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AD#:	TITLE:
P004 305	A tracked vehicle test plant for the simulation of dynamic operation
P004 306	Proffleinfluss auf Widerstands-und lenkkraft frei rollender reifen (The influence of the tyre tread on the rolling resistance and steering forces on undriven wheels) (including English translation)
P004 307	Basic study on the turning resistance of track
P004 308	Study on steerability of articulated tracked vehicles

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A TRACKED VEHICLE TEST PLANT FOR THE SIMULATION OF DYNAMIC OPERATION

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ABSTRACT

In 1982, a modern test plant was taken into operation at the University of the German Armed Forces in Hamburg, which has very good dynamic properties that make it possible to simulate not only rolling resistance, but also the inertia of vehicle mass and the turning resistance of a tracked vehicle. Herewith, the dynamic loads of real vehicle action on the road and in the terrain can be run on the test stand, true to reality, as it has not been feasible on power test stands employed so far.

The test stand conception is presented and functions are explained. The fundamental principles for performance lay-out of the test stand for tracked vehicles are pointed out, referring specially to the requirements to be met for simulation of acceleration and turning conditions. The applications for testing are discussed, and several versions of control systems are presented for different modes of operation.

INTRODUCTION

In the past, many attempts were made to replace road and terrain testing of the performance of tracked vehicles by simulators. On several test stands in use, steady-state conditions can be realized (e.g. at TACOM, Detroit). However, dynamic testing, especially simulation of inertia forces and turning resistance, is not possible thereon. Dynamometer Trucks for towing and braking vehicles on the road or in the terrain were developed for the Bundeswehr Erprobungsstelle 41 and for Yuma Test Station. But with these tracked vehicles indoor tests are impossible. For indoor testing, several studies were made, for instance of roller-type dynamometers, but none of them was materialized, due to cost and technical reasons.

A most promising concept was proposed in 1976 by the University of the German Armed Forces [1] and developed by the Brown Boveri Company in close cooperation with the user. This test stand installed in Hamburg was started up in 1982. In the meantime, the test stand system has met with much approval. In the USA, it has become known under the designation of "Power and Inertia Simulator" or PAISI.

TEST STAND FACILITIES

Figure 1 shows the test stand with the tank JAGUAR of the GERMAN Armed Forces as the test vehicle. The test stand consists of two power units to which the vehicle sprockets are combined by means of cardan shafts. The tracks have been taken off. The vehicle is operated here by the driver just the same as in actual driving. That means he accelerates, he switches the gears, he brakes and he steers.

The vehicle engine drives, the two test rig power units brake, that is to say they absorb the energy. Under particular driving conditions, the power flux can be vice-versa, which means that the power units will then be driving.

Thus the test rig machines must be able to drive, as well as to brake, and furthermore, they must be able to rotate in both directions. To put this into practice, direct-current generators, so-called dc-dynamometers, are most suitable. They are thyristor controlled.

An essential part of the test rig unit is an adaptive gear box, suitable to adapt torque and speed of the generator to the test vehicle. Between the adaptive gear box and the generator, there is the torque measuring shaft.

The test hall is a sound-proof room with a 12-ton-crane, Figure 2. The vehicle is clamped on a rigid and heavy foundation. The foundation mass is approximately 1600 tons. The necessary cooling air is provided by a high-capacity blower (150 kW), via a scoop. The exhaust gases are collected at the exhaust pipe by a funnel and escape via a metal hose.

Inside the control room, the control desk consists of two identical halves, showing for each of the power units the same configuration of operating and control elements, Figure 3. From here, the condition of operation of the vehicle tested can be predetermined. The control system which is working in closed loop will then automatically provide the desired torque or the desired speed. Additionally, on the control desk, there are all instruments necessary to observe the values to insure a troublefree test run.

TEST STAND CONTROL

The control system is located in the measurement cabinets, Figure 4. For the total of 32 different modes of operation for which the plant has been designed, considerable electronics is required. For some particular cases, an external hybrid computer governs the test rig control system. Speed and torque of the power units can be controlled for each unit or for both units in combination. Thereby, different modes can be realized:

Steady-state Control

Needless to emphasize, steady-state operating conditions can be realized with the test stand. To that purpose, the torque or the speed must be kept at a constant level.

$$n_{desired} = \text{const.} \qquad (1\text{ a})$$

$$M_{desired} = \text{const.} \qquad (1\text{ b})$$

These modes are needed to measure vehicle drive characteristics. Of the two possibilities, the speed control is mostly preferrable, because it allows a more stable setting of operational points than a torque control does: In view of the flat curve of the combustion engine's torque characteristics, difficulties may arise when trying to maintain with torque control a determined load point at a constant speed.

Programmed Control

Alternatively, control can be based on a desired value being variable with the course of time:

$$n_{desired} = n(t) \qquad (2\ a)$$

$$M_{desired} = M(t) \qquad (2\ b)$$

Herewith, the locomotion and load programs can be assessed, and that is where the advantages of the test stand become evident.

Motion Resistance Simulation

An even higher requirement can be met by simulating the motion resistances, and that in a way to obtain the loads as in real locomotion, dependent on any travelling condition.

The total resistance can be expressed by a function as follows:

$$T = a_0' + a_1' \cdot v + a_2' \cdot v^2 + b' \cdot \frac{dv}{dt} \pm C\,(R,\ v) \qquad (3)$$

Therein ($a_0' + a_1' \cdot v + a_2' \cdot v^2$) represent rolling resistance, slope resistance and air drag, whereas b depends on the mass inertia, and C refers to the turning resistance (v means driving speed, R is the radius of curvature).

The total resistance is split to the two sprockets. Thus the torques at the sprockets are

$$\begin{aligned} M_I &= a_0 + a_1 \cdot n_I + a_2 \cdot n_I^2 + b \cdot \frac{dn_I}{dt} + C\,(R,\ n_I,\ n_{II}) \\ M_{II} &= a_0 + a_1 \cdot n_{II} + a_2 \cdot n_{II}^2 + b \cdot \frac{dn_{II}}{dt} - C\,(R,\ n_I,\ n_{II}) \end{aligned} \qquad (4)$$

(n is the rotational speed at the sprockets.)

It should be noted that the sign of + or - before C is opposite at the two sprockets. Simulation of rolling resistance, air drag, and slope resistance on vehicle is well known to be general state of the art. However, simulation of vehicle inertia as practised in the way reported here, and above all simulation of the turning resistance of tracked vehicles, has not been realized so far, but is single in the test stand presented here. Therefore, some considerations must be added as to simulation of the acceleration resistance as well as of the turning resistance.

SIMULATION OF TURNING RESISTANCE

As is known, tracked vehicles are steered by a higher speed at the outer track and a lower speed at the inner track, viewed from the center of curvature. Hereby, a turning resistance arises - a yawing torque - which must be overcome by an additional driving force at the outer track and by a braking force at the inner track. When simulating a curve on the test rig, the two sprockets must accordingly take up different torques at different speeds. To give an example, Figure 5 shows the torque taken up by each sprocket, depending on the rolling speed at various values of the radius of curvature. This diagram was calculated for a 43-tons tank (engine performance 610 kW, superpositioning-steering-gear box) on the road. The rolling resistance was considered to be independent from driving speed, the air drag was neglected. For calculation of the turning resistance, a relatively simple model [2] was used.

As can be seen, in circular motion with growing speed the torque at the outer sprocket increases and reaches a multiple value as compared to the torque in straight-line motion. The speed of the vehicle on a radius, however, can increase no further than to a point where the performance limit of the tank drive will be reached.

As the torque is increased at the outer sprocket, a reduced torque will arise at the inner sprocket, which will become negative when the radius drops. In that case, the inner sprocket has a braking effect.

Those torques - which are provided by the tank drive - must, in case of simulating a curve on the test stand, be counteracted by the test rig unit, as is indicated in equation (4) by the term of $\pm$ C.

In the easiest case, the turning resistance can be set manually at the potentiometer of the motion resistance simulator. This is, however, not satisfactory in a driving simulation. A better possibility is the computer input of the turning resistance as a diagram of characteristics (e.g. Figure 5). This requires the availability of measured or calculated characteristics which are, however, valid only for determined vehicle data and fixed road or terrain conditions. It has, therefore, been envisaged for test stand simulation to establish the turning resistance by the aid of an analytical model of general validity, e.g. [3], [4], in on-line operation.

SIMULATION OF INERTIA OF VEHICLE MASS

In the case of battle-tank test stands, the inertia of the vehicle mass has not yet been simulated. It could for instance be realized by flywheels. Due to the substantial tank mass, however, they would have to be very large. It would then probably become necessary to provide a flywheel set with removable rotary masses to allow an adaptation by steps to the tank mass in question. But this would be quite expensive and would involve problems in simulating steering conditions. Due to changes of sprocket speed during steering, inertia forces would be induced which do not occur in reality.

Therefore, on the test stand to be considered, the mass inertia of the tank is simulated not by mechanical flywheel masses, but by the electrical generators. This involves the need of an equivalent power to be available at the test rig. This power requirement is reduced by the fact that the test rig unit has a certain basic inertia in terms of its mechanical inertia Θ_{PU} so that only the difference up to total vehicle inertia must be supplied as an electrical supplementary torque M_s on each power unit:

$$M_s = [\Theta_v - \Theta_{PU}] \cdot \frac{2\pi}{60} \cdot \frac{dn}{dt} \qquad (5)$$

where Θ_v Inertia of the vehicle simulated (50 %)

Θ_{PU} Inertia of a power unit related to the interface between test rig and unit under test

Equation (5) indicates that generation of the supplementary torque M_s involves a differentiation of the rotational speed signal n. However, in this procedure, the possible accuracy is not too good. Furthermore, feeding back the signal in a closed loop control will result in problems with regard to stability. Consequently, the control circuit is sensitive to oscillations. A certain damping could be helpful, but this would be unfavourable for obtaining a fast dynamic response, as it is necessary to simulate mass inertia.

TEST STAND DYNAMICS

The dynamic behaviour of the test stand includes the

- dynamic response to desired changes of torque or rotational speed
- the mechanical and electrical oscillations of the test stand system, which are, of course, not desirable.

When optimizing both items, problems will arise because there is a certain conflict of goals. However, the realisation of a fast control at fairly low oscillations has become possible by the development of active-damping electrical networks [5] that were harmonized with the mechanics of the test stand.

Figure 6 shows an example for the dynamic quality achieved. It can be seen that the torque follows the desired value quite well, when the desired torque increases from 30 to 50 % of its maximum within 50 ms. In the case of increasing the desired value within 20 ms, the actual value is overshooting to a certain degree. A slower increase can be followed more accurately. Considering real acceleration conditions, the behaviour demonstrated is judged to be sufficient.

Analyzing the oscillations by considering the transfer function, it could be found that the control system is capable to control the test stand up to 10 Hz. Torsional oscillations of the test stand are kept within allowable limits by adaption measures at the control system and at the mechanical system.

TORQUE MEASUREMENT

The accuracy of measurement is essential not only for the precision of the result itself but also for the test stand control in a closed circuit. While for the speed measurement at the two power units rotational speed generators are used, which do not involve any problem, the measurement of the torques is carried out be means of torque measuring shafts (Hottinger F 30 TN 10 kNm) arranged between dc-generator and adaption gear box.

Insertion at that place has the advantage that, when supposing equal test powers, the torques will be of the same order of magnitude for all test stand versions. (The test stand has been designed as a system of modules, allowing by various combinations of the testing components different additional test stand versions, such as test stands for engine, gear box, torque converter, vehicle brakes). It is, therefore, not necessary to exchange the measuring shaft for adapting the measuring range to the requirements of each case. One accepts, however, that the losses of the test stand gear box must be eliminated by a compensating circuit. There is no problem as the losses show regularity, Figure 7. Thus an accuracy of the torque measurement of at least 0,5 %, referred to the final value, will be achieved, Figure 8.

PERFORMANCE LAY-OUT

The performance requirement of the tank test stand results from the power of the vehicle drive train to be tested. It must be considered that the engine power is split to the two sprockets. Figure 9 shows torque versus speed characteristics at each sprocket, depending on the gear box steps, for a tank LEOPARD I (engine power 610 kW). Gearing losses (≈ 20 %) are hereby neglected, and the increase of torque due to the hydrodynamic torque converter is not recorded.

Each of the two test stand power units has to counteract the power at the sprockets. Therefore, the characteristics of a power unit must cover the torque and speed range. In the present case, this is fulfilled by dc-generators of a nominal power of 400 kW each, in combination with the test stand gear boxes. Gear box step A 1 allows high torque, gear box step A 2 high speed.

The torque versus speed characteristics considered so far are true for even power split to the sprockets. However, in a curve, the load distribution will become asymmetrical. As it was indicated in Figure 5, the torque at the outer sprocket, viewed from the center of curvature, can increase to high values and exceed the power unit margin.

For this reason, simulation of the turning resistance of the vehicle LEOPARD I would need 600 kW at each sprocket. Due to the fact that the turning resistance will occur for short periods only, it is not necessary to install dc-generators for 600 kW, because the 400 kw units may be overloaded to 600 kW for a limited time. The decisive limiting factor is the temperature of the armature winding, which then must be measured and observed, Figure 10.

Overloading makes also possible an increase of the driving torque by means of the torque converter, because its action is of short time as well. The same applies to braking of the vehicle. Therefore, braking tests are also possible on the test stand, up to a power of 600 kW at each sprocket.

The considerations above should line out that for a new test stand for tracked vehicles, the maximum engine power is not the absolute dimensioning factor. There shall be emphasized that, in spite of substantial gearing losses (≈ 20 %), the peak power transmitted at the outer sprocket in a curve can exceed even the drive performance gross value. At this sprocket, there can be transmitted more than the engine power, which is only possible because at the inside track of the curvature, power is taken in from the ground, and this power is transmitted by the steering gear to the outside of curvature. The test stand units must meet these performance requirements.

The performance rating of any new test rig must be based on the power requirements with the special data of vehicles to be tested and under consideration of the duty conditions to be simulated.

The user must, therefore, define the vehicles to be tested, as well as the terrain conditions and the missions to be simulated, thus enabling the manufacturer of the test rig to take care of an adequate dimensioning of performance. In view of the dynamic control of the test rig, a certain power reserve should be included, which will be required to provide dynamic behaviour of the test stand.

MODES OF OPERATION

The test stand with its multiple control versions allows several possibilities of testing and simulation, which are indicated in the following:

Steady-state Control

The easiest mode of operation is a steady-state test run with constant control. In this version, the operator at the control desk sets the desired values of the rotational speeds at the sprockets by means of potentiometers, whereas the driver provides the load, i.e. the desired torques. The test rig control system regulates the sprocket speeds in a closed loop, Figure 11.

This kind of operation is suitable for reaching step-by-step various operational conditions, in order to measure in steady state the data of torque, performance, fuel consumption, exhaust gas, noise emission, and so on. In this way, the characteristic curves can be developed. Although this is possible with conventional test stands, the advantage of the power test plant in question is a quicker and more accurate assessment of the operational points due to the control electronics. The diagrams of characteristics which have to be established point by point can thus be rapidly compiled.

Programmed Control

A larger scope of possibilities is offered by speed and torque control based on preset desired values changing as a function of time, e.g. according to the load conditions of a real locomotion. This means a repetition of load programs.

To obtain such a program, the torques at the two sprockets and the rotational speeds must be measured in the field during the fulfillment of the vehicle mission. The signals recorded on a tape represent the desired values which are preset to the test stand control system, Figure 12. The test rig power units are torque-controlled. Contrary to the previously considered cases, the driver is, however, replaced here by a driving actuator, an autopilot, which operates the vehicle. This driving actuator has the function of controlling in closed loop the speed at the two sprockets in such a way that the measured speed from the tape will be maintained.

Of course, with this kind of test rig controlling it is also possible to carry out any desired program of load and speed. For example, a test program for checking the various functions of engine, gear box, and steering gear may be carried out. Therein the selection of the desired points of operation or the passing through of determined courses of load, is controlled automatically. Therefore, the test program will be carried out in a minimum of time resulting in a minimum period of utilization of the test rig and, consequently, in greater testing capacity of the test plant.

For time-consuming endurance tests, as is, for example, the 400 hours NATO test or the 24 hours battle-day, it seems hardly advisable to use tape recordings requiring many hours of run. Here, the use of statistical programs will be more suitable.

Such programs can be obtained by measuring torque and speed to be evaluated as a histogram. To that purpose, the range of speed and the range of torque are subdivided into different classes, Figure 13. Every operational state being characterized by a torque and a rotational speed belongs to a determined cross-square of speed and torque. Every point on the histogram marks that 1 % of the time of operation is attributed to the square in question. From that kind of histogram you will be able to see at once how often and how long the different operational states occur. This depends, of course, on the terrain and on the tactical task. The example given here is the histogram for engine load during a driving test, but, of course, you can record in the same way the histograms at the two sprockets during a military maneuver.

From such histograms obtained from actual field practice, programs for operation of the test rig can be generated, Figure 14. The statistical data are reconverted to values of torques and speeds as a function of time, which are used as desired values for controlling the test rig. The desired values do not correspond to the recorded course exactly; however, from the statistic point of view, they are equivalent to reality. From the programs for different vehicle missions, a representative total program for a long range mission can be derived. It is replayable over an unlimited test period on the test rig, without repetition of uniform phases of load and speed. This mode of operation has not been executed so far, but it is feasible.

Driving Resistance Simulation

In the case of driving resistance simulation, the operator at the control desk is replaced by a driving resistance simulator, Figure 15. The latter computes the desired torques corresponding to the resistances, dependent on speed, acceleration and - if applicable - radius of curvature.

The driver in the vehicle actuates the accelerator, the gear stick, the brake; then the driving resistance as the external load occurs automatically just like under driving conditions on the road. Of course, the factors of a_0, a_1, a_2 and b must be preset at the driving resistance simulator, according to the vehicle and terrain data. These parameters can, however, be changed during the test run, for example to simulate the changing inclination of the road. This can be commanded to the test stand control system from outside by an additional driving program.

As has been discussed before, in an easy case, the turning resistance is controlled by hand. For circular motion, a fixed value is then preset; only the influence of speed and acceleration on driving resistance is simulated. By presetting turning resistance characteristics according to characteristic curves, such as in Figure 5, for instance by means of digital function generators, it is possible to simulate the turning resistance for a given vehicle that operates on a given road or on homogenious terrain.

In a complete driving resistance simulation which comprises in particular the turning resistance as a complex analytical model, the driving resistance simulator of the test rig is overburdened. Then it will be useful to replace the driving resistance simulator by an adequate program in an external computer, Figure 16. For the reason of accuracy, it will then be necessary to work with digital output. Therefore, signal converters D/A and A/D are to be used between the analog control systems of the test stand and the digital computer. The computer treats the speed data and calculates in an on-line process the desired values for the torques at the two sprockets. Those desired values are in general not constant but are subject to permanent changes, and the test rig control system in a closed loop makes sure that on the test rig, the real values of the torques follow the desired values as accurately as possible.

In this mode of operation, the dynamic behavior of locomotion with accelerations and circular path can be simulated. The driver can freely choose the speed. The only restriction is that he must follow a given course providing a fixed dependence of the rolling resistance parameters on the distance travelled. This does not meet the requirements of cross-country locomotion, in which the course may be chosen in any direction of the whole area.

The final goal of development for the test stand controls is, therefore, locomotion with free choice of speed <u>and</u> free choice of course in the terrain. To that purpose, the driver <u>must</u> be able to see the terrain: This is shown on a monitor either by the visual scene from the vehicle or by a map of terrain on which the driver sees his own vehicle. Selection of speed and driving direction is up to the driver. The computer determines the coordinates of the tank on the map, calculates the driving resistance, and considers at the same time the terrain parameters changing with the coordinates, such as inclination or solidity of the ground. Those parameters of terrain must have been stored in the computer as a map of terrain. This kind of simulation of a freely selectable cross-country course offers possibilities of investigating the dynamic behavior of the tank also from the tactical point of view.

In driving simulation, it is a particular advantage that determined vehicle parameters can be changed, thus allowing a clear demonstration of their influence. As an example, it is possible to modify the vehicle mass by potentiometer settings and to investigate in this manner the influence of the specific performance (kW/tons) on mobility.

It should be noted that the development of the possibilities of simulating the turning resistance on the test stand has not come to a close so far.

Testing Possibilities

The spectrum of possibilities for test and simulation with steady-state control, programmed control, and driving resistance simulation has been compiled in key points, in Figure 17. In view of its high flexibility and the good simulation of dynamic loads of operation, the test stand system PAISI is adequate for a large scale of tasks in research, in the development of drive trains, in trial work as well as for inspections of acceptance after a repair.

ACKNOWLEDGEMENTS

The development of the test plant has been carried out by the Brown Boveri Company (BBC), Mannheim, West Germany, in close contact with the Automotive Institute at the University of the German Armed Forces, Hamburg. The author thanks his assistants, Dipl.-Ing. W. Ehlert and Dipl.-Ing. G. Fietz, and is very grateful to the chief of the BBC test stand division, Dipl.-Ing. H.W. Weyland and his staff for their good and engaged cooperation.

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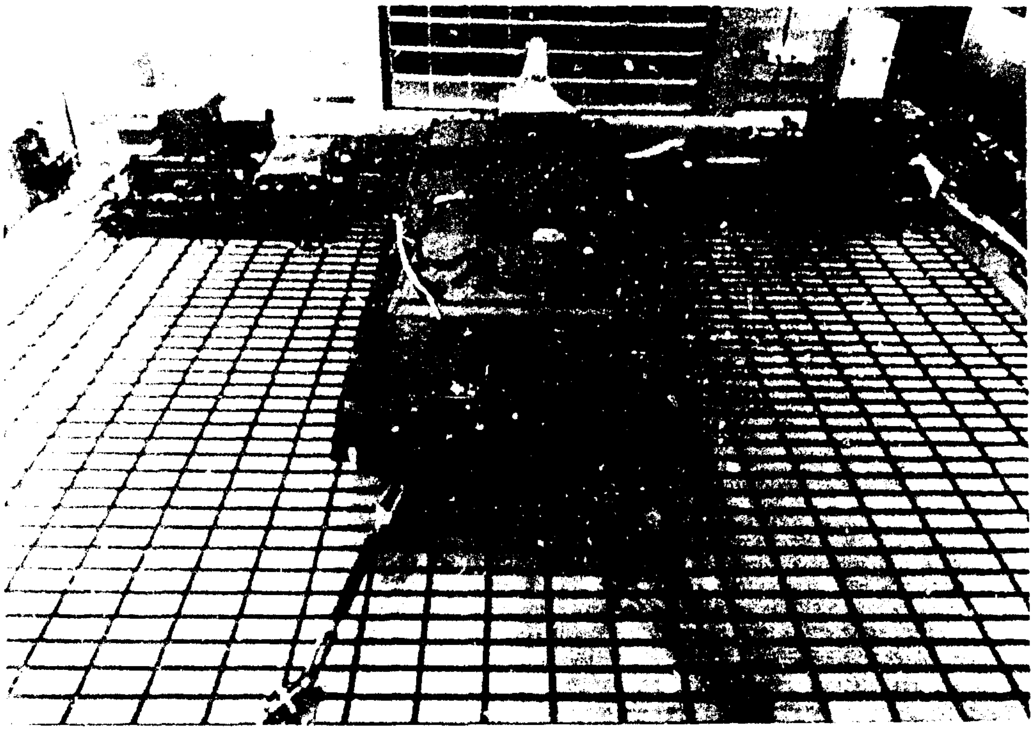

Fig. 1: Power and Inertia Simulator (PAISI)
with a test vehicle

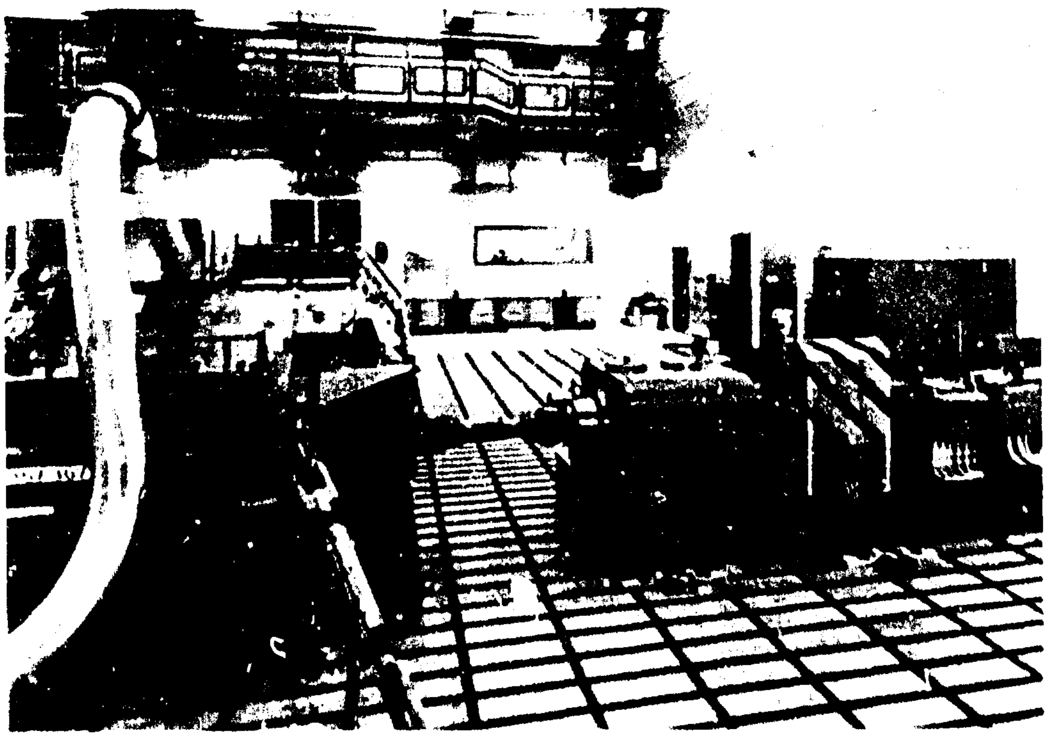

Fig. 2: Operating hall of the test stand

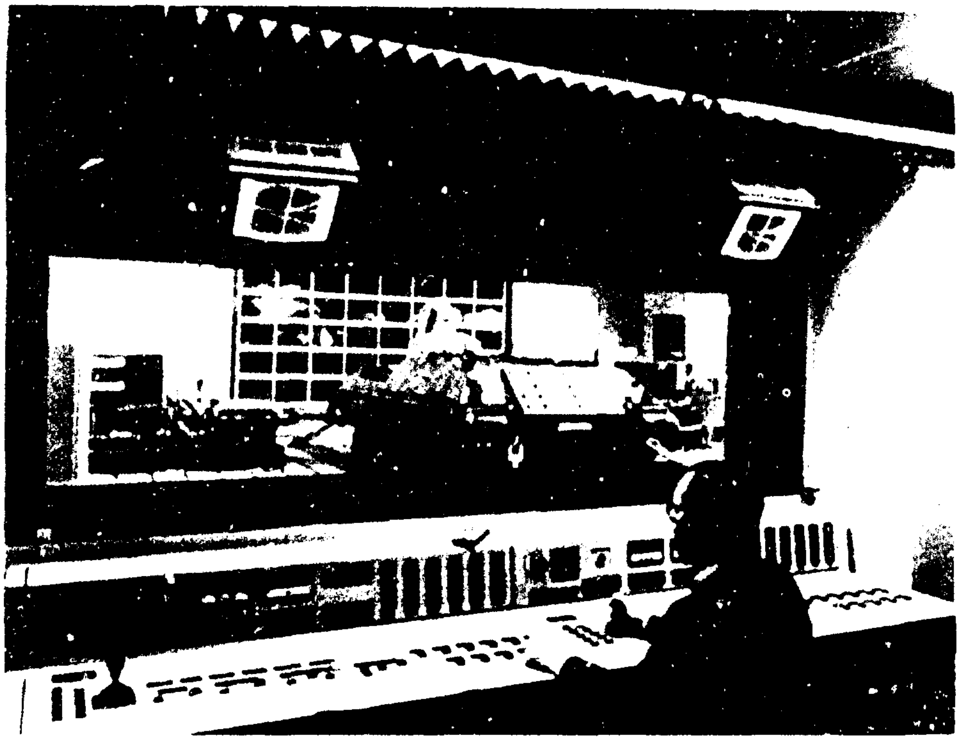

Fig. 3: Control desk and view to the test stand

Fig. 4: Control room with the electronics of the test stand

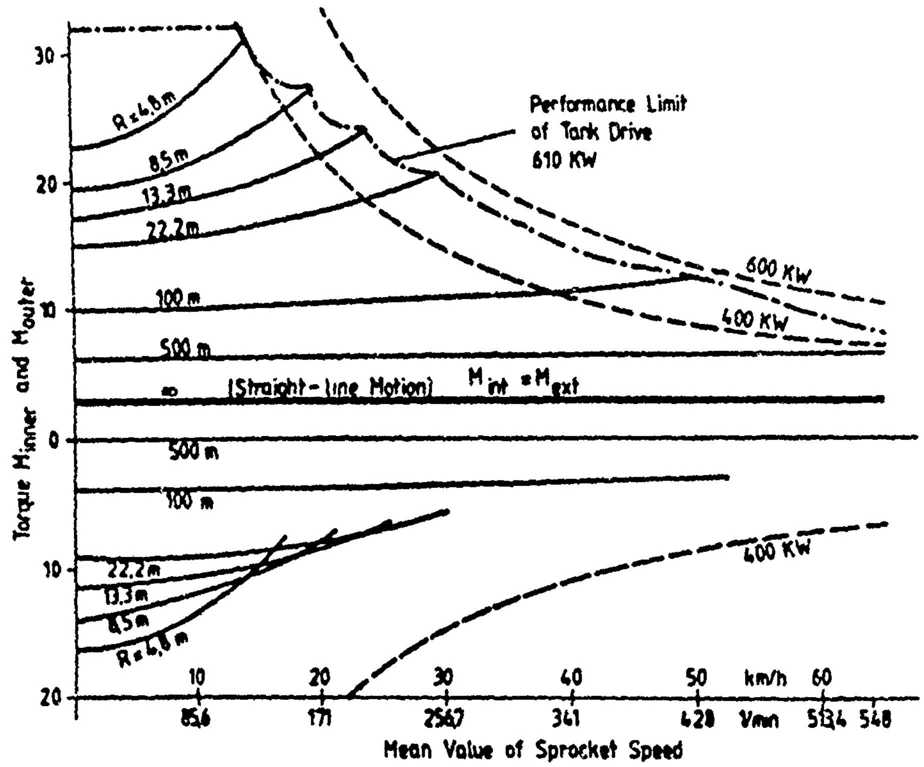

Fig. 5: Torques at the sprockets of Battle Tank LEOPARD I
in circular motion (radius R) on the road

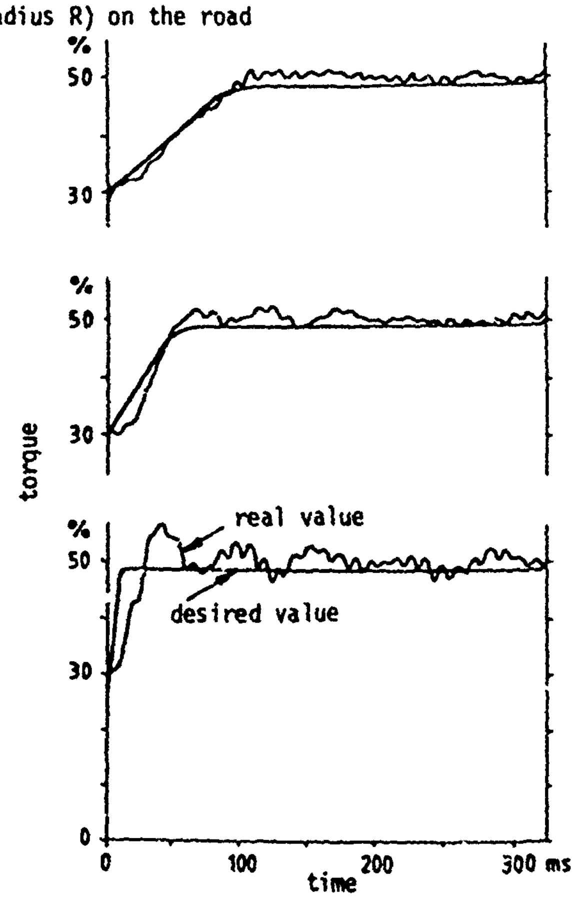

Fig. 6: Course of torque (desired and actual value) at a sudden increase of the desired value

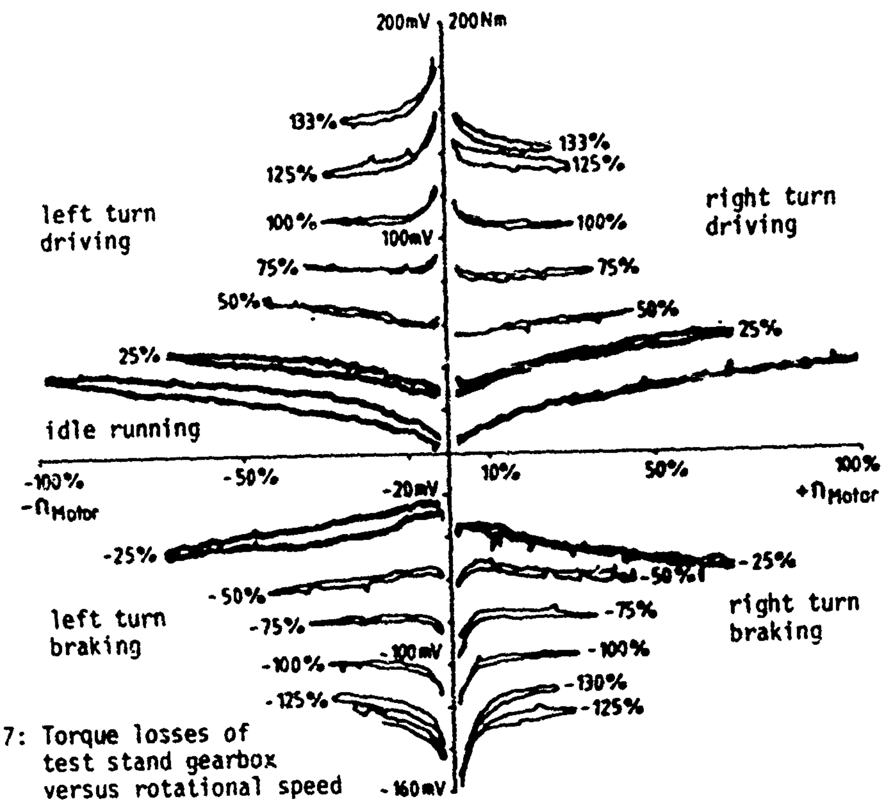

Fig. 7: Torque losses of test stand gearbox versus rotational speed

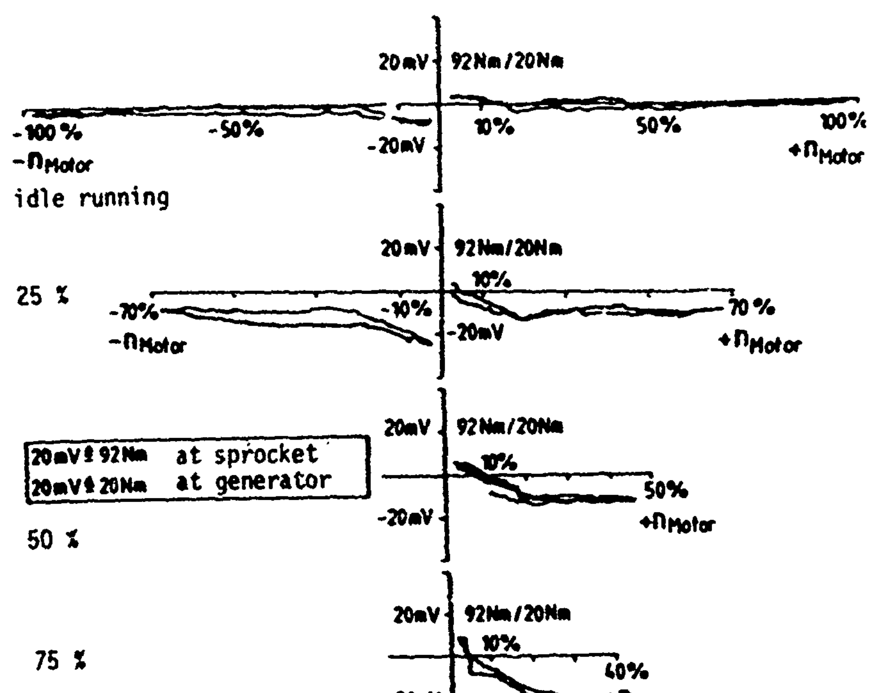

Fig. 8: Torque measurement accuracy considering losses of the test stand gearbox after compensation for different steps of load

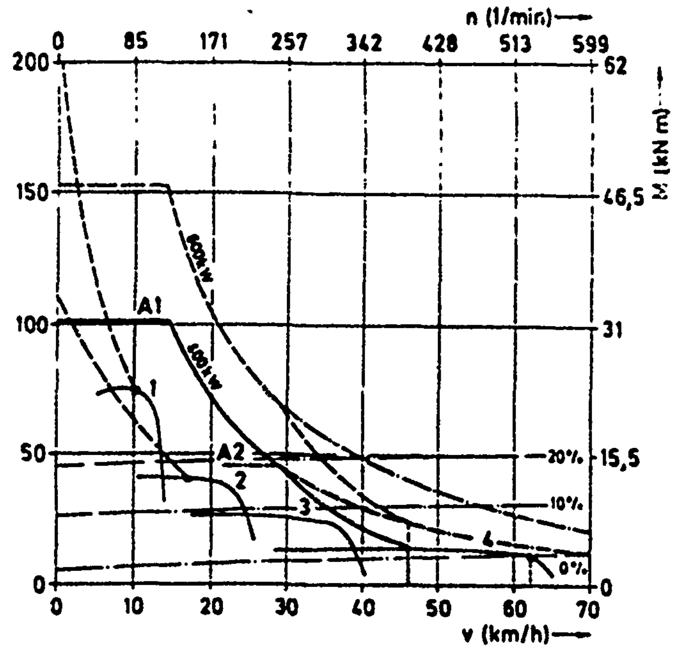

Fig. 9: Drawbar pull/torque - speed characteristics at each sprocket of a Tank LEOPARD I

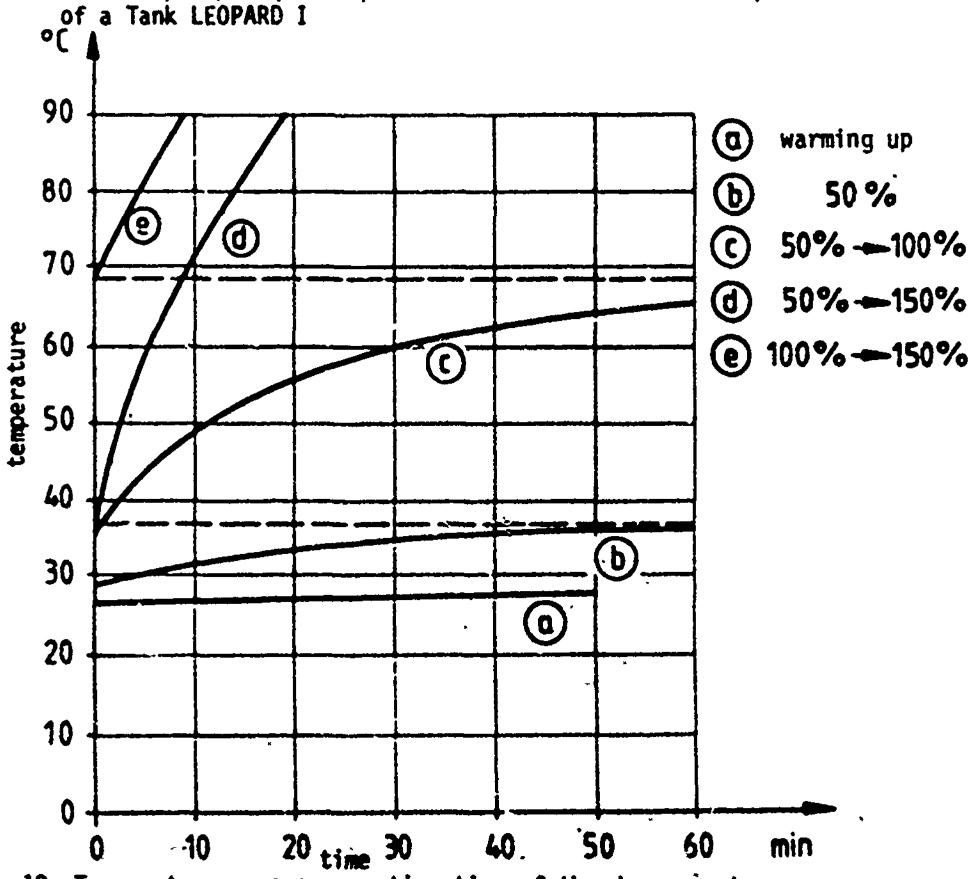

Fig. 10: Temperature versus operation time of the dc-generator during overloading

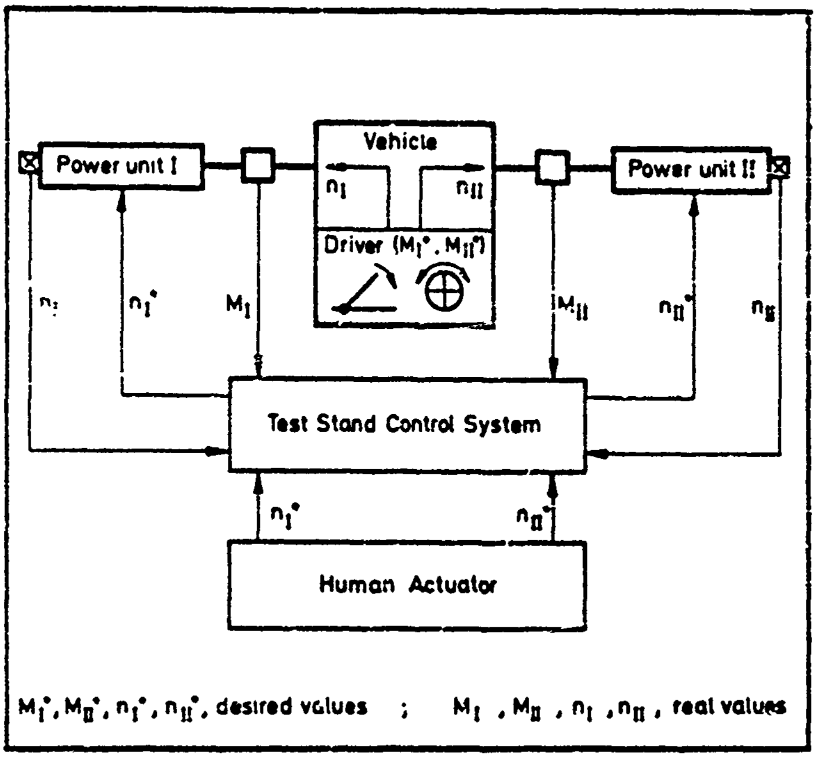

Fig. 11: Steady-state control of test stand

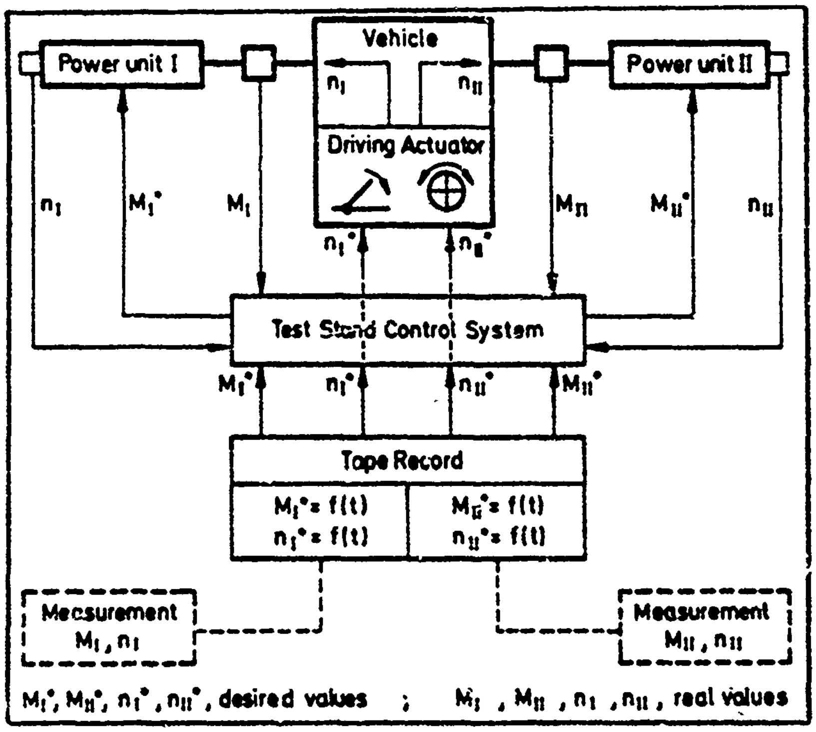

Fig. 12: Programmed control of test stand

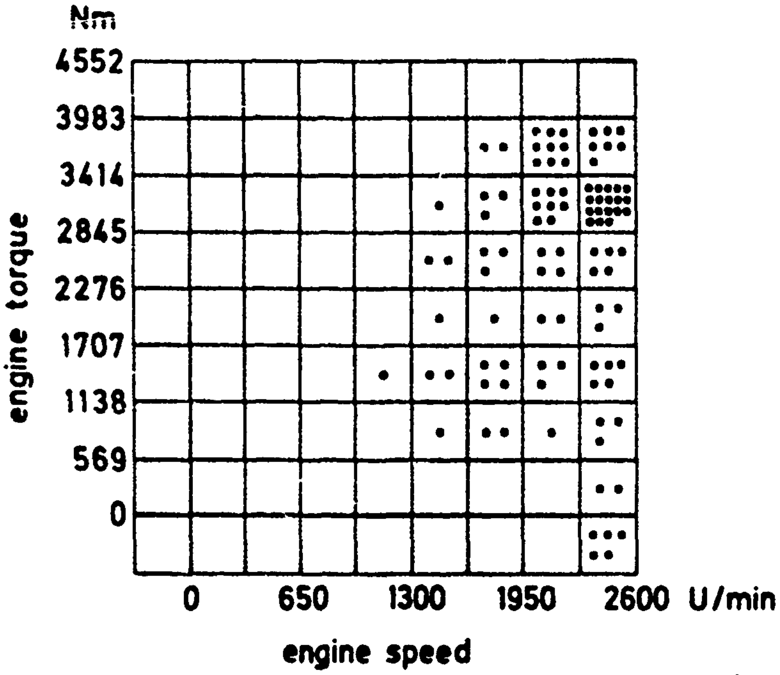

Fig. 13: Example of a load - speed collective (histogram)

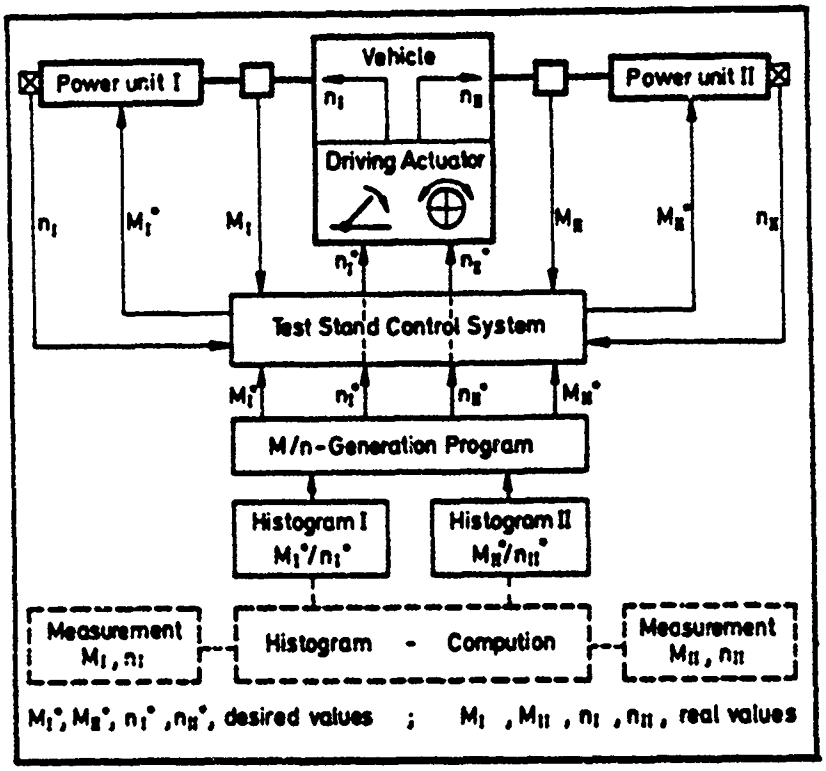

Fig. 14: Generation of stochastic load and speed program (histogram)

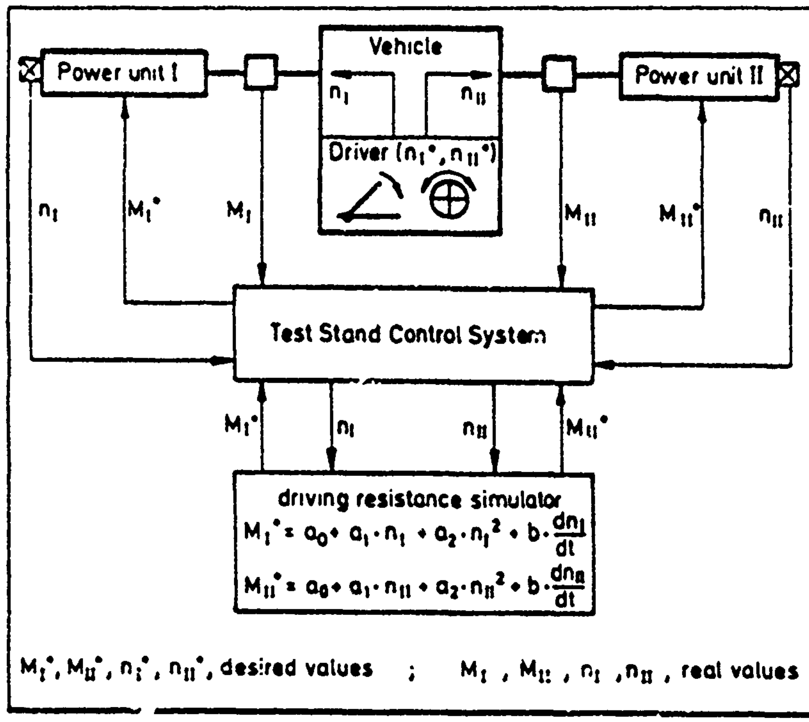

Fig. 15: Driving resistance simulation (without turning resistance)

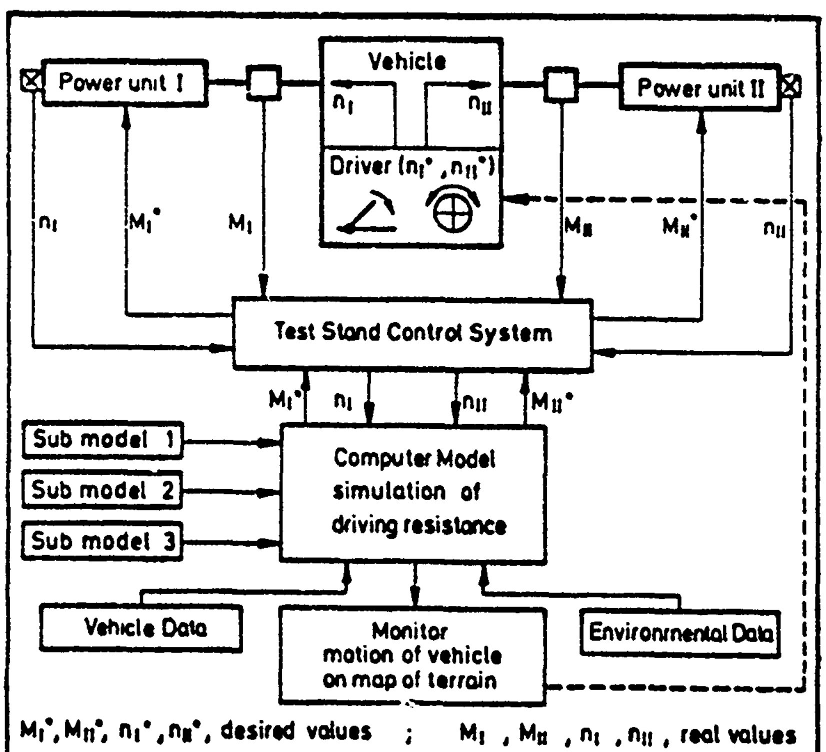

Fig. 16: Complete driving resistance simulation

TESTING AND SIMULATING POSSIBILITIES OF THE
POWER AND INERTIA TEST STAND
(TANK TEST STAND "PAISI"

CONSTANT CONTROL $n = const\ (M = const)$	PROGRAMMED CONTROL $M = f(t),\ n = f(t)$	DRIVING SIMULATION $M = a_0 + a_1 \cdot n + a_2 \cdot n^2 + b \cdot \frac{dn}{dt} \pm C$
OPERATIONAL POINTS	LOAD CHARACTERISTICS	SIMULATION OF MISSION
CHARACTERISTICS OF PERFORMANCE CHARACTERICS OF TRACTION VERSUS SPEED CHARACTERISTICS OF FUEL CONSUMPTION CHARACTERISTICS OF EXHAUST GAS MEASUREMENT OF EFFICIENCY NOISE DRIVE TRAINS OSCILLATION STARTING BEHAVIOUR	FUNCTIONING TESTS ENDURANCE TESTS MECHANICAL AND THERMICAL LOADS ON ENGINE, CONVERTER, GEAR BOX, STEERING GEAR, AIR AND OIL HEAT EXCHANGER	DYNAMIC DRIVING TACTICAL DRIVING OPTIMIZING OF SYSTEM MAXIMUM SPEED ACCELERATING CAPACITY SLOPE CLIMBING CAPACITY BRAKE TEST FUEL CONSUMPTION VERSUS DISTANCE TRAVELLED EXHAUST GAS EMISSIONS

Fig 17: Spectrum of testing possibilities