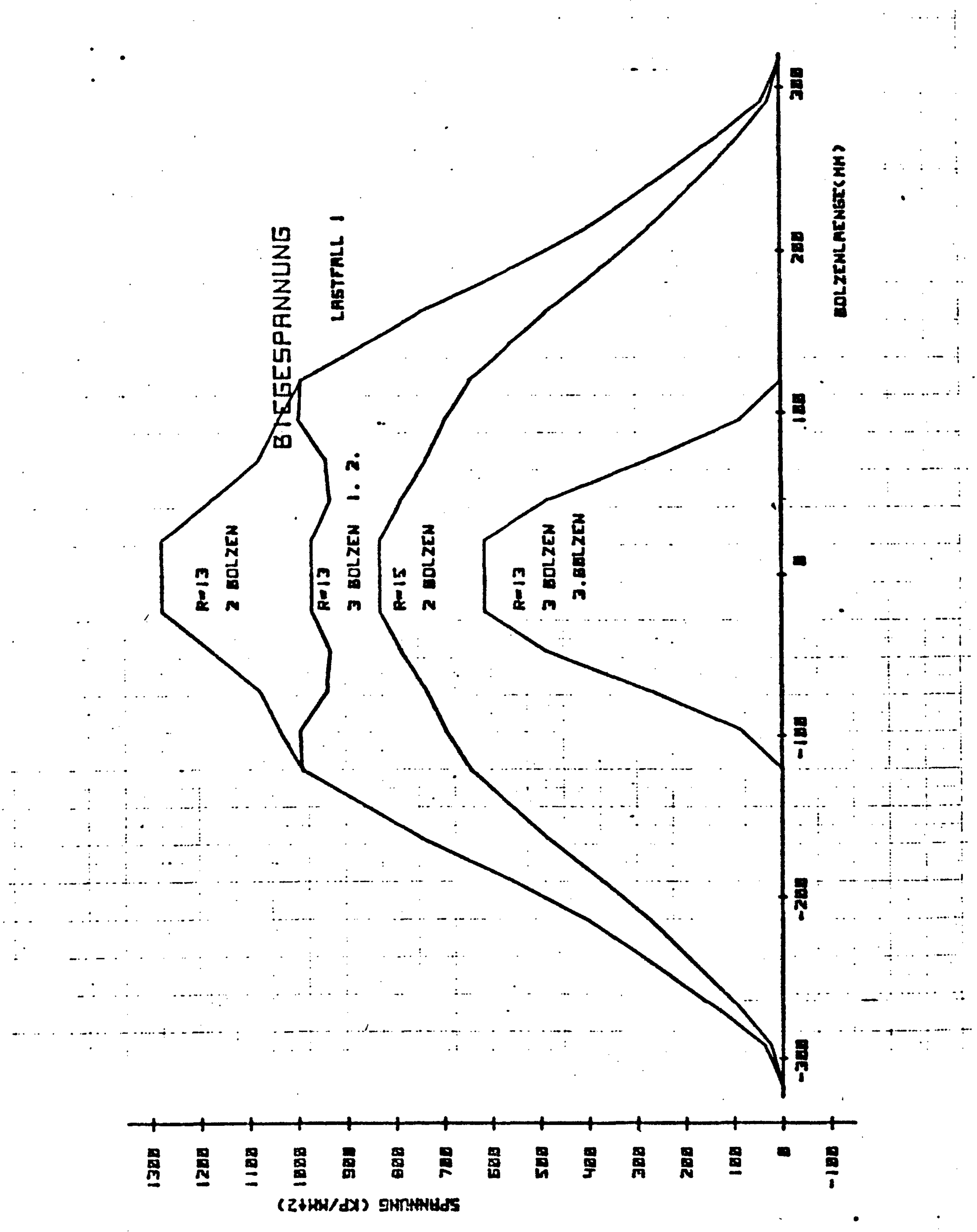
Tabelle 2

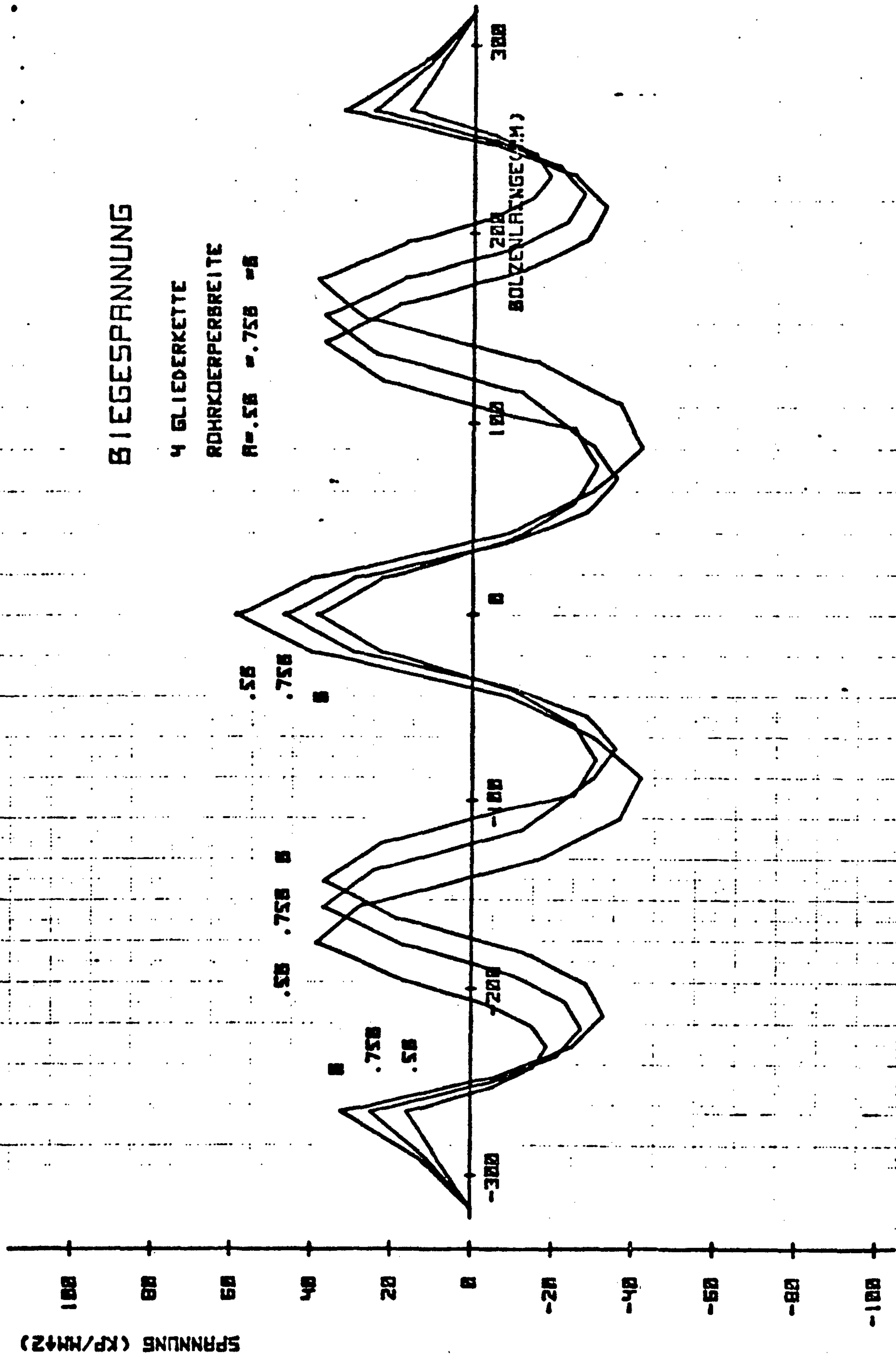


IABG	WÖHLERVERSUCHE AN PANZERKETTEN

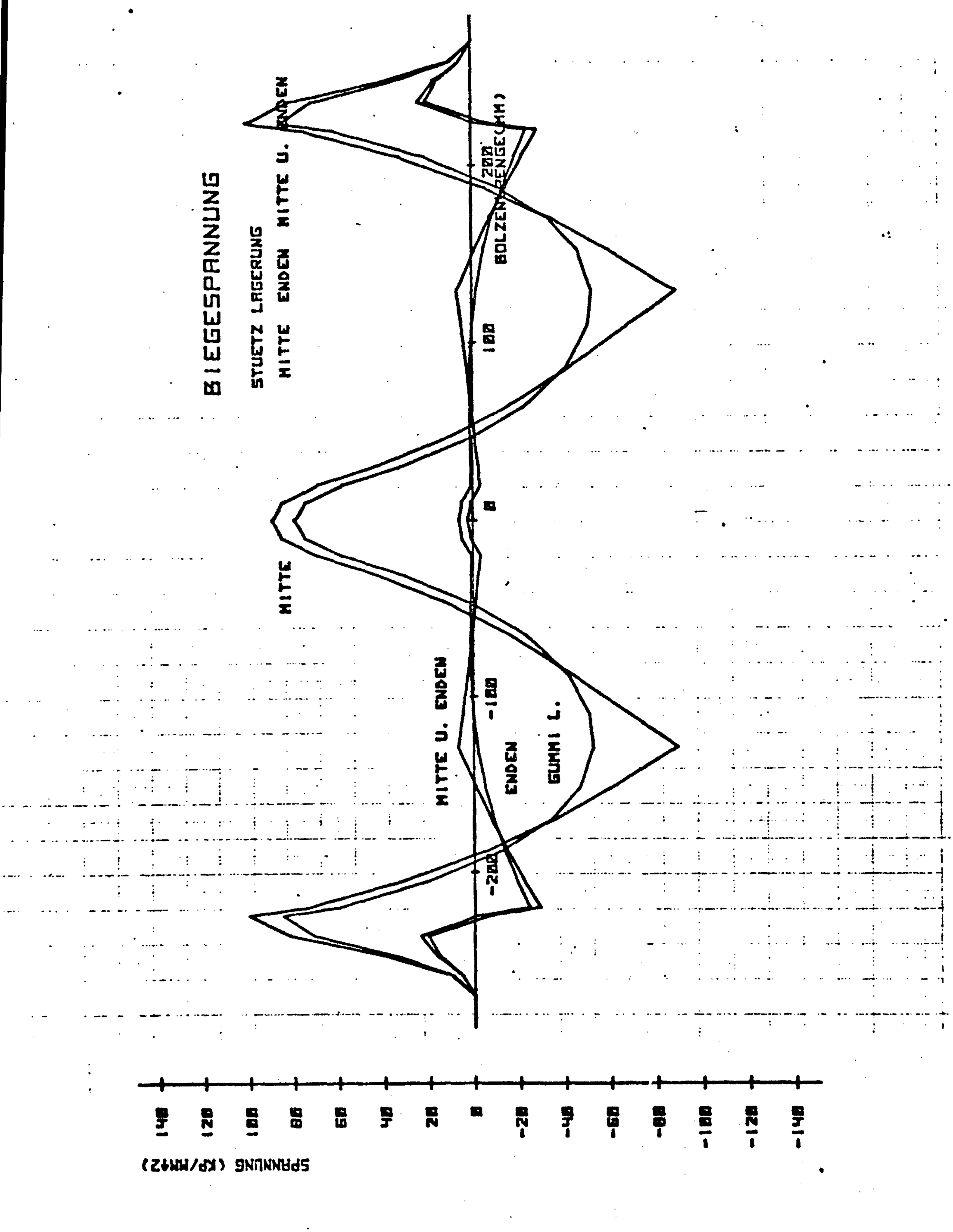

BIEGESPANNUNG
ENDVERB. BREITE
52 40 30 MM
SPANNUNG (KP/MM↑2)
100
50
0
-50
-200
-100
0
100
200
BOLZENLAENGE(MM)
52
40
30
30
40
52

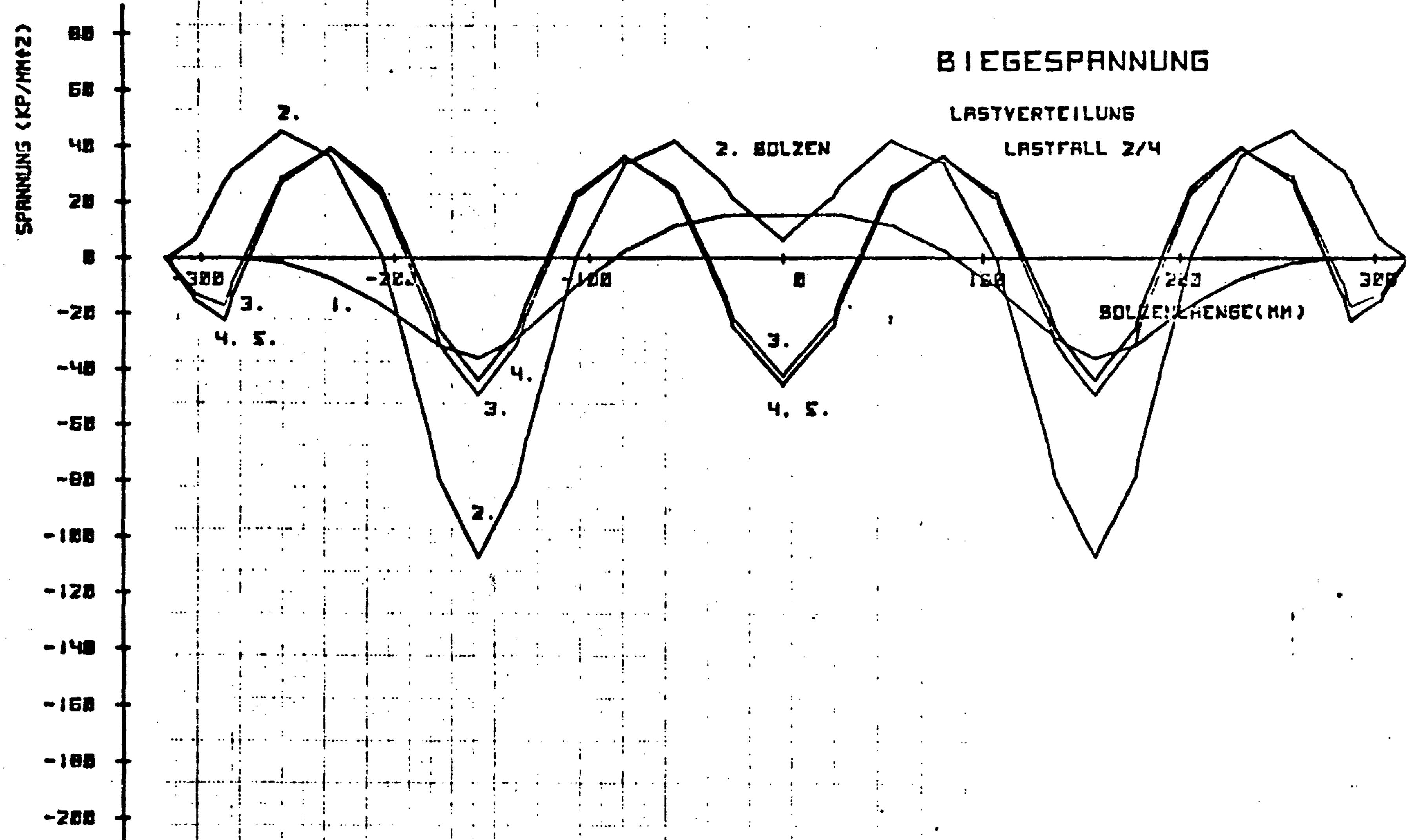

BIEGESPANNUNG
BOLZEN RADIUS
R= 10 13 14.95 20 MM
R=10
R=13
R=14.95
R=20
-200
-100
0
100
200
BOLZENLAENGE(MM)
200
150
100
50
0
-50
-100
SPANNUNG (KP/MM↑2)

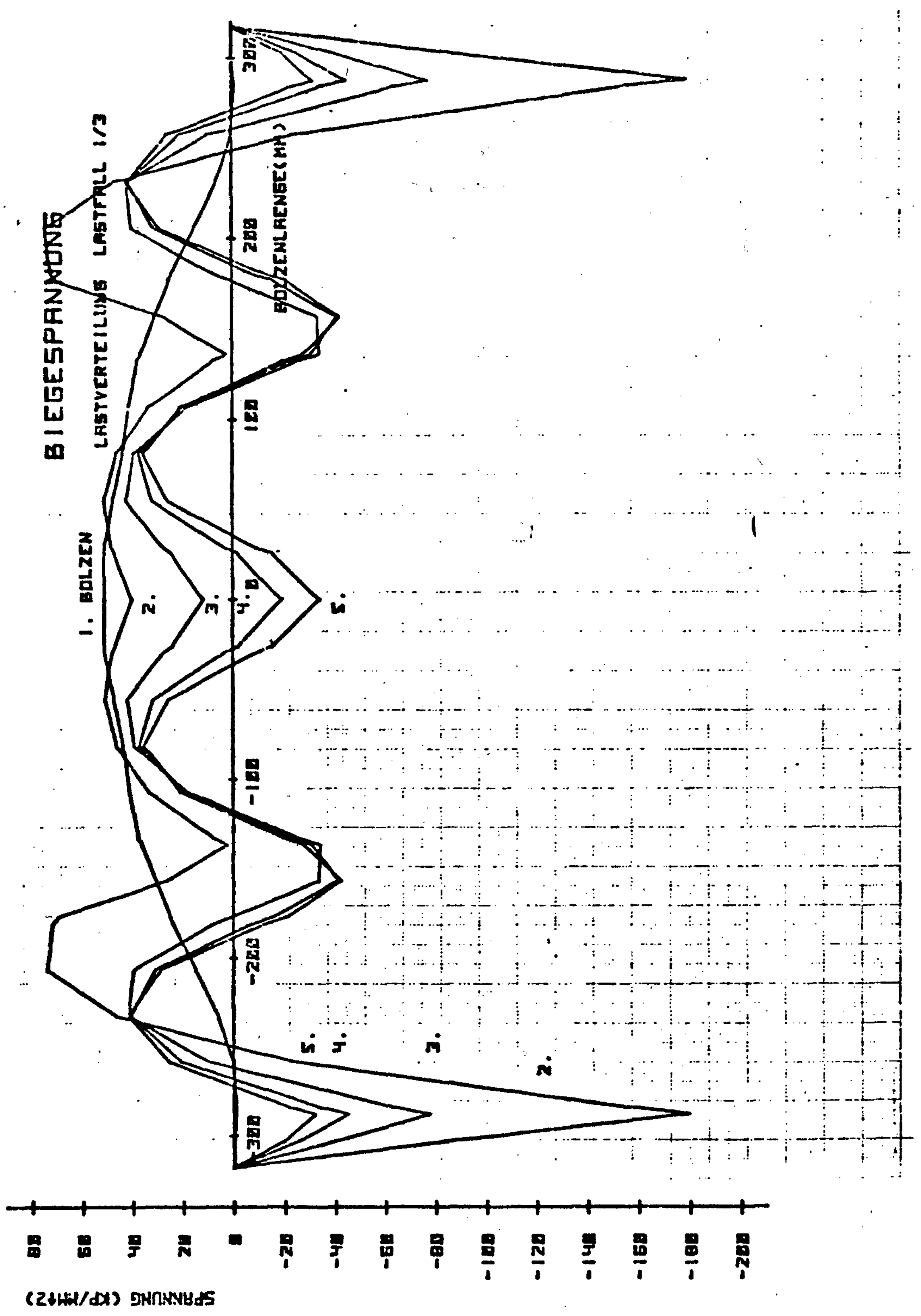

BIEGESPANNUNG
LASTFALL 1
R=13
2 BOLZEN
R=13
3 BOLZEN 1. 2.
R=15
2 BOLZEN
R=13
3 BOLZEN
3.BOLZEN
SPANNUNG (KP/MM↑2)
1300
1200
1100
1000
900
800
700
600
500
400
300
200
100
0
-100
BOLZENLAENGE(MM)
-300
-200
-100
0
100
200
300

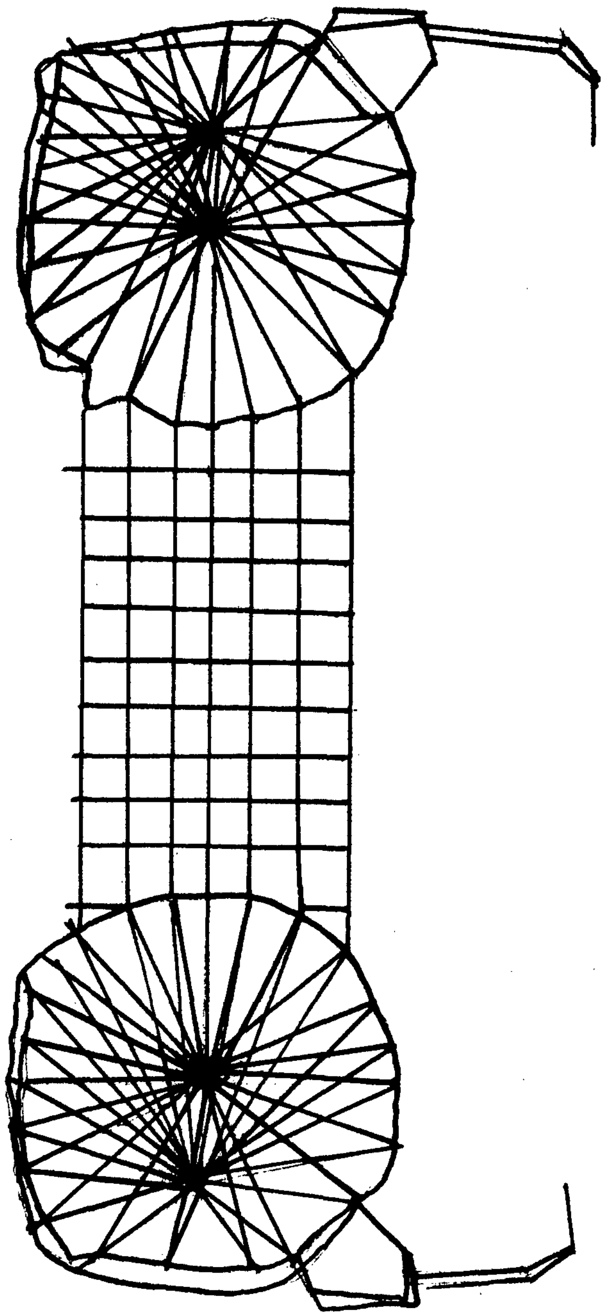


BIEGESPANNUNG
4 GLIEDERKETTE
ROHRKOERPERBREITE
A=.5B =.75B =B
BOLZENLAENGE(MM)
SPANNUNG (KP/MM↑2)


BIEGESPANNUNG
STUETZ LAGERUNG
MITTE ENDEN MITTE U. ENDEN
MITTE
MITTE U. ENDEN
ENDEN
GUMMI L.
BOLZENLAENGE(MM)
200
100
0
-100
-200
SPANNUNG (KP/MM↑2)
140
120
100
80
60
40
20
0
-20
-40
-60
-80
-100
-120
-140


BIEGESPANNUNG
LASTVERTEILUNG
LASTFALL 2/4
2. BOLZEN
SPANNUNG (KP/MM↑2)
BOLZENLAENGE(MM)
80
60
40
20
0
-20
-40
-60
-80
-100
-120
-140
-160
-180
-200
-300
-200
-100
0
100
200
300
1.
2.
3.
4.
4. 5.


BIEGESPANNUNG
LASTVERTEILUNG LASTFALL 1/3
1. BOLZEN
2.
3.
4.
5.
BOLZENLAENGE(MM)
SPANNUNG (KP/MM**2)

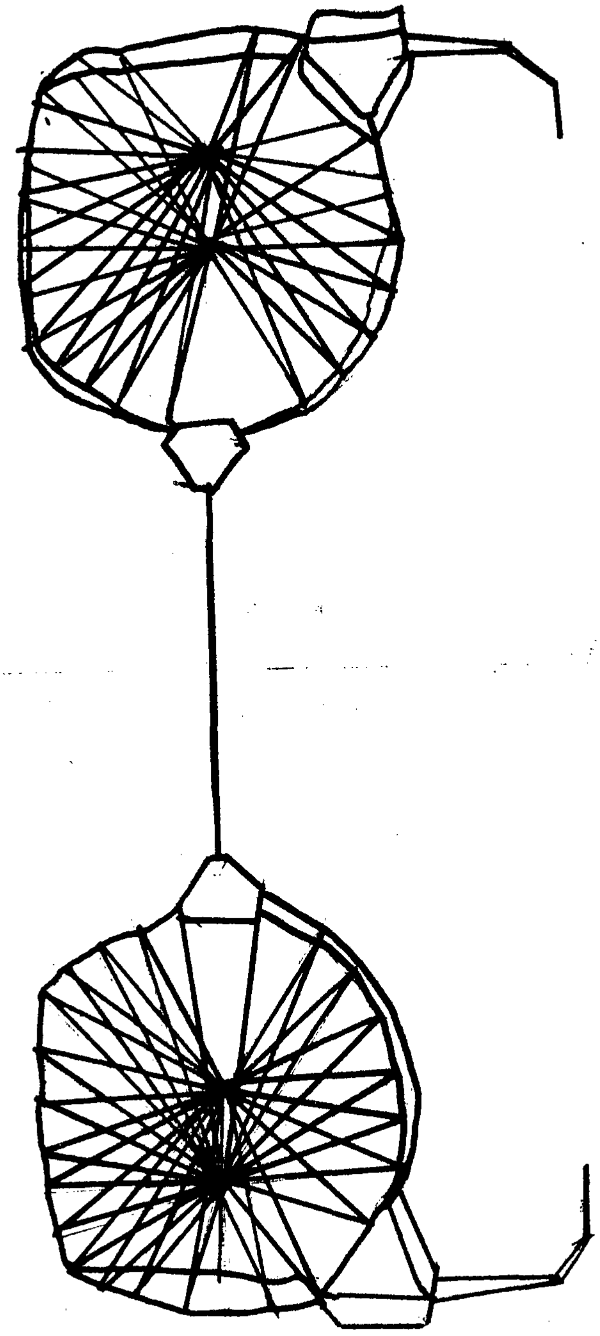


ETTE 5 ROHRABSCHNITT L = 25. MITTENSTEIFE

QUADZ - IDEALISIERUNG MIT MPC

ROHRKOERPER LEO I

STATIC DEFOR. SUSCASE : LOAD :



MODELL 5 ROHRABSCHNITT LAENGE - 25 MM

QUADZ - IDEALISIERUNG

ROHRKOERPER LEO I OHNE MITTENSTEIFE STATIC DEFOR. SUSCASE : LOAD :

ADVANCED OBSTACLE PERFORMANCE MODEL

Presented By

R. JACOBSON
U.S.

PURPOSE: The purpose of the advanced obstacle model is:

VG-1 - to obtain information necessary to calculate the additional tractive effort required to overcome an obstacle.

- to obtain an interference history between the vehicle and the obstacle.

This information is obtained by moving the vehicle over the obstacle in steps and finding its orientation with respect to the obstacle at each step. Knowing the orientation, the clearance between the vehicle and the obstacle can be found and the tangential forces at each wheel can be found.

1. The improvements of this model over previous ones are:

a. This model will accept a vehicle which is made up of two units and will allow articulation in the pitch plane between the two units.

b. The model takes into account the effects of suspension compliance.

c. It can accept obstacles of arbitrary shape.

VG-2 2. To find the orientation of the vehicle with respect to the obstacle, we calculate equations of static equilibrium for the vehicle at each step as it crosses the obstacle. To do this we represent the vehicle as a beam, located at the vehicle upper suspension supports. This beam is supported on springs which represent the suspension units, either independent or bogie type suspensions can be used. The lower end of these springs is located at the wheel hubs. Each suspension support has an initial load which is the weight of the vehicle on that support for the vehicle with no payload. The payload of the vehicle is represented as a concentrated load located at the center of gravity of the actual payload.

3. Using this information, where ℓ_i is the location of each support x_i is the relative displacement of the wheel hub with respect to the body of the vehicle.

K_i is the spring constant of the suspension support in the wheel plane d_1 and d_2 are the distances to the centers of gravity of the payloads delta W1 and delta W2. We can solve equations of static equilibrium for Z_1 and θ, and Z_2 and ϕ, where Z is the vertical displacement of the location of the P1, θ is the angle of each unit. Using these equations we can solve for Z_1 and θ for the first unit of the vehicle, and Z_2 and θ for the second unit, using

$$V = \sum_{i=1}^{N} \frac{Ki}{2} (x_i - Z_1 - (d_1 - \ell_i)\theta) + (\Delta W)Z$$

VG-3 4. To obtain an interference history of the vehicle going over an obstacle. The obstacle is first modified to show the path that the wheel hubs would follow. This is the dashed line shown on the Vu-Graph. The vehicle is moved across the obstacle in steps. At each step the equations of static equilibrium are solved for Z_1, Z_2, θ and ϕ. Knowing these two values locates the vehicle exactly with respect to the obstacle. The clearance between the obstacle and the vehicle is checked point by point for the entire vehicle for this step. The clearance is checked in this manner at each step. Also each time the equations are solved for Z_1,Z_2, θ andϕ, the wheel travel and bogie swing angle are checked to insure that they are not out of their limits. If they have reached their limits the equations are changed to account for these conditions.

5. The tangential force between the vehicle and the obstacle is calculated at each step. This is done by finding the angle of the tire contact patch at each wheel and calculating the component of the total on that wheel in the direction parallel to the obstacle. The value for each wheel is added to obtain a total value for the vehicle.

6. The clearance data is used to determine the ability of the vehicle to physically get over an obstacle. The tangential force data is used in AMC '74 to determine the additional tractive effort required to overcome the obstacle and to determine the speed loss in crossing the obstacle.

7. This concludes my part of this briefing. Are there any questions?

ADVANCED OBSTACLE CROSSING MODEL FOR AMC '74

PURPOSE:

- TO OBTAIN THE ADDITIONAL TRACTIVE EFFORT NECESSARY TO OVERCOME AN OBSTACLE
- TO OBTAIN AN INTERFERENCE HISTORY OF THE VEHICLE CROSSING THE OBSTACLE

$$\left[\sum_{i=1}^{N} K_i\right] Z + \left[\sum_{i=1}^{N} K_i (d - \ell_i)\right] \theta = -\Delta W + \sum_{i=1}^{N} K_i X_i$$

$$\left[\sum_{i=1}^{N} K_i (d-\ell_i)\right] Z + \left[\sum_{i=1}^{N} K_i (d-\ell_i)^2\right] \theta = \sum_{i=1}^{N} K_i (d-\ell_i) X_i$$

ℓ_i = LOCATION OF THE i^{TH} SUPPORT

K_i = SPRING CONSTANT OF THE i^{TH} SUPPORT

X_i = LOWER SUPPORT DEFLECTION

ΔW_1 = WEIGHT OF PAYLOAD FOR FIRST UNIT

d_1 = LOCATION OF THE CG OF THE PAYLOAD ΔW_1

ΔW_2 = WEIGHT OF PAYLOAD FOR SECOND UNIT

d_2 = LOCATION OF THE CG OF THE PAYLOAD ΔW_2

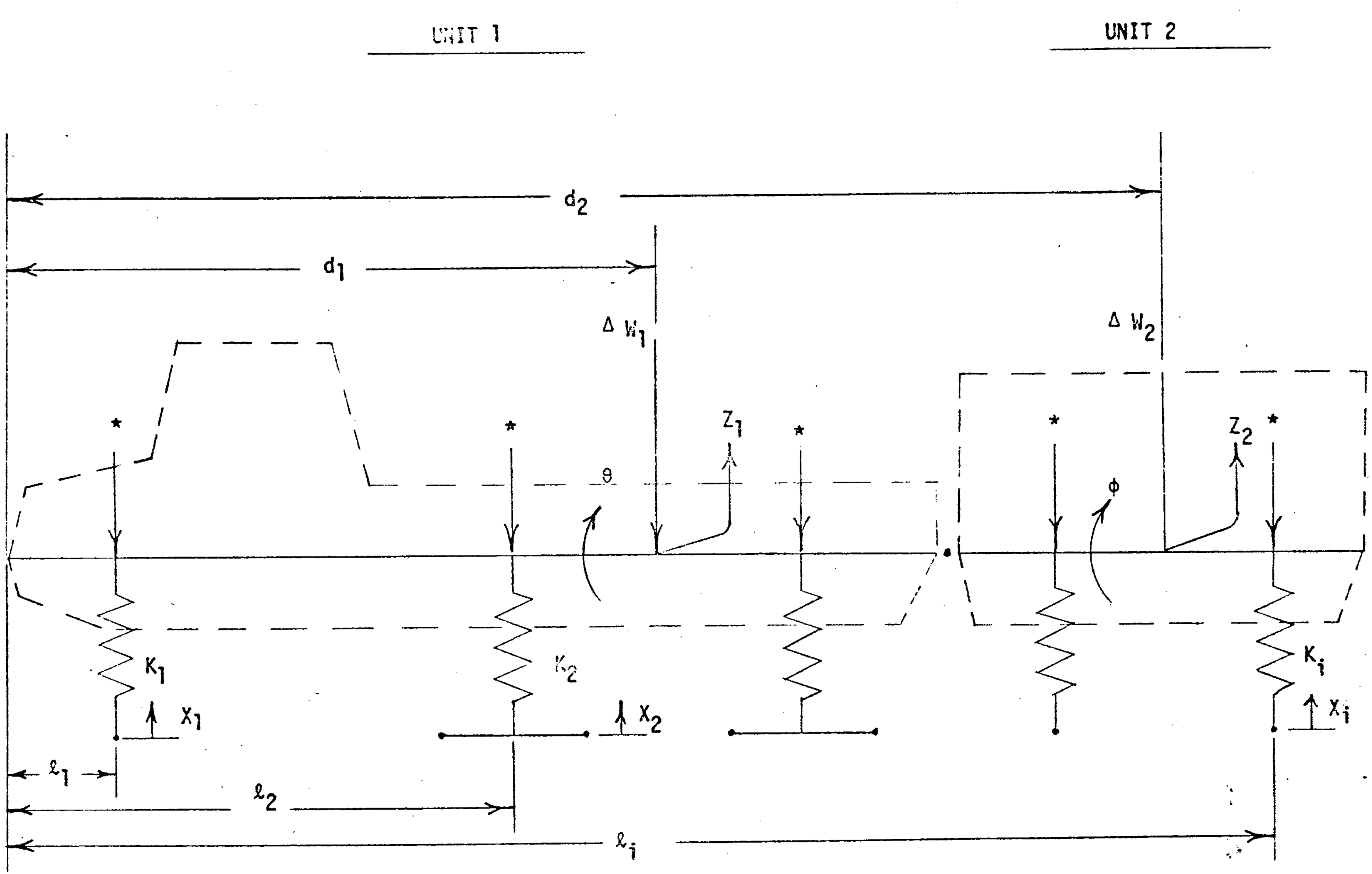
Z_1 = VERTICAL DISPLACEMENT OF THE FIRST UNIT AT THE POINT OF APPLICATION OF THE PAYLOAD ΔW_1

θ = INCLINATION ANGLE OF THE FIRST UNIT

Z_2 = VERTICAL DISPLACEMENT OF THE SECOND UNIT AT THE POINT OF APPLICATION OF THE PAYLOAD ΔW_2

ϕ = INCLINATION ANGLE OF THE SECOND UNIT

VEHICLE REPRESENTATION FOR AMC 74 OBSTACLE MODULE



*Equilibrium load of vehicle with no payload at each support.

APPENDIX