PROCEEDINGS OF THE EPA/INDUSTRY EMISSIONS TESTING PRACTICES SYMPOSIUM

DYNAMOMETER CALIBRATION AND CHARACTERISTICS

AUGUST 16, 1979

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EMISSIONS TESTING PRACTICES SYMPOSIUM

DYNAMOMETER CALIBRATION AND CHARACTERISTICS

August 16, 1979
9:00 a.m. - 4:00 p.m.

HELD AT

EPA Laboratory
2565 Plymouth Road
Ann Arbor, Michigan 48105

Written and Compiled
by Don Paulsell

Proceedings of the Dynamometer
Testing Practices Symposium

Introduction: In 1977, personnel from the EPA Motor Vehicle Emission Laboratory and the auto industry formed an ad hoc committee to conduct a series of symposia on emissions testing quality control practices. Five sessions were held to discuss dynamometers, lab correlation, gas traceability, interlab test site diagnostics, and curve generation/ data validation. Each symposium was summarized by its chairman in a document similar to this one.

A new committee has been convened this year to plan, coordinate and conduct a new series of symposia. The first symposium was held at the EPA Laboratory on Thursday, August 16, 1979 to discuss areas related to dynamometer calibration and characterization.

In preparation for the symposium, a questionnaire was mailed with each announcement to gather information related to dynamometer characteristics and acceptance parameters for calibrations. The responses received have been summarized in Attachment III.

Synopsis of Discussions: The general outline in Attachment I summarizes the areas discussed during the symposium. This synopsis does not represent a transcript of exactly what was said, but will attempt to summarize in general terms the highlights as noted during the meeting. Item 1 in the outline was an introduction and discussion of the symposium program. The chairman of this symposium, Don Paulsell from EPA, briefly described what the committee planned to achieve by holding a new series of symposiums. It was emphasized that this program would be a series of informal technical information exchanges dealing with topics of interest to both EPA and the industry. The new program would be called Emission Testing Practices rather than Quality Control Symposiums to reflect that emphasis would be on standardization of testing practices. However, quality control is a significant part of any standardization practice; therefore, quality control techniques will still be emphasized. It was also reiterated that participation is the key note of these symposia. If they are to be effective and achieve their stated goal, informed and prepared participation is a very necessary condition. This particular symposium had approximately 60 attendees, but it is estimated that only 15 actually participated with information. While nonparticipating attendees may have learned a great deal, it seemed that the presence of a large number of people was somewhat inhibiting to others who might have been tempted to participate in a smaller group of people. The purpose of these proceedings is to provide a mechanism for everyone to share in the discussions at the symposium without making their attendance mandatory. The committee hopes that future symposia can also reflect this philosophy.

The second area discussed during the symposium was calibration practices used for calibrating the Clayton ECE50 dynamometer with 125 lb. inertia flywheels. EPA recently finalized and distributed two procedures dealing with the subject of calibration practices; one on the electronics circuitry setup, and one on the frictional horsepower calibration. A third procedure dealing with weekly verification and quality control practices will also be distributed.

In January, 1977, EPA had published a draft procedure and recommended practice discussing the technically correct method of subtracting 155 lbs. of inertia for the horsepower calculations. This practice offers significant quality control assessment capabilities, but at the same time introduces a nonuniform bias in dynamometer horsepower settings. Therefore, the industry contended that a notice of proposed rulemaking would be required to implement this change. However, the benefits of this technique were judged to be so significant that EPA continues to utilize the technique for quality control purposes, but reprocesses the data in the manner that it has always used for the certification purposes.

Don Paulsell discussed the acceptance criteria and the validation process which is used to verify the data collected in Test Procedure 202. A copy of the data sheet was discussed and EPA offered to provide these to any manufacturer who would collect the data specified. EPA would process the data using the EPA program to provide a comparative analysis between EPA and another facility. In response to a question regarding the availability of the EPA computer program, it was stated that EPA's policy was not to release software because, at times, the documentation is not complete and routines that are incompatible with other computer programs frequently cause problems and consume considerable time in trying to get the program to work. The information, however, in the Test Procedure 202 should be sufficient for other facilities to model their computation practices around the EPA procedure, if they so wish to do this.

All of the acceptance criteria used to validate Test Procedure 202 are included in the procedure. The following paragraphs will briefly summarize the areas discussed and some of the information provided by the participants at the symposium regarding related criteria used throughout the industry.

The data manipulation routines used for Test Procedure 202 are specified through an interactive program. First of all, one can specify whether full inertia weight or (IW-155) is to be used in the computation of AHp. The initial QC processing normally uses the (IW-155) routine. The technician can also specify a regression of thumbwheel (TW) vs AHp or can use the integrated data (T*S) vs AHp. In both cases, no certification table is produced. This information is printed out in the upper right hand corner of the first page of the printout. All of the general descriptive information concerning the dynamometer and its calibration are listed in the upper left hand corner of the first page of the printout. The last item in that listing is the bearing friction as calculated from the average of five rear roll coastdowns. EPA performs the coastdowns of the rear roll during several flywheel warmup periods in the procedure. The average coastdown time for this rear roll is usually 50-60 seconds; the variation is ±.5 seconds from the average time.

At this time, a discussion of bearing friction took place. The requirement for stable and constant bearing friction during a calibration and dynamometer operation is very critical to the correlation of dynamometers. EPA typically uses a break-in period for new bearings of approximately eight hours. Some participants at the symposium mentioned that they have found it necessary at times to use 24 hours of vehicle operation to break in the bearings and at times General Motors has used up 72 hours before they considered the bearings stable. The most critical parameter related to bearing friction seemed to be the alignment of the bearing housings and the shaft couplings. The use of laser equipment or measurement devices capable of detecting ±.005 inches of misalignment were mentioned as standard techniques. Although the quantification of bearing friction values has not become a well established practice, limited data seems to indicate that the frictional values for the new C3 bearings used in the Clayton dynamometers do not differ significantly from those values obtained with standard bearings. Attachment IV illustrates the frictional values obtained across all EPA dynamometers for these new bearings. Also shown on that table are the frictional values obtained two years ago with the standard 250 lb. configuration.

Only the trim friction appears to be lower for the new units. As a final aside to the discussion of bearing friction, we briefly touched on the use of machine rolls. It was noted that machined rolls produced better driveability and seemed to have lower noise level, and for obvious reasons, more consistent circumference. This may be reflected in the statistics associated with the actual distance measurements made on EPA dynamometers as compared to General Motors dynamometers, which have the machined rolls. These data are attached in the survey. EPA will be installing the machined rolls late in 1979 on all of their dynamometers, so the assessment can then be made as to whether machined rolls provide better precision for distance measurements.

The discussion turned to the description of the validation process of EPA's calibration data. The top three sections on the printout illustrate the speed calibration data, load cell calibration data, and data collected to quantify the exponent of the power absorber unit.

The tach generator signals are compared to an absolute reference speed as detected by the frequency of a 60-tooth digital output from a gear and magnetic pickup. Other laboratories are using similar techniques such as high resolution pulse encoders and micro processor data acquisition to perform the calibrations on the tach generators. EPA performs an 11 point tach generator calibration, an 11 point dead weight calibration on the load cell, and an 11 point power absorber curve. However, on the power absorber curve, only the data from 25-60 mph are used in the log-log regression to obtain the exponent. EPA's criteria on exponent control is 1.85 to 2.15. Generally, the exponents are much closer to the value of two and can be adjusted, although the circuit cards have sealed adjustments. The participants from VW agreed that the value of two was an appropriate exponent for track data. However, GM noted that 2.0 may not be the right value for the dynamometer simulation, which differs because of the tire roll interface characteristics. EPA's quantification of the PAU exponent comes from steady state type of data collection. General Motors explain that they also employ an XY plotter to look at the relationship under dynamic conditions. They generally see that the torque speed relationship is slightly low during the accelerations and comes back a little high during a deceleration. This characteristic was noted during the last symposium, but it is generally accepted that the automatic control mode reduces the historesis band significantly, as compared to the manual control mode.

The next section of the computer printout used at EPA shows the three point regressions used to fit the thumbwheel (TW) vs total horsepower (AHp). When the (IW-155) calculation is used, the slope of these curves all approximate 1.00 and should be very uniform across all of the inertia weights. The occurrence of a slope of 1.00 means that the friction does not change as a function of a PAU or TW setting. A slope that is less than one indicates that friction is increasing with increasing thumbwheel setting and a slope greater than one means that friction is decreasing. Theoretically, the slope could also be affected if the inertia weight used in the calculation was not physically equivalent to the value assumed in the calculation. EPA has verified the dimensional parameters of all of the inertia weights and has discussed this aspect with Clayton. Deviations from assumed inertia values are not considered to be a significant problem.

EPA has experienced some deviations in slopes from the theoretical value of 1.00. Special investigations have not revealed the cause of these differences, although, shaft misalignment and dead band adjustments are prime candidates for causing this characteristic. In particular, the deviations from theoretical behavior appear to be associated with the engagement of the 2,000 lb. flywheels. From another viewpoint, the shifting in average slopes can also be associated with thumbwheel values set for the PAU, since the values change at 3000 and 5000 pounds, coincident with the engagement of 2K flywheels.

EPA also prints out the average speed during the coastdown as a measure of uniformity of data. Although one would assume that the average speed of a 55-45 coastdown would be 50 mph, the integrated speed is more on the order of 49.7. The representative of Ford mentioned that their microprocessor data collection method indicates the instantaneous derivative of the speed time trace is higher at 50 than would be approximated by the average computation of $\Delta v/\Delta t$ over the 55-45 range.

The frictional horsepowers determined in the individual inertia computations are also summarized in a table of flywheel frictions. This information has become very useful in detecting significant changes in flywheel friction and in assessing the uniformity of the values obtained through the selective inertia subtractions shown in the third area of the printout. These subtractions determine the flywheel friction contributed to the total friction by the addition of the single flywheel. When large amounts of scatter are observed in the statistical summary, this table is extremely useful in identifying which inertia weights are associated with the abnormal data. The final printout on the (IW-155) processing is a regression of integrated horsepower T*S and IHp @ 50 versus the thumbwheel value set. This information is useful in verifying the stability of Hp control across thumbwheel values of 3 to 22 horsepower.

All of the techniques previously discussed and the acceptance criteria that are employed in validating the calibration data set are specified in Test Procedure 202. Furthermore, a comprehensive example of each area is printed in the attachment to 202. Once the data set has passed this validation procedure, the data can be reprocessed by requesting a certification table. This option automatically defaults to TW vs AHp and the Full IW computation for total horsepower. Although this example was not attached to Test Procedure 202, an example of this printout for the data set used in 202 is in Attachment V of the appendix.

EPA Test Procedure 207 was not discussed in detail since it is primarily a reiteration of what Clayton specifies in their calibration procedure for the electronics. However, there are some functional checks that EPA performs and there is a log sheet in Test Procedure 207 that is filled out for diagnostic purposes which may be of some interest to people establishing new calibration procedures.

Test Procedure 302, which is the weekly verification and monitoring procedure, was briefly discussed, although this procedure is in its final review phase. A copy of the data sheet that is used with Test Procedure 302 and the control charts, which are plotted to assess the stability and trends in verification data, were distributed. The use of control chart techniques for data of this type cannot be emphasized too strongly. While many facilities employ a go, no-go criteria to verification coastdown times, the use of control chart can show trends in data long before the criteria are exceeded. EPA's experience generally indicates that it is shifts in the control circuit, primarily the dead band, which cause the coastdown times to deviate from their theoretical values. This indicates that the dynamometer friction obtained during the initial calibration is probably a very stable, consistent parameter.

Closely related to the discussion of the calibration practices was the review of the data collection and instrumentation methods which are employed to collect the coastdown data. Most laboratories are utilizing an automated type of triggering device to measure the coastdown times. EPA uses an analog comparator which has a filtered tach signal as an input, which is compared to precision reference voltages set at 4.500 and 5.500. The EPA circuit can control and trigger to within $\pm .02$ mile per hour of the given set point. General Motors stated that they use an optical encoder on the front roll shaft and a digital comparator circuit to trigger the 55 and 45 set points. Although the precision of their timing circuit was not mentioned, they did say they had a repeatability of about .2 horsepower over an eight-month period across two dynamometers. Ford uses a single pulse per rev input to a microprocessor which can time the period of each pulse compared to a set point. At 50 mph, each period would translate to 30 milliseconds. However, since Ford obtains coastdown times that are quite long, in excess of 30 seconds, this uncertainty translates to a triggering accuracy of $\pm .01$ mile per hour.

Many laboratories stated that they perform multiple coastdowns to minimize the uncertainty in the coastdown value. While repetitions are a valid technique to reduce uncertainty, it becomes very time consuming when calibrating the 125 lb. inertia configuration because of the large number of flywheel inertia weights needing calibration. The utilization of a highly precise triggering circuit and the quantification of its precision can eliminate the need for these multiple coastdowns. The data validation techniques discussed earlier can also eliminate the need for time-consuming data collection by taking advantage of the characteristics of the <u>complete</u> data set.

Another technique which was discussed for reducing the amount of time required to calibrate a dynamometer was the use of what is referred to as "prime" inertia weights. This method can utilize either the successive addition or the successive subtraction of individual flywheels. Since there are seven flywheels, one can determine the frictional horsepower of each individual flywheel by the successive subtraction of the total friction of each of the "prime" combinations.

Prime combinations can be formed starting from the 1,000 lb. trim and adding individual flywheels one at a time, or can be derived by starting at 6,875 and successively dropping each flywheel from the total combination. EPA has elected to use the former method because it offers the ability to get diagnostics data on each particular flywheel. Ford has had considerable experience using the latter method and has found very good agreement between the prime calibration technique and the total calibration technique. Their investigations are still in the developmental phase and other laboratories are encouraged to try this technique and to report their results at a later time. The use of prime flywheel friction determinations requires the subtraction of the rear roll 155 lbs. However, this manipulation can be retransformed by using (IW-155)/IW to generate the current certification type equation for each inertia weight. As more laboratories begin to collect and analyze individual flywheel frictions and calibration slope characteristics for the (IW-155) type of analysis, more data will be gathered to quantify whether the prime inertia calibration method is, in fact, an acceptable equivalent.

The third area discussed during the morning session dealt with dynamometer characteristics. Some of these have already been discussed, but two additional parameters are worth noting. First is the sensitivity of the load cell to changes in barometric pressure. Representatives of Clayton have noted that the hermetically sealed load cell displays this characteristic. They have instituted a modification to minimize the sensitivity. This has been done by venting the internal cavity of the load cell through a sealed type vent. General Motors stated that they had simply drilled a hole and put a filter into the hole on the side of the load cell cavity. Clayton personnel said this would be an acceptable practice, but any metal chips that might fall inside during the drilling operation could be detrimental to the operation of the load cell.

The second area discussed related to the slip characteristics of the dynamometer during a horsepower verification or setup procedure. Slip is defined as the difference in speed between the front roll and rear roll caused by the difference in loads imposed on the vehicle tire by the two rollers. This slip characteristic will not affect the control of the power absorber unit in the automatic mode, but can affect the indicated horsepower readout when observed in the rear roll speed position. EPA has standardized its practice of verifying all dynamometer load control functions at the PAU reference speed of 50 mph. Laboratories that use the manual mode of load control should be particularly aware of the effects of slip and should avoid setting any calibration or test setup PAU values in the rear roll mode. In other words, front roll (PAU) speed should always be used for calibration and test setup. The driving trace is generated by the rear roll signal.

After a one-hour break for lunch, the symposium reconvened to discuss the fourth topic on the general outline, calibration monitoring and correlation. The topic of site-to-site variability was discussed and areas were highlighted which affect the variability from one dynamometer to another. The tire roll interaction is considered by most to be the largest source of variability related to loaded vehicle coastdowns. Although not too many laboratories within the industry perform quick checks, EPA's data tends to indicate that repeatability of a quick check on one particular dyno is on the order of $\pm$.2 seconds or approximately 1% of the coastdown time. However, when determining alternate power absorber settings across different dynamometers, the variability can be much larger, because the amount of friction in the vehicle drive train considerably dominates the frictional component of the dynamometer. Therefore, the sensitivity of coastdown time to horsepower setting is approximately one to two or 7% change in coastdown time would equate to 15% change in dynamometer horsepower setting. The participants from Mercedes Benz noted that the restraints that are used during the test can also contribute to significant variability in dynamometer loading. This was quantified in a submission made during the 1977 symposium on dynamometers.

Load control and transient response characteristics are another source of variability. EPA has instituted a monitoring practice to verify that the high rate load and unload response times are typically less than eight seconds for each dynamometer. This represents a change of 5 to 15 horsepower and from 15 to 5 horsepower. Plugged valves due to water deposits can frequently make the response time much longer than this. Some laboratories have gone to closed water systems for the power absorber unit, while others employ the standard Clayton plug for removing these deposits. EPA briefly discussed its experience with using a grease gun to unplug clogged ports. The lower temperatures now being used in the power absorber units do not allow the grease to be purged from the system. Therefore, we are recommending that water soluble hand cleaner be used at all times. This technique works just as well and does not cause the problems of grease deposits and the almost certain reocurrencce of plugged ports.

The use of the lower temperature setting specified for the Clayton 125 lb. dynos was briefly discussed. Clayton previously recommended a power absorber heat exchanger setting of 135°, but now specifies 90-100°. Some of the laboratories had collected data on this change and found a negligible difference in coastdown times for the two temperature settings.

Other techniques that can be used to assess calibration stability are the use of repeatable cars and dynamometer quick-check results. The statistical analysis of the quick-check results from a regularly scheduled repeatable vehicle can provide data for highlighting significant differences in dynamometers. These differences can then be diagnosed by other special tests. EPA's repeatable car has an average quick-check coastdown time of about 12 seconds. The overall variation from day to day is on the order of ±2%. The average difference from dyno to dyno is less than 1.5%. It is recommended that a rear-wheel drive, medium-size tire, vehicle be used for doing these repetitive coastdowns. The most important aspect of this procedure is performing it in a standardized, uniform manner so that the variability can be minimized and the data collected become more meaningful.

A lot of the specific details and data relative to the areas discussed in this synopsis are summarized in the composite response to the questionnaire survey sent out prior to the symposium. These are shown in Attachment III of the appendix. EPA's response to the questionnaire is also shown. Even if a laboratory did not respond to the questionnaire, they are encouraged to pursue the quantification of some of the parameters discussed in the questionnaire. This information could be submitted to the Quality Assurance Staff at EPA for a follow-up report to the symposium, if enough information can be compiled.

The last topic discussed in the general outline was a report on three special projects. The first was a dynamometer calibration project performed by a European group. Dr. Shürmann from Volkswagen reported on this project and a copy of the calibration technique is in Attachment V of the appendix. The report differentiates the various components of the total road load force required for vehicle load simulation and discusses the techniques which can be used to simulate this load on both the water brake and electric chassis dynamometer.

The second project reported was a technique for determining the power absorber curve by the use of torque wheel measurements. Wolfgang Berg from Mercedes Benz presented the results of their development activities using torque wheels and some of the instrumentation problems that can occur in the use of such a system. A copy of his report is in Attachment VI of the appendix for your reference. The third report was made by Glen Thompson of the Emission Control Technology Division of the EPA Laboratory. This report summarized the results obtained by comparing load/speed data from the track to that obtained on a Clayton dynamometer using three modes of speed reference; one using rear roll, the second using the front roll, and the third using the rolls coupled together with a synchronizing chain drive. This investigation concluded that the coupled rolls provided the best simulation to the on-road data. The elimination of the tire-to-dyno slippage has an associated increased net loading on the vehicle with a resultant decrease in fuel economy. A copy of this report is also attached in the appendix for your reference.

Following the symposium, a short tour of the EPA test facility was conducted for those visitors who had not seen the laboratory before.

Index of Attachments

ATTACHMENT I

EPA/INDUSTRY TESTING PRACTICES SYMPOSIUM
DYNAMOMETER CALIBRATION AND CHARACTERISTICS

August 16, 1979
9 a.m. - 4 p.m.

GENERAL OUTLINE

I. Introduction and Discussion of Symposium Program

II. Calibration Practices

- A. Describe EPA TP-202, 207, 302
- B. Data Collection and Instrumentation
- C. Analysis and Acceptance Testing

III. Dynamometer Characteristics

- A. Speed, Load, PAU Curves
- B. Flywheel Frictional HP
- C. Slip Characteristics and Effects

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IV. Calibration Monitoring and Correlation

- A. Site to Site Variability
- B. Quick Check Test Results
- C. Control Charts/Statistics

V. Special Projects

- A. CEC Dyno Calibration Project
- B. Alternate PAU by Torque Wheels
- C. Roll Coupling

ATTACHMENT II

EPA/INDUSTRY TESTING PRACTICES SYMPOSIUM
DYNAMOMETER CALIBRATION AND CHARACTERISTICS
August 16, 1979
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ATTACHMENT III

DYNAMOMETER CHARACTERIZATION QUESTIONNAIRE

This questionnaire has been developed to characterize the various physical components of the dynamometer system. It has been written around a Clayton ECE-50 (125#) dyno, but if you have data or information that can be supplied for other dynamometers it may be valuable for comparitive purposes. Many questions relate to how tests and analyses are done; if the space provided is insufficient to give an abbreviated answer, please use separate pages or send a copy of your company's procedure. If you do not have data readily available that is pertinent to a question or can't interpret the question as stated, mark it as N/A. If you have a particular question that you would like discussed at the symposium, please include it on the last page of the questionnaire.

The responses to these questions will be tabulated for discussion at a dynamometer symposium which will be held at the EPA Laboratory on Thursday, August 16, 1979. Your response to this questionnaire is voluntary. However, the generation of a more complete tabulation will facilitate a more thorough discussion of this subject. Please send your response to:

Quality Assurance Staff
EPA Laboratory
2565 Plymouth Road
Ann Arbor, Michigan 48105

If you have questions, please call us at 668-4342, 668-4239, or 668-4355.

DYNAMOMETER CALIBRATION PROCEDURE

1) Describe in general terms the calibration procedure used for your dynamometers, such as thumbwheel values, inertias used, repetitions, warmups, etc.

GM

Dynamometer warm-up is performed by driving a vehicle at 50 mph for 15 minutes on the dynamometer with the 6875 pound inertia test weight engaged and an indicated horsepower setting of 10.0 hp.

All inertia test weights used are calibrated at the three indicated horsepowers listed in Table 1.

Three coast downs are performed at each horsepower setting. These coast down times are averaged to calculate actual horsepower using the following equation:

$$\text{Actual horsepower} = \frac{0.06073 \times \text{Inertia Weight (pounds)}}{\text{time (seconds)}}$$

Table 1

Indicated Horsepower Table

Inertia Test Weights (pounds)	Verification Indicated Horsepower	Calibration Indicated Horsepowers
6875, 6750, 6625, 6500	12.0	18.0, 12.0, 6.0
6375, 6250, 6125, 6000	25.0	30.0, 25.0, 20.0
5875, 5750, 5625, 5500	11.0	16.5, 11.0, 5.5
5375, 5250, 5125, 5000	10.5	15.8, 10.5, 5.2
4875, 4750, 4625, 4500	10.0	15.0, 10.0, 5.0
4375, 4250, 4125, 4000	9.4	14.1, 9.4, 4.7
3875, 3750, 3625, 3500	8.8	13.2, 8.8 4.4
3375, 3250, 3125, 3000	8.0	12.0, 8.0, 4.0
2875, 2750, 2625, 2500	7.2	10.8, 7.2, 3.6
2375, 2250, 2125, 2000	6.4	9.6, 6.4, 3.2

FORD

Coastdowns at 5 thumbwheel settings per inertia calibrated - 22 inertias; warmup min. 30 minutes at start, other warmups as needed, - 50 MPH; 3 repetitions at first thumbwheel per inertia repeat within 0.2 sec., one coastdown at remaining 4 thumbwheels - Evaluating "Prime" wheel technique.

CHRYSLER

Complete calibration of each dynamometer performed each month:

Inertia Wt.	Thumbwheel values
2250-2500	4, 6, 9, 12
2625-3500	5, 8, 12, 16
3625-4500	6, 10, 14, 18
4750-5500	6, 11, 15, 20

Coastdowns are performed over the range of 2250-5500 lb.IW, at the increments required for certification testing. Three repetitions at each IW setting must agree within 0.2 sec. Warmup for 10 min. at 50 mph at 5500# IW before starting coastdowns.

IHC

Warm up with all inertia weights engaged for 20 minutes, select test inertia, strobe rear and front rolls at 1,800 rpm and observe speed on digital readout. Adjust Thumbwheel Settings every 4 h.p. from 4 to 20 horsepower. Repeat each setting a minimum of 2 times. Calculate actual horsepower.

TOYO KOGYO

50mph set Power; 4-15 HP(4-5 points) Inertia; 2000-3500 lb

Repetitions; 2 or 3 times each power set

Warm up; 15 minutes at 30mph

NISSAN

Warmup : 5375 lbs. at least 40 minutes at 50 mph.

	Inertia	Thumbwheel value
Inertia used and thumbwheel values:	2250~3125	
Repetition : repeat to obtain 3data	(every 125 lbs)	5, 7, 9, 11
within ±0.05 AHP	3250~3750 (every 250 lbs)	7, 9, 11, 13
	3750 lbs	17, 19, 21, 23

ERI.

tions, warmups, etc. Warmups: 4875 lbs., 30 minutes at 50 mph.

	INERTIA	THUMBWHEEL VALUE (HP)
Inertia used and thumbwheel values;	3750 lbs.	5.0, 10.0, 15.0, 20.0
aceitan if the indicated HP is within	3500 ~ 3000 (every 125 lbs)	5.0, 10.0, 15.0
± 0.1 HP of thumbwheel value at each	2875 ~ 2000 (every 125 lbs)	2.0, 6.0, 10.0

point. Repetition: repeat to obtain 3 data within ± 0.05 AHP.

EPA 15 min @ 6875 TW = 10Hp

33 IW's	1000 - 2875	3,6,10	TW	3 repeats on warmup
	3000 - 4750	5,10,15	TW	1 coastdown @ each
	5000 - 6500	10,16,22	TW	TW value

Read & record indicated HP @ FR50 for each TW. & integrate torque & speed. Data analysis with computer. See TP202.

2) Describe the curve fit method used to reduce the data.

GM

A least squares straight line is fitted through the three points of actual versus indicated horsepowers. From these coefficients, a look-up table of actual versus indicated horsepowers is tabulated in 0.1 hp steps. The deviation of any data point from the straight line must not be greater than 0.3 hp.

FORD, CHRYSLER, IHC, TOYO KOGYO, NISSAN, and ERI

Linear least squares fit of thumbwheel values versus the actual horsepower obtained.

EPA

LLSQ QC Validation

T*S vs AHp (IW-155), TW vs AHp (IW-155) and RRFhp

TW vs AHp (FULLIW) for Cert Table. No RRFHp used.

3) What other analyses are performed as part of the calibration data reduction, such as prime inertias, average slopes, or repeatability measurements?

GM

A slope check is performed monthly on the 6875 pound inertia test weight to compare current slope to the slope at the time the look-up table was generated. The equation used in this comparison is:

$$\% \text{ Difference} = \frac{\text{slope (current)} - \text{slope (old)}}{\text{slope (old)}} \times 100$$

If the slope is within +3%, a spot-check is performed using the verification indicated horsepowers (Table 1) for all inertia test weights used. The spot-check requires the actual horsepower to be within +0.4 hp of the current look-up table value. If a spot-check fails, only that inertia test weight needs to be recalibrated. If the slope check fails, complete recalibration of all inertia test weights is required.

The three coast down times at each indicated horsepower during a spot-check, slope check or calibration must be within 0.3 second of each other, or the coast downs are repeated until a group of three repeat within 0.3 second.

FORD

FHP_{new} vs. FHP_{old} $\pm$ 0.5 hp, slope = 1.000 $\pm$ 0.015, actual vs. calculated $\pm$ .3

CHRYSLER

The following additional calculations and tolerances are performed.

a) difference between actual coastdown HP and calculated HP from curve fit must be $\leq$ 0.15.

b) slopes (IW -155) are printed out in a summary and averaged. No tightly defined tolerance, but average slope should be .950 to 1.050, and individual slopes should be reasonably close to average.

c) Prime inertia method has been looked at, but is not being used to generate coastdown data. Frictional contribution is calculated from 7 weights (2250, 3000, 3125, 3250, 3500, 4000, 5000). Data has been summarized for informational purposes, but has not been used to generate horsepower settings at different inertia weights.

IHC

Calibration curves are compared to previous calibration and changes noted.

NISSAN and ERI Check the average slopes and the frictional horsepower of each inertia combinations.

EPA

PRIMES, SLOPES, σ, %C.V., 5 RR Δts, speed curves, torque curves, PAU exponent, T*S vs TW, IHp @ 50 vs TW, FW FHp analysis.

Toyo Kogyo replied N/A.

COASTDOWN TIMER FUNCTIONS

1) Describe your coast down timing circuit.

GM

The coast down timer is a microprocessor based system that utilizes a 160 pulse per revolution optical encoder coupled to the dynamometer front rolls. The instrument is programmed to determine when a coast down is in progress. At the completion of a test, the coast down time is displayed. Speed is also continuously displayed for the purpose of checking the indicated horsepower setting at a 50 mph front roll speed.

FORD

Describe your coastdown timing circuit. Microprocessor operating from Front Roll one pulse/rev. signal, measures period of revolution vs. period for 55 & 45 mph.

CHRYSLER

The front rolls tach generator voltage is monitored by a computer which triggers a timer at voltages corresponding to 55 and 45 mph. The computer prints out the elapsed time between the two speeds.

IHC

Speed tachometer and stop watch.

NISSAN

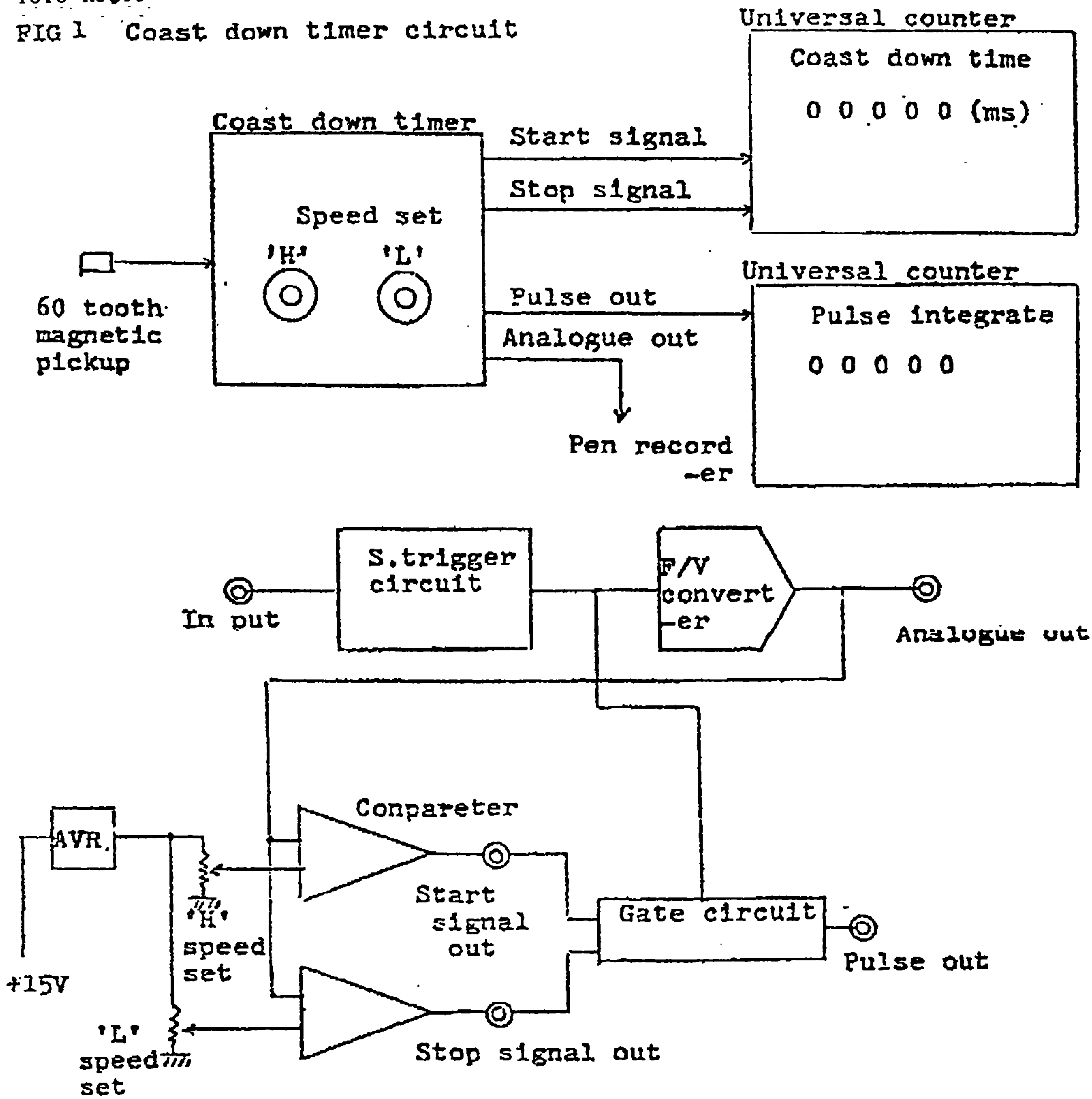
Describe your coastdown timing circuit. A pulse disc is mounted on the front roller shaft(60 pulse/rev). The pulse signal is compared with design pulse-wide(55mph and 45mph) in the comparator and timer gate is driven. The gate open time is measured with 1 MHZ clock signal automatically and displaced on the digital counter.

ERI

Describe your coastdown timing circuit. The computer scans the front roller tach-generator signal every 1/40 sec. and calculates the coastdown time comparing it with the stored values of 55mph & 45 mph.

TOYO KOGYO

FIG 1 Coast down timer circuit



EPA

Describe your coastdown timing circuit. Analog comparator.

See detailed circuits in Attachment 6.

2) What is the precision of this circuit using a clean ramp signal?

GM

The units' specifications are:

Display Resolution
Time - 0.01 second
Speed - 0.1 mph

Accuracy
Time - within one roll revolution; e.g., one revolution at 50 mph equals 30.8 milliseconds.
Speed - trimmed to display exactly 50.0 mph

FORD

Time + 0.01 sec; 4 MHz clock; 0.2% accuracy for speed detection.

CHRYSLER

± 0.1 mph, and ± 0.2 sec.

TOYO KOGYO

TABLE 1 Inspection results

In put		Analogue out	
Speed mph	Freq. Hz	Voltage V	Err. %
100	3886	10.001	+0.01
90	3497	9.003	+0.03
80	3109	8.002	+0.02
70	2720	7.001	+0.01
60	2392	6.003	+0.03
55	2137	5.501	+0.01
50	1943	5.002	+0.02
45	1749	4.503	+0.03
40	1554	4.000	0
30	1166	3.000	0
20	777	1.999	-0.01
10	389	1.001	+0.01
Precision		±0.03%	

Set Speed	Start signal out			Stop signal out		
mph	Freq. Hz	Speed mph	Err. %	Freq. Hz	Speed mph	Err. %
100	3878	99.80	-0.20	3886	100.01	+0.01
90	3491	89.84	-0.16	3498	90.02	+0.02
80	3104	79.88	-0.12	3107	79.96	-0.04
70	2720	70.00	0	2720	70.00	0
60	2332	60.01	+0.01	2331	59.99	-0.01
55	2137	55.00	0	2138	55.02	+0.02
50	1945	50.05	+0.05	1944	50.03	+0.03
45	1750	45.04	+0.04	1749	45.00	0
40	1555	40.02	+0.02	1556	40.04	+0.04
30	1168	30.06	+0.06	1168	30.06	+0.06
20	780	20.07	+0.07	778	20.02	+0.02
10	387	9.96	-0.04	387	9.96	-0.04
Precision	±0.07% (Less than 70mph)					

NISSAN
Delay time : less than 0.0006 sec.

ERI

Using 12 bits for A-D conversion, system readability is 0.02 mph & 1/40 sec.

EPA

±.005 secs

IHC replied N/A.

3) What are your set point accuracy or hysteresis criteria?

GM

Since the timer uses a digital pulse train, no hysteresis criteria is required. The optical encoder has a gray code output, which eliminates any jitter in the output. The microprocessor program is loaded for each dynamometer roll diameter to trigger on the exact digital comparison for 55 to 45 mph interval. The timer system runs off a 5 Megahertz crystal clock.

FORD

55.00 + 0.01, 45.00 + 0.01

CHRYSLER

± 0.1 mph

TOYO KOGYO

±0.03 mph (55-45 mph)

NISSAN

Using 60 pulse/rev signal, less than 0.001 sec.

ERI

The speed set point accuracy is +0.02 mph because of its readability.

The speed calibration has been made in steady state using the strobe.

EPA

±.002 VDC 4.500/5.500 HYS <.005

IHC replied N/A.

4) What types of integration or sampling techniques are used for torque, speed, or horsepower?

GM

Speed is continuously updated in a shift register until it agrees with the preset start and stop coast down speed values. Torque and horsepower are not integrated over the coast down.

FORD

No integration

IHP checked vs. dial @ 50 mph. must be within ± 0.2 hp, speed indication checked vs. microprocessor - ± 0.1 mph.

CHRYSLER

The front rolls tach generator voltage is monitored by a computer which triggers a timer at voltages corresponding to 55 and 45 mph. The computer prints out the elapsed time between the two speeds.

EPA
torque, speed, or horsepower? torque → VCO → TOTALIZED

SPEED → VCO → TOTALIZE. Could use 60 tooth gear.

IHC, Toyo Kogyo, Nissan, and ERI replied N/A.

5) What is the repeatability of dynamometer coast downs at TW = 10, IW = 6875?

GM
Repeatability was within ±0.25 hp as observed for two sites over an 8-month period.

FORD
Better than 0.2 sec.

CHRYSLER
Coastdown times are repeatable within ± 0.2 seconds

TOYO KOGYO

Coastdown time: 14.358,14.305,14.376 sec. (IW=3000 lb.Power=10 HP)

NISSAN

At TW = 10,IW = 5375 lbs,n = 10, coastdown time : σ = 0.13
(AHP : σ = 0.06)

EPA

Coastdown time: 31 ±0.1 secs

IHC and ERI have not collected any data on this.

6) Have you assessed the set point stability for the coast down timer?

GM

Digital comparator does not have any instability.

FORD

No detectable set point drift over 6 months.

CHRYSLER

Tach generator voltages (front rolls speed deflections) have been found to stay constant over periods of six months or more, unless the tach generator is changed.

NISSAN

$\pm 3 \times 10^{-11}$ sec/day

ERI

The computer is given the trigger speed values from a speed calibration line. This calibration @ 46.32mph has been made within $\pm 0.1\%$ for a year.

EPA

Set points 4.5 - 5.5 Volts, with a drift of ±2 mv/month, checked monthly reset if > ± 5 mv, variation : 3 mv, usually less than 1 mv.

IHC and Toyo Kogyo replied N/A.

7) What effect do the uncertainties associated with your triggering function have on the coast down time?

GM

A digital system does not have any uncertainties in triggering the start and stop function.

CHRYSLER

They give variations in coastdown time of up to ± 0.2 seconds.

EPA

The calibration of tach generation is held within 0.1% @ 50 mph.

Ford, IHC, Toyo Kogyo, Nissan, and ERI have no data on uncertainties associated with triggering function.

8) Do you apply any adjustment of speed data for triggering errors?

GM

The design characteristics of this digital coast down timer do not require any adjustment to be performed.

ERI

The 6 points speed data are averaged for comparison with the trigger value to avoid the fluctuation error of the tach-generator signal.

Ford, Chrysler, Toyo Kogyo, IHC, Nissan, and EPA make no adjustment.

RLPC CIRCUITS

1) Describe the signal readouts incorporated with your dynamometer?

GM

The two digital meters above the RLPC display front roll speed in mph and torque in ft-lbs. The Driver's Aid has a digital horsepower display and selectable digital front or rear roll speed display.

FORD

Digital displays of rear roll speed, front roll speed, indicated horsepower, and torque.

CHRYSLER

Front/Rear Rolls Speed and Horsepower is displayed on Drivers Aid Stand. digital meters with a resolution of .1 MPH and .1 HP.

IHC

Digital read out.

ERI

The CLAYTON ECE-50 RLPC standard type (Model No. A-32206) without reconstruction is used.

EPA

Digital speed, selectable front or rear roll hp always based on front roll speed. Vehicle factor pot not used, hp is varified at PAU 50.

Toyo Kogyo, and Nissan replied N/A.

2) Do you have control limits for the power absorber exponent? How is it assessed?

GM

At installation, the exponent module was set at 2.0 using Clayton's procedure. Monthly an X-Y-Y plot of torque and horsepower versus speed are compared to the theoretical plot (see Figure 1) when driving the second cycle of the FTP schedule. The RLPC electronics are adjusted, if necessary, to minimize hysteresis and exponent level changes.

CHRYSLER

No special control limits. The Load and Unload functions and dead-band are calibrated monthly per RLPC procedure in manual R-8713.

ERI No. It was confirmed that the power absorber exponent on the condition of trangient and steady state is 3.00 ± 0.08 at a speed of 20 to 60 mph.

EPA Control limits: 1.85-2.15. Typically 2.0 in automatic mode. 11 pt. Steady State PAU curve, torque and speed data used in log-log linear regression.

Ford, IHC, Toyo Kogyo, and Nissan have no control limits.

3) Are any tests performed to validate the accuracy of the thumbwheel signal?

GM

Yes, thumbwheel span is calibrated at one span value of 39.0 hp. New thumbwheel switches have been installed on the Driver's Aids that have the outputs of the three decades summed and weighted. These digital switches have had their output signals verified for linearity at installation.

FORD Drive 50 mph front roll speed & observe IHP - must be within ± 0.2 of thumbwheel; regress thumbwheel vs. AHP (corrected for rear roll) - slope within 1.000 + 0.015

CHRYSLER

Lo Span and High Span adjustment of Thumbwheel per RLPC procedure monthly.

IHC

Thumbwheel signal is calibrated per Clayton procedure.

ERI The agreement on the thumbwheel set value and the digital readout is checked to be within + 0.1 HP at 50 mph.

EPA

Only during electronics calibration as specified by Clayton.

Toyo Kogyo and Nissan replied N/A.

4) How is the indicated horsepower signal calculated?

GM

For the purpose of dynamometer calibration, the indicated horsepower is defined to be equal to the thumbwheel setting.

For the purpose of defining the theoretical torque and horsepower versus speed plot (Figure 1), the indicated horsepower is calculated as follows:

$$\text{Indicated horsepower} = \frac{\text{Torque x rpm}}{5252}$$

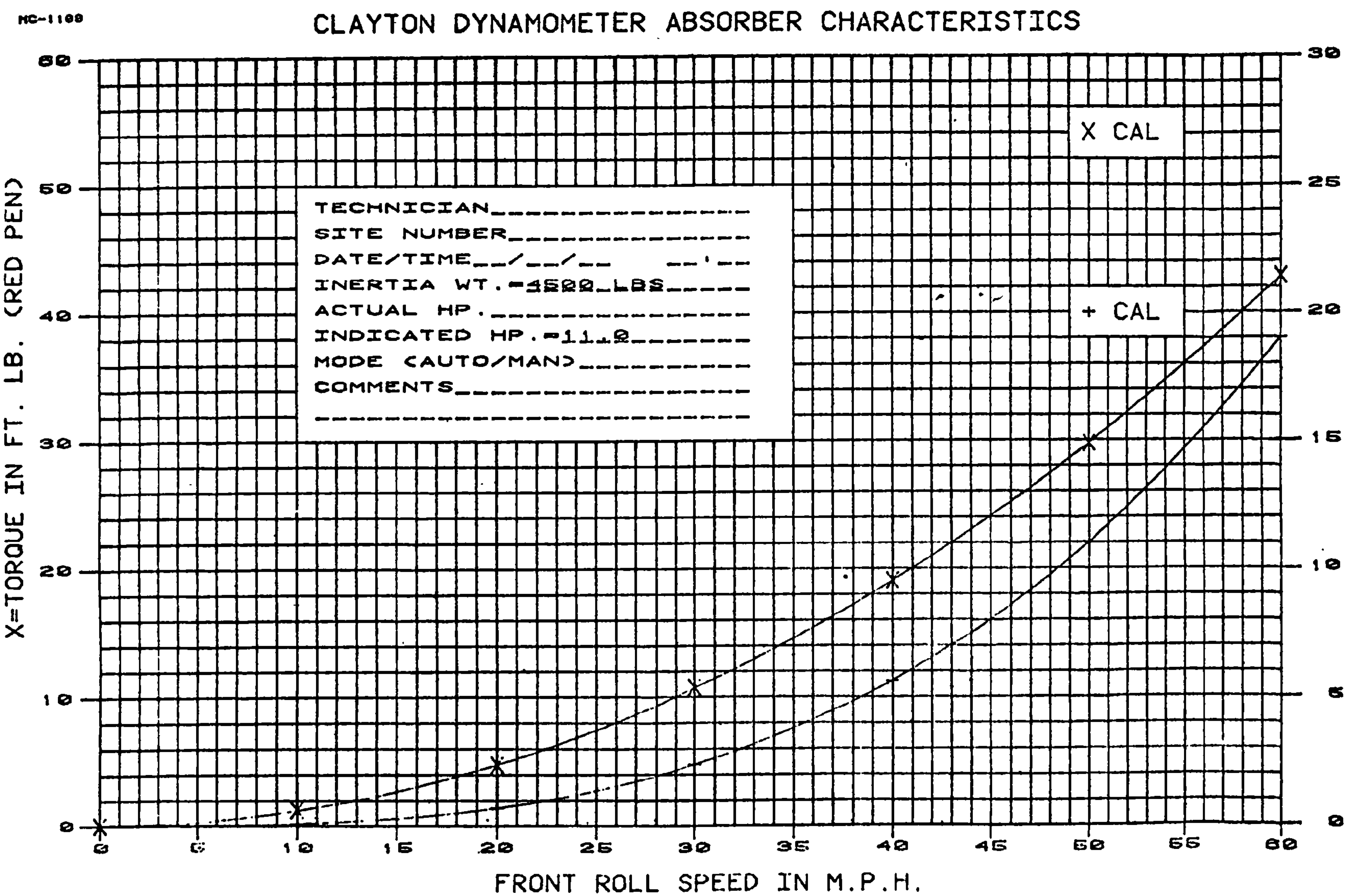
FORD

Front roll torque times front roll speed.

CHRYSLER

Board 8 of RLPC electronics calculates indicated horsepower. Torque cell is calibrated and the linearity of torque cell is verified per RLPC monthly calibration procedure.

Figure 1


MC-1100
CLAYTON DYNAMOMETER ABSORBER CHARACTERISTICS
X CAL
+ CAL
TECHNICIAN
SITE NUMBER
DATE/TIME __/__/__ __'__
INERTIA WT.=4500 LBS
ACTUAL HP.
INDICATED HP.=11.0
MODE (AUTO/MAN)
COMMENTS
X=TORQUE IN FT. LB. (RED PEN)
+=INDICATED HORSEPOWER (BLUE PEN)
FRONT ROLL SPEED IN M.P.H.
0
5
10
15
20
25
30
35
40
45
50
55
60

GM

IHC
Composite of front roll speed and torque signal.

ERI
Clayton's RLPC standard type without reconstruction is used.

EPA
How is the indicated horsepower signal calculated? EPA rewired circuit to always use front roll speed and torque for hp calculation. (Standard Clayton circuit uses front or rear roll speed as selected on control.)

Toyo Kogyo, and Nissan replied N/A.

5) What limits do you have for deadband control in the automatic mode and what problems have you encountered in meeting these limits?

GM
The deadband adjustments are performed monthly under static conditions. The low rate deadband is adjusted to be centered exactly at 3.0 hp and 39.0 hp and to be no wider than ±0.2 hp. The high rate deadband is adjusted to be centered at 39.0 hp and to be no wider than ±1.5 hp.

The problem with adjusting these deadbands is that they interact and several iterations are sometimes required. The driver cut-off module is being adjusted prior to the deadband adjustments to eliminate another potential interaction (more appropriate than Clayton's procedure).

FORD Thumbwheel ± 0.2 hp at all thumbwheel settings.

(1) Precise centering of deadband is difficult.

(2) Deadband width below 6 hp tends to be less than ± 0.2

CHRYSLER
.4 ± 0 H.P. Differences in calibration techniques existed when equipment was first installed. No problems in obtaining .4 HP deadband. A procedure modification was made to Clayton low range adjustment and driver cut off adjustment to meet .4 HP deadband. See Procedural Attachment.

CHRYSLER

Modification to Clayton Calibration Procedure

This change will make possible the .4 HP difference tolerance specified in the Thumbwheel low range adjustment.

A. Recommend adding the following procedure to the Thumbwheel Low Range Adjustment after Para 4-16-e-2.

NOTE: If unable to obtain a .4 difference in the thumbwheel low range adjustment then perform the following steps.

a. Adjust P6/B8 to obtain a reading of 2.8 HP on the Drivers' Aid Power Meter.

b. Adjust P1/B7 until low rate load light just comes on.

c. Adjust P6/B8 to obtain a reading of 3.0 HP on the Drivers' Aid Power Meter and observe that low and high rate load and unload lights are out.

d. Adjust P6/B8 to obtain a reading of 3.2 HP on the Drivers' Aid Power Meter.

e. Adjust P2/B3 unitl low rate unload light just comes on.

f. Adjust P6/B8 clockwise until the low rate load light just starts to pulse. Record the power reading.

g. Adjust P6/B8 counter-clockwise until the low rate unload light just starts to pulse. Record the power reading.

h. If unable to obtain a .4 difference then repeat steps 4-16-a through 4-

B. Recommend Para 4-19-e of the Driver Cut-Off Adjustment to read as stated below:.

Adjust P2/B3 clockwise to jsut stop low and high rate unload lights. This adjustment should vary no more than 1/8 turn from its original setting.

CHRYSLER

Modification to Clayton Calibration Procedure

This change will make possible the .4 HP difference tolerance specified in the Thumbwheel low range adjustment.

A. Recommend adding the following procedure to the Thumbwheel Low Range Adjustment after Para 4-16-e-2.

NOTE: If unable to obtain a .4 difference in the thumbwheel low range adjustment then perform the following steps.

a. Adjust P6/B8 to obtain a reading of 2.8 HP on the Drivers' Aid Power Meter.

b. Adjust P1/B7 until low rate load light just comes on.

c. Adjust P6/B8 to obtain a reading of 3.0 HP on the Drivers' Aid Power Meter and observe that low and high rate load and unload lights are out.

d. Adjust P6/B8 to obtain a reading of 3.2 HP on the Drivers' Aid Power Meter.

e. Adjust P2/B3 unitl low rate unload light just comes on.

f. Adjust P6/B8 clockwise until the low rate load light just starts to pulse. Record the power reading.

g. Adjust P6/B8 counter-clockwise until the low rate unload light just starts to pulse. Record the power reading.

h. If unable to obtain a .4 difference then repeat steps 4-16-a through 4-

B. Recommend Para 4-19-e of the Driver Cut-Off Adjustment to read as stated below:

Adjust P2/B3 clockwise to jsut stop low and high rate unload lights. This adjustment should vary no more than 1/8 turn from its original setting.

IHC Dead band control limits are ± 0.2 HP. We sometimes have experienced the fluctuation of torque at steady state.

ERI
± .2 hp -- no problems.

EPA ±.2 Hp - This is sometimes difficult to set @ all 3, 30, 39 points. Use a little wider @ 30.

Toyo Kogyo and Nissan replied N/A.

6) Do you utilize the load cell shunt calibration check function on the Clayton dyno?

GM
Yes, to verify static horsepower calibration. If this check is within ±0.1 hp of the tagged reading (reading after last complete deadweight calibration), a deadweight calibration is not required. If the difference is greater than ±0.1 hp, a complete static torque cell and horsepower calibration (deadweight calibration) is performed according to the Clayton procedure.

FORD Yes - used as diagnostic to assess load cell calibration when quality control performance check is failed.

CHRYSLER
Calibration check function is recorded as part of monthly calibration but is not used by operational personnel for comparison check.

ERI The calibration check button is used every week and the indicated value is recorded as reference.

EPA Performed at cal. and logged on data sheet.

Checked for diagnostics or during monthly maintenance.

IHC, Toyo Kogyo, and Nissan do not.

7) Describe the failures, retrofits and general problems that you have encountered with respect to the Clayton road load power control circuits.

GM

The two types of components that have failed the most frequently are the multifunction generator (4302) and the triacs.

Any component failure on boards 8, 7 or 6 requires reoptimization of all three boards, because of circuit loading.

FORD (1) Found some factory supplied circuit boards to be out of calibration, (2) Frequent failure of SCR's on boards 2 & 3, (3) Drift of front roll speed reference signal, (4) Circuit boards #6, 7 & 8 come in two configurations.

CHRYSLER

No significant failures. Procedural change as noted above.

EPA

Added R/C filters; IHp is based on FRVDC. Tach. brush tension can cause high A/C ripple. B7,B8 mods., Load cells digital meters.

IHC, Toyo Kogyo, Nissan, and ERI have had no problems.

8) Highlight any solutions that you have developed to these problems.

GM

The following modifications have been added to the RLPC system to increase its reliability and maintainability:

- Added zero horsepower switch to facilitate checking the horsepower meter electrical zero level.

GM

- Replaced all original triacs with a device which has higher voltage and current ratings.

- Replaced original indicated horsepower thumbwheel switches with a unit that has only one analog output and performs decade weighting internally.

- Modified all RLPC units to be current with Clayton's latest circuit changes. These changes are as follows:

  (1) Increased resistor R14 on board 7 from 50K ohms to 200K ohms.

  (2) Removed R18 (10K ohm resistor) from board 7.

  (3) Removed R35 (10K ohm resistor) from board 6.

General Motors is currently working on an improved method of optimizing the Clayton RLPC system.

FORD

(1) Have developed calibration and repair procedures for all Clayton circuit boards, (2) Alternate source for SCR's.

CHRYSLER
See Procedural Attachment.

EPA
Exponent control could be improved.

IHC, Toyo Kogyo, Nissan, and ERI have no solutions to these problems.

TACH GENERATORS

1) What method is incorporated to assess the relationship between tach voltage and true speed?

GM
A strobotac is used monthly to synchronize the dynamometer rolls with the front and rear roll tachometer voltage outputs. These voltage levels are adjusted to read exactly what the calculated roll surface speed is for that roll diameter at 1800 rpm.

FORD

Tach. voltage vs. microprocessor (1 pulse/rev.)

CHRYSLER

Utilize a photo cell pickup into a crystal controlled counter for direct speed readout.

IHC

Strobe light per Clayton procedure.

TOYO KOGYO

Based on strobotach or digital tach generator

NISSAN

Strobe light which syncronizes to line frequency (50±0.2Hz) and pulse signal from pulse gear fitted on the roller shaft are used.

ERI

Strobe light which uses line current (60 Hz) is used. Model type: Clayton's D-6784

EPA

Count 60 tooth vs[VDC → VCO → TOTALIZED/10]

2) What tolerance do you place on this relationship?

GM

The speed signal is trimmed to read exactly the roll surface speed at 1800 rpm.

FORD

Speed meters: ± 0.1 mph, RLPC: ± 0.002 VDC

CHRYSLER
With a resolution of .01 MPH, the relationship between true speed and tachometer speed is less than .1 MPH.

TOYO KOGYO

Less than ±1%

NISSAN

±0.3 mile/h

ERI
4.632 ± 0.002 volts at 46.32 mph and 0.000 ± 0.002^{v} at 0 mph.

Roller speed is synchronized with strobe light signal (1800 rpm).

EPA

± 1.% & random   50 = 5.000 ± .010 VDC.

IHC has no tolerance.

3) Do you have limits or control for tach generator A/C ripple?
GM
RLPC system has low pass filter to reduce effects of A/C ripple. Maintenance is performed if effects are visually observed on Driver's Aid pen.

CHRYSLER
AC ripple is recorded at monthly cal on both Tach generators. Runs between 100 to 300 MV - RMS

TOYO KOGYO
Less than ± 1%

NISSAN

So adjust amplifier as less than ±0.13 mile/h on the Driver Aid.

EPA

40 mv @ 50 mph.

100 mf/5k ohms

4) What is the stability of the tach generator amplifier signal?

GM

The analog speed signal is checked against the digitally derived speed from the optical encoder prior to every highway fuel economy coast down test. These measurements must agree within the ± 0.1 mph resolution of the speed meter. No discrepancies have been noted.

FORD

Poor - generally the cause of failed performance checks.

CHRYSLER

Stable D.C. output into RLPC amplifier and to Process Computer System (PCS)

NISSAN

$\pm$ 0.5% / 8hrs

EPA

Good.

IHC, Toyo Kogyo, and ERI have no data on the stability.

5) Describe the maintenance problems you have had with tach generator circuits.

GM

No maintenance problems have been noted with the tach generator circuit. Some instability problems have been noted on the Driver's Aid pens and were found to be caused by tach generator pulley wear and misalignment.

FORD

None - stability problems are in RLPC.

CHRYSLER

Normal wear out as is expected of rotating devices. Replace very seldom

IHC, Toyo Kogyo, Nissan, ERI, and EPA have had no problems.

6) Describe in general the procedure used to zero and set the gain for the tach generator (reference signal).

GM

The output of the tachometer circuit is zeroed with the rolls stationary. Span is adjusted by substituting precision voltage supply set to the output level of the tach generator which was read at the strobe speed.

FORD

for the tach generator (reference signal). Rolls stopped, brake off, adjust B8-P3 to 0.000 ± 0.002 VDC; drive 50.0 mph by microprocessor - adjust B8-P4 to 5.000 ± 0.002 VDC

CHRYSLER

Run vehicle at 50.0 MPH (after rolls warm up) and set voltage on Fluke DVM to 5.000 VDC ± .002 (Both front and rear tachometers) Monitored zero must be 0.000 ± .002

IHC

Clayton Calibration Procedure -- Adjust zero and set gain for 4.63 volts @1,800 rpm.

NISSAN

for the tach generator (reference signal). Comparing the speed signal with pulse generator signal or the speed determined by strobe light, adjust the constant of the amplifier.

ERI

Clayton's manual has been followed.

EPA
ZERO, 1800 rpm Ext. SYNC STROBE TO SET TACH @ 4.632 VDC, then ZERO AGAIN & RESPAN

Toyo Kogyo replied N/A.

7) Have you assessed the linearity of the tach generator signal?

GM
Yes, tachometer output was checked at installation and periodically as a diagnostic. The linearity is checked at 5 mph steps from 0 to 60 mph.

FORD
Rear tach only in driver's aid calibration.

CHRYSLER
Linearity front and rear tachometers are measured monthly from 0 to 60 MPH in 5 MPH increments. If the band of tach linearity exceeds ±.1 MPH the tachometer is replaced.

TOYO KOGYO
Less than ± 1%

NISSAN
Yes, we have. 20 ± 0.3 mph, 38 ± 0.3 mph, 58 ± 0.3 mph.

ERI
23.2 mph (900 rpm) ± 0.1 mph and 69.5 mph (2700 rpm) ± 0.1 mph

are checked monthly, as well as 0, 46.32 mph, with strobe light.

EPA
Yes, very straight ± .3%

IHC replied N/A.

8) What problems have you experienced with the speed signal readouts?

GM
A couple of meters (Weston current type) failed and had to be replaced.

EPA

Some failures and excessive noises.

Ford, Chrysler, IHC, Toyo Kogyo, Nissan, and ERI have had no problems.

LOAD CELLS

1) Generally describe your calibration procedure, i.e., number of dead weight values and data recorded.

GM
The load cells are deadweight calibrated at a simulated 50 mph using the following sequence and tolerances:

Conditions	Readings: Horsepower Meter	Readings: Torque Meter
Three weights & arbor	40.7 ±0.1 hp	110.0 ±0.3 ft-lbs
Two weights & arbor	27.8 ±0.1 hp	75.1 ±0.3 ft-lbs
One weight & arbor	14.8 ±0.1 hp	40.0 ±0.3 ft-lbs
Arbor	1.9 ±0.1 hp	5.1 ±0.3 ft-lbs
None	0.0 ±0.1 hp	0.0 ±0.3 ft-lbs

FORD
Arbor plus 3-35 lb. weights - record torque & IHP

(with 50.0 mph synthetic signal) at 5 points.

CHRYSLER

Weights of 0, 5, 10, 20, 35, 70 & 105 lbs with indicated HP recorded within ± .1 HP.

IHC

Zero set plus 4 dead weight values. Record and adjust h.p.

to ± .1 at each dead weight value.

TOYO KOGYO

TOTAL WEIGHT	DEAD WEIGHT 5	10	10	35	35	INDICATE TORQUE(FT-LB)
5	x					5.0
15	x	x				15.0
25	x	x	x			25.0
40	x			x		40.0
50	x	x		x		50.0

NISSAN and ERI
dead weight values and data recorded. Fundamentally follow the Clayton manual. Dead weight --- 4(Arbor,10,10,35lb),data recorded--- record the IHP in condition of increasing & decreasing dead weights.

EPA
dead weight values and data recorded. 0 & 110 St-lbs. 11 points for zero & span.

5-90 data

2) Have you assessed the barometric sensitivity of the load cell? If so, what is the relationship?

GM
The load cells on all Clayton dynamometers have been vented, which makes them insensitive to changes in barometric pressure.

TOYO KOGYO
About 0.5 HP at 30mmHg

EPA

Yes, barometric pressure changes shift the zero up and down. (3lbs/ psi).

Ford, Chrysler, IHC, Nissan, and ERI have not.

3) What data do you have describing the stability of the load cell calibration?

GM, FORD, IHC, TOYO KOGYO, NISSAN ERI, and EPA
The calibration stability has not been determined.

CHRYSLER
Monthly calibrations show no abnormal drift or non linearity.

4) Do you have zero and span tolerances for load cell signals?

GM
Yes, the zero signal must be within ±0.1 hp of zero and the span must repeat within ±0.1 hp.

FORD
± 0.002 VDC for both

CHRYSLER
Yes -- ± .1 HP

IHC
Yes, ± .1 h.p. at each dead weight. (Torque Cell Calibration)

TOYO KOGYO
Less than 0.1 ft-lb

NISSAN and ERI
Zero tolerance --- 0.0 ± 0.0 IHP
Span tolerance --- 40.7 ± 0.1 IHP

EPA
± .002 VDC for zero and span. Zero is set as specified by Clayton to.

5) Please describe any failures and maintenance procedures which you have encountered with the dynamometer load cell.

GM
The universal couplers can wear and stick causing mechanical hysteresis.

Ford, Chrysler, IHC, Toyo Kogyo, NIssan, ERI, and EPA have had no failures.

PAU CHARACTERISTICS

1) Please describe the methods you use to characterize the power absorber unit.

GM
The X-Y-Y plot of speed versus torque and horsepower with the vehicle driven over the second cycle of the FTP schedule is used monthly to characterize the PAU. These signals are compared to theoretical curves. The plots are also examined for hysteresis to determine if tighter deadband or pulser adjustments are required.

CHRYSLER
a. At a thumbwheel setting of 10.0, IW setting of 4000 lbs., indicated horsepower is recorded at 5 mph increments up to 60 mph.

b. Speed is held constant at 50 mph with a horsepower setting of 10.0 in manual mode. Indicated horsepower is observed to verify that the horsepower setting stays constant.

TOYO KOGYO
Coastdown,Steady state(10-100Km/h),Instrumented test vehicle

(shaft-mounted torque transducers)

NISSAN The relation of front roller speed and indicated torque of the dynamometer in steady state is used.

ERI The relation of front roller speed and indicated torque of the dynamometer in steady state is used.

EPA 11 point 10-60 mph 5 mph ΔV - integrate torque/ speed then do log-log fit 25-60.

Ford and IHC replied N/A.

2) What torque and speed data are used in the characterization and how are they measured?

GM
The torque, horsepower, and front roll speed measurements are all derived internally from the RLPC system. The plots are scaled using synthetic measurement levels.

CHRYSLER
a. At a thumbwheel setting of 10.0, IW setting of 4000 lbs., indicated horsepower is recorded at 5 mph increments up to 60 mph.

b. Speed is held constant at 50 mph with a horsepower setting of 10.0 in manual mode. Indicated horsepower is observed to verify that the horsepower setting stays constant.

TOYO KOGYO
Indicate torque(pen recorder)

Load roll revolution(digital tach),(F-V converter-pen recorder)

NISSAN
how are they measured? AHP=10HP at 80 km/h(50 mph).

Speeds for steady state are 10,20,30,40,50,60,70,80,90 km/h.

We evaluate the PAU characteristic reading the torque value at each speed,

ERI
how are they measured? IW = 2750 lbs., IHP = 7.5 HP at 50 mph. Speeds for steady state are 20, 30, 40, 50, 60 mph. The data is recorded and measured by strip chart recorders.

EPA
how are they measured? Eleven data points from 10-60 mph are collected. A log-log fit of torque versus speed from 25-60 mph will show the exponent as the slope of the torque vs. speed.

Ford and IHC replied N/A.

3) Has a comparison of the manual vs automatic control exponent function been performed?

GM
Manual mode has excessive hysteresis which makes this comparison extremely difficult.

CHRYSLER
A brief comparison was done, which showed that indicated horsepower v.s. speed was the same in the manual mode as in the thumbwheel mode.

ERI
function been performed? Manual; 2.74 ~ 2.87. Automatic; 2.96 ~ 3.08 Under the conditions of trangient and steady state.

EPA
function been performed? Yes. M ≈ 2.2 - 2.3 A ≈ 1.9 - 2.1
This difference has negligible effect on 55-45 coastdown.

Ford, IHC, Toyo Kogyo, and Nissan have not.

4) What is the effect of the exponent value on vehicle loading?

GM

No actual data, but theory indicates that lower exponent would result in increased vehicle loading during FTP schedule.

EPA

Very small effect from 20-60 for same set point @ 50.

Ford, Chrysler, IHC, Toyo Hogyo, Nissan, and ERI replied N/A.

5) Does the power absorber exponent differ under transient vs steady state response?

ERI Steady state; 3.04. Acceleration (90 mph/30 sec.); 2.96. Deceleration (90 mph/30 sec.); 3.05. Average 3 times each on AUTO.

EPA

Have not quantified, but inertia loading dominates transient operation.

GM, Ford, Chrysler, IHC, Toyo Kogyo, and Nissan have no data.

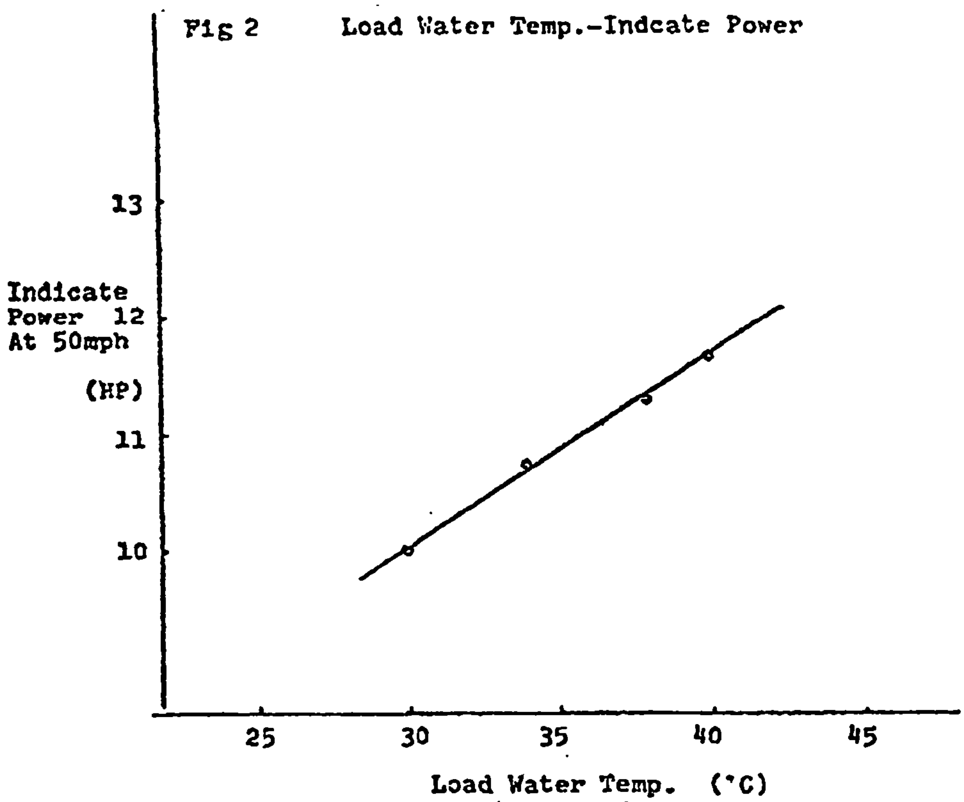
6) Has the effect of temperature setting on PAU exponent been assessed?

GM

No effect has been observed when the system is operated in the automatic mode. In the manual mode, increasing temperature results in increasing exponent.

EPA EPA uses 95 ± 5°F, used to be 135°F. Haven't analyzed or quantified affects.

TOYO KOGYO



Ford, Chrysler, IHC, Nissan, and ERI have not.

7) What is the response time of your dynamometers in changing the thumbwheel from 5 to 15 horsepower or 15 to 5 horsepower?

FORD
5 to 15 hp - 3 sec; 15 to 5 - 2 sec (observed with vehicle being driven on rolls)

CHRYSLER
We do not do this. We check for leaks as described in 1) b.

IHC
7 seconds

ERI

(95% Response Time)

#1 C/D; from 5 HP to 15 HP ---3.8 sec. from 15 HP to 5 HP ---2.5 sec.

#2 C/D; from 5 HP to 15 HP ---2.3 sec. from 15 HP to 5 HP ---2.5 sec.

EPA

8 seconds or less.

GM, Toyo Kogyo, and Nissan have no data available.

FLYWHEEL FRICTIONAL HORSEPOWER

1) Do you have frictional horsepower values for each flywheel as determined by the subtractive inertia method described in the Clayton manual?

FORD

Yes - at least one run on each of 7 cells.

CHRYSLER

The data we have on each flywheel shows the following range of frictional horsepower values:

125 and 250# wheels	0.25	up to	0.5
500# wheel	0.3	up to	0.6
trim and 1000# wheels	0.5	up to	1.0
2000# wheels	0.75	up to	1.5

Values above the maximum value listed indicate increasing bearing wear. Repeatability from month to month is approximately ± 0.1 HP for the low weights and about ± 0.2 HP for the higher weights.

TOYO KOGYO

FLYWHEEL FRICTIONAL HORSEPOWER 1) & 2)

TW	N	AVG	SIGMA
250	3	0.271HP	0.033
500	2	0.382	0.028
1000	2	0.493	0.011
2000	1	0.690	-
TRIM	8	1.082	0.026

ERI Results from #1 C/D.

845 lbs. (frim flywheel 0.829 HP), 125 lbs. (0.155 HP), 250 lbs.

(0.160 HP), 500 lbs. (0.218 HP), 1000 lbs. (0.403 HP), 2000 lbs. (0.636 HP)

EPA
Yes, see Attachment VIII

GM, IHC, and Nissan do not.

2) If multiple determinations of flywheel frictions have been made using various inertia combinations, what is the repeatability of those values?

FORD SIGMA value between 0.04 and 0.15

Prime wheel method (back-to-back tests): SIGMA equals

0.02 to 0.05.

NISSAN and ERI
Repeatability (Max. - Min.) : 0.2 HP

EPA
Generally < .05 Hp. Sometimes around .10 Hp for 2K wheels.

GM, Chrysler, and IHC replied N/A.

3) What are the transient warm-up characteristics associated with flywheel friction?

FORD FHP decreases with warmup time until

bearing temperature stabilizes.

EPA
Seem to stabilize after 10-15 minutes.

GM, Chrysler, IHC, Toyo Kogyo, Nissan, and ERI have not determined.

4) Once the dynamometer is stable, how does the warmup time constant differ with various flywheel combinations?

EPA A new flywheel combination will reach a stabilized coastdown time in about 4 minutes.

GM, Ford, Chrysler, IHC, Toyo Kogyo, Nissan, and ERI have not determined this.

5) What is the warm-up or cool-down time constant for frictional horsepower?

ERI
30 minutes at 50 mph warmup.

EPA Limited data indicates that fritional horsepower remains stable longer than it takes to reach that stable value, ie $\zeta_{wu} < \zeta_{cd}$.

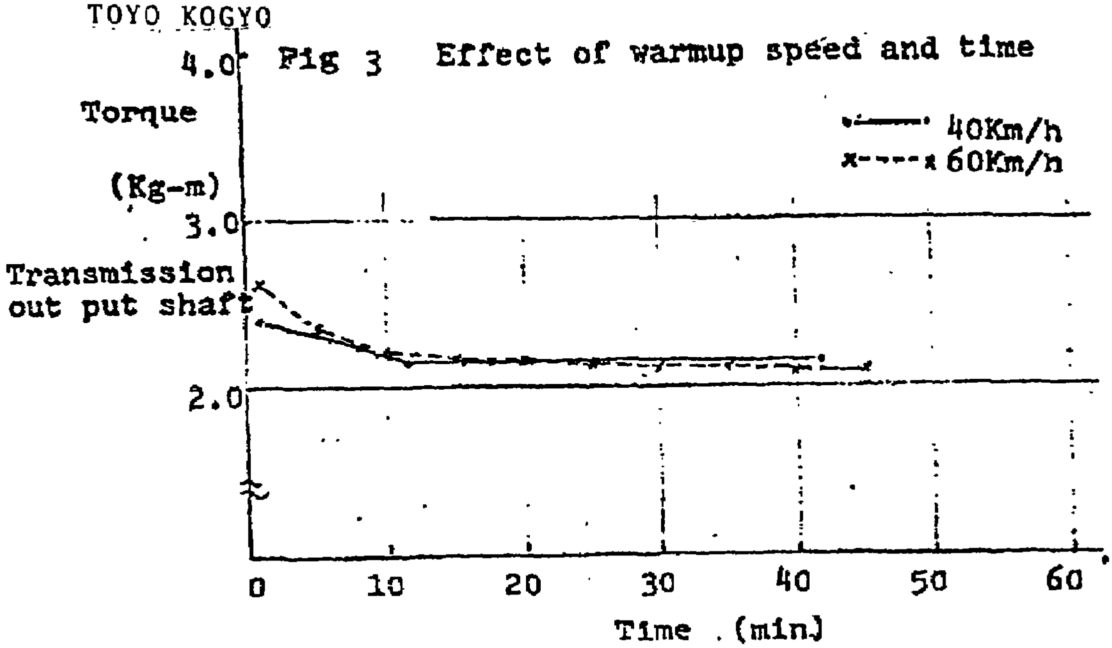
GM, Ford, Chrysler, IHC, Toyo Kogyo, and Nissan have not determined this.

6) What is the effect of warm-up speed and time on achieving stable fricitonal characteristics?

CHRYSLER
We have no data on this, but feel that 10 minutes at 50 mph is minimum needed for stability.

TOYO KOGYO

Fig 3 Effect of warmup speed and time



NISSAN

IW-5250lbs.AHP-10HP : 30min. at 50mph, 60min. at 37mph, 90min. at 25mph.

EPA

frictional characteristics? Has not been quantified, but FHp increases with the square of velocity, so higher speed should warm-up faster.

GM, Ford, IHC, and ERI have no data on this.

7) Do you detect frictional horsepower changes as a function of power absorber thumbwheel setting? If so, what magnitude?

FORD

No. Our coastdown data shows frictional horsepower is independent of thumbwheel setting, with the average slope of indicated v.s. actual horsepower very close to 1.00 over time and across all of our dynamometers. (After subtracting rear rolls inertia weight.)

CHRYSLER
No apparent change in FHP has been observed with changes in thumbwheel setting.

EPA

Analysis by(1W-155) method generally shows increasing FHp with increasing TW. New 125# dynos show this moreso than older units.

GM, IHC, Toyo Kogyo, Nissan, and ERI detected no changes.

8) What are the effects of coupling lubrication and alignment on flywheel FHP?
GM
No data available. However, all dynamometer shafts are leveled and aligned with each other to within ±0.005 inches. At the present time, the couplings are lubricated every three to six months.

CHRYSLER
No effects have been identified although in one case, misalignment caused bearing failure within a short period of time. The coupling itself failed also.

EPA
flywheel FHP? Coupling misalignment could cause increased FHp as a function of PAU load, but has not been verified.

Ford, IHC, Toyo Kogyo, Nissan, and ERI have no data.

9) How long does it take to stabilize the frictional value for a new bearing?
GM
This has not been defined by any engineering study; however, our experience indicates that a minimum of 72 hours of steady state or test cycle driving is sufficient to stabilize the friction of new bearings.

FORD
All new bearings - 10 to 15 hours

CHRYSLER
No data available. We use an arbitrary time of 24 hours at 50 mph.

NISSAN
Roughly ____ , 50 hrs at 31 mile/h, ____ . 32 hrs at 50 mile/h

EPA EPA operates dyno for 8 hours at 30-50 mph. Spot checks of the 6875 lW coastdown time are made.

IHC, Toyo Kogyo, and ERI have not measured this.

10) What problems or FHP differences have been detected for C3 vs standard fit bearings?

GM
We have not experienced any C3 bearing failures. The C3 bearings have less friction than the standard fit bearings.

EPA
C3 bearings do not have significantly lower FHp. Failure rates are about equal.

Ford, Chrysler, IHC, Toyo Kogyo, Nissan, and ERI replied N/A.

11) How does flywheel friction vary with dynamometer speed (function and intercept)?

GM
Limited data indicates flywheel frictional horsepower to be a function of speed raised to a power of 2.2. Torque intercept may range from 0.3 to 1.5 ft-lbs.

FORD
1976 study shows friction torque linear with speed

NISSAN
Equivalent friction force f

$f = 0.28334\ V + 1.7740$ V = mph, f = lb

ERI
and intercept)? Flywheel friction (torque) is a linear curve with dynamometer speed. $T = 8.089 \times 10^{-2} S + 0.7603$ (T -- lb. ft, S -- mph)

IW=2750lbs

EPA
and intercept)? Flywheel FHp increases with roll velocity squared.

Static torque is less than 0.2 FT-LB.

Chrysler, IHC, Toyo Kogyo replied N/A.

12) What percentage of total frictional horsepower can be attributed to aerodynamic drag?

GM
Theoretical analysis indicates that the aerodynamic drag contribution to total frictional horsepower ranges from 25 to 70% depending upon selection of inertia wheels.

EPA
Data from the bearing manufacturer and a theoretical analysis indicated about 60-75% of total FHp comes from aerodynamic drag on the flywheel.

Ford, Chrysler, IHC, Toyo Kogyo, Nissan, and ERI had no data on this.

ROLL DIAMETER AND SPACING

1) What measurement techniques do you have for measuring roll diameter?

GM
At installation the diameter of all rolls were measured at five locations using a micrometer and the concentricity (run-out) was measured at three locations. The rolls are checked periodically using a fifth wheel and digital odometer. The rolls are rotated 100 revolutions and the distance traveled is measured. From this measurement, the diameters can be calculated and compared to the previously determined values. Periodically the rolls have been examined with an indicator which is slid along the roll surface to determine if any wear pattern can be detected.

FORD, NISSAN, and ERI
Rear roll diameter was measured with micrometer.

CHRYSLER
Nominal diameter is used.

2) What is the wear rate for machine rolls?

GM
No measurable wear has been detected over an 18-month period.

NISSAN
Max. 0.09% (roller dia.) / 5 years

Ford, Chrysler, IHC, Toyo Kogyo, ERI, and EPAdo not use machine rolls.

3) What statistics do you have for actual distance measurements?

GM
Table 2 contains statistics of actual distance measurements for 1978 Certification tests at the General Motors Milford-Vehicle Emission Laboratory.

CHRYSLER
No statistics were ever tabulated, although correlation testing with EPA shows very close agreement.

GM

Table 2

Actual Distance Measurements
from 1978 Certification Tests at M-VEL

FTP
Phase 1

Mean	% Diff*	Std. Dev.	Min	Max
3.592	0.06	0.016	3.53	3.65

Theoretical Distance - 3.59 Miles

Phase II

Mean	% Diff	Std. Dev.	Min	Max
3.860	0.0	0.022	3.79	3.93

Theoretical Distance - 3.86 Miles

Phase III

Mean	% Diff	Std. Dev.	Min	Max
3.593	0.08	0.016	3.55	3.66

Theoretical Distance - 3.59 Miles

HWFE

Mean	% Diff	Std. Dev.	Min	Max
10.255	0.1	0.038	10.04	10.39

Theoretical Distance - 10.242 Miles

* $\%Diff = \frac{Actual-Theoretical}{Theoretical} \times 100$

EPA

	$\bar{x}$	S.D.	% CV				
Bag 1	3.5934	.0241	.67%	Bag 3	3.5947	.0221	.61%
Bag 2	3.8899	.0349	.89%	HWFE	10.259	.049	.48%

Ford, IHC, Toyo Kogyo, Nissan, and ERI have no statistics.

4) What tolerance do you have on distance measurements for phases 1, 2, 3, and HWFET?

GM

The actual distance traveled tolerances for an FTP schedule are listed below:

- o Phases I & III - 3.591 mi ±2% (3.519 to 3.663 mi)
- o Phase II - 3.859 mi +2% (3.782 to 3.936 mi)

The actual distance traveled tolerance for a HWFET schedule is ±2% of 10.242 mi (10.037 to 10.447 mi).

FORD Phases 1 & 3: 3.51 to 3.69 miles

Phase 2: 3.77 to 3.96 miles

HWFET: 9.99 to 10.50 miles.

CHRYSLER

We use the tolerances EPA uses which are:

Phase 1 - 3.606 ± 2%
Phase 2 - 3.901 ± 2%
Phase 3 - 3.607 ± 2%
HWFE - 10.295 ± 2%

IHC

Calculated distance from roll revs must be within ± 2%.

TOYO KOGYO

Approximately 2% of the theoretical mileage

NISSAN
± 0.2 mile for phase 1 , 3

± 0.3 mile for phase 2 , HWY

ERI + 2.0% for each phase value, CT: 3.6 mile,

CS: 3.9 mile, H.T.: 3.6 mile, HWYFET: 10.242 mile.

EPA

Bag 1 3.52 - 3.66 Bag 3 3.52 - 3.66

Bag 2 3.83 - 3.99 HWFE 10.04 -10.45

5) What is the effect of roll spacing on actual loading and tire slip rate (both steady state and transient) and on quick check coast downs?

GM
No data available on new Clayton dynamometer systems.

IHC
Vehicle coastdown on 20" spacing vs. 17" spacing is within 7% criteria.

EPA
N/A All EPA rolls are 17.25" spacing.

Ford, Chrysler, Toyo Kogyo, Nissan, and ERI have no data.

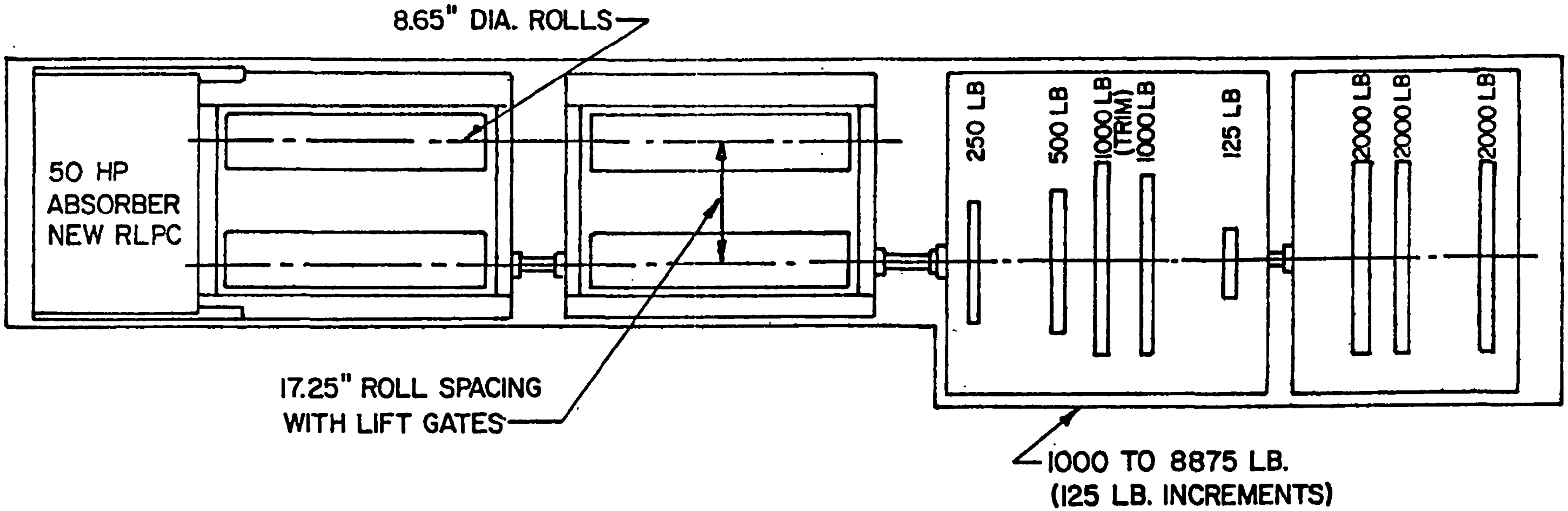
6) What value for rear roll inertia and frictional horsepower do you obtain in your laboratory? Describe the roll configuration.

GM
See Figure 2 for M-VEL's roll and inertia weight configuration.

Frictional horsepowers were calculated using Clayton's nominal rear roll inertia of 77.5 pounds. Frictional horsepower per split roll for all sites ranges from 0.07 to 0.17 hp. The total rear roll frictional horsepowers range from 0.16 to 0.32 hp.

FIGURE 2

CLAYTON CTE-50 CONFIGURATION



FORD

155 lb. for all rear roll inertias

3 cells are ECE-50; 4 cells are CTE-50.

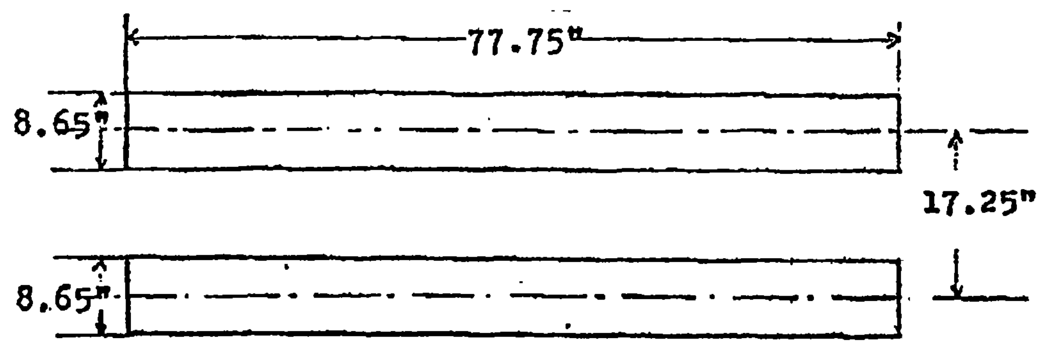
CHRYSLER

Configuration is Clayton CPE-50. Rear rolls inertia is 155 lbs. (per Clayton).

TOYO KOGYO

Inertia; 155lb, FHP; about 0.1 HP at 50mph

Roll configuration; Fig.4



NISSAN and ERI

Rear inertia is 155 lbs. (from Clayton's manual) Average frictional horsepower is 0.253 HP (including roll revollution counter) on ECE-50 model arrangement B.

EPA has Clayton configuration (B) with a single roll. Assigned IW = 155 lbs.

IHC replied N/A.

7) Describe the coast down repeatability and statistics for rear roll friction across all dynamometers.

GM

Rear roll coast down technique repeatability, as measured on two dynamometers, was within ± 0.006 hp.

FORD

CLAYTON DYNAMOMETER REAR ROLL FRICTION STUDY

CELL NUMBER 1
NUMBER OF REAR ROLLS 2

55 TO 45 TIME (SEC)		RETARD FORCE (LB)	HORSEPOWER
57.80		1.21	0.16
59.70		1.17	0.16
53.00		1.32	0.18
54.19		1.29	0.17
60.60		1.15	0.15
51.70		1.36	0.18
54.60		1.28	0.17
64.00		1.10	0.15
60.80		1.15	0.15
		----	----
	AVERAGE	1.23	0.16
	STD DEV	0.09	0.01

CELL NUMBER 2
NUMBER OF REAR ROLLS 2

55 TO 45 TIME (SEC)		RETARD FORCE (LB)	HORSEPOWER
48.29		1.45	0.19
41.20		1.70	0.23
51.70		1.37	0.18
		----	----
	AVERAGE	1.51	0.20
	STD DEV	0.17	0.02

CELL NUMBER 3
NUMBER OF REAR ROLLS 1

55 TO 45 TIME (SEC)		RETARD FORCE (LB)	HORSEPOWER
41.00		1.69	0.23
40.25		1.72	0.23
46.20		1.50	0.20
43.86		1.58	0.21
36.20		1.91	0.26
46.50		1.49	0.20
46.61		1.49	0.20
45.00		1.54	0.21
45.70		1.51	0.20
45.74		1.51	0.20
48.10		1.44	0.19
[illegible]		1.55	0.21
		----	----
	AVERAGE	1.53	0.21
	STD DEV	0.13	0.02

FORD

CELL NUMBER 4
NUMBER OF REAR ROLLS 2

55 TO 45 TIME (SEC)		RETARD FORCE (LB)	HORSEPOWER
38.80		1.81	0.24
32.00		2.19	0.29
39.42		1.78	0.24
37.50		1.87	0.25
42.50		1.65	0.22
44.30		1.58	0.21
38.20		1.84	0.24
36.95		1.90	0.25
37.40		1.88	0.25
		----	----
	AVERAGE	1.83	0.24
	STD DEV	0.17	0.02

CELL NUMBER 5
NUMBER OF REAR ROLLS 1

55 TO 45 TIME (SEC)		RETARD FORCE (LB)	HORSEPOWER
45.00		1.54	0.21
41.70		1.66	0.22
44.24		1.54	0.21
41.30		1.68	0.22
42.10		1.64	0.22
39.90		1.74	0.23
		----	----
	AVERAGE	1.63	0.22
	STD DEV	0.08	0.01

CELL NUMBER 6
NUMBER OF REAR ROLLS 1

55 TO 45 TIME (SEC)		RETARD FORCE (LB)	HORSEPOWER
34.70		2.00	0.27
33.16		2.09	0.28
34.87		1.99	0.26
30.88		2.24	0.30
35.50		1.95	0.26
27.50		2.52	0.34
		----	----
	AVERAGE	2.13	0.28
	STD DEV	0.22	0.03

FOR ALL TH[illegible]		----	----
	AVERAGE	1.65	0.22
	STD DEV	0.31	0.04

CHRYSLER
Range of values across 8 dynamometers for four months was .095 to .203, with a mean value of .139. Maximum range of variation for one dynamometer during this time was .150 to .203. With the exception of two data points, frictional horsepower was within .15 ± .05.

ERI
Results of dynamometers were 0.252 HP and 0.254 HP. Repeatability at each dynamometer is + 0.8%.

EPA
friction across all dynamometers. Repeatability during calibration should be ± .5 seconds from the average of about 50-60 seconds. Longterm changes in the nominal coastdown time shoud be less than 5 seconds.

CALIBRATION MONITORING PRACTICES

1) What pre-test horsepower check procedure and tolerance are used in your laboratory?

GM
Prior to each Certification test, a non-Certification vehicle is used to set the dynamometer horsepower. Using the displayed indicated horsepower and the calibration curves, the actual horsepower must be within ±0.5 hp of the desired setting.

FORD
Drive front roll 50 mph & read IHP: must be within ± 0.3 of thumbwheel.

CHRYSLER
Not formally done. Any discrepancy exceeding ± 0.2 horsepower between thumbwheel and indicated readings at 50 mph will be resolved before the rolls is used for further testing.

IHC
Vehicle is operated at 50 mph and horsepower checked to ± .5 h.p.

TOYO KOGYO

A) Select the inertia. B) Check the power meter (zero,span)
C) Set the warm up car. D) Warm up (15min. at 30mph)
E) Set the proper horsepower for the test vehicle at the front roll speed 1943rpm(50mph) with manual sw.
F) Accelerate the dynamometer to 60mph and return to 50mph. Verify that the indicate power within ±0.5HP of the set power.

NISSAN and ERI Indicated horsepower value is checked at 50 mph before each test. Tolerance : ± 0.1 HP

EPA The thumbwheel value is verified at PAU 50. Acceptance criteria are ± .5Hp, but properly adjusted dyno circuits can maintain ± .3 Hp.

2) If multiple quick checks are performed to confirm road load horsepower, what tolerance do you place on repeatability?

FORD

3 repetitions: highest-lowest times less than 0.3 sec.

CHRYSLER

Three repetitions within ± 0.2 seconds are required. A fourth repeat is performed if the first three are more than 0.2 seconds apart.

NISSAN

± 0.1 HP

EPA

Data show that the range (max-min) of 3 loaded quickcheck times will be less than .1 seconds on the average. Range values greater than .3 are abnormal.

GM, IHC, Toyo Kogyo, and ERI do not perform multiple quick checks.

3) If a weekly dynamometer verification procedure is performed, please describe it in general (IWs checked, warm-up procedure and tolerance).

GM

Only monthly dynamometer calibrations are being performed.

FORD 30 minute warmup; coastdowns at three inertia weights - must be within ± 1 sec. or ± 0.5 hp whichever is smaller; slope at highest inertia 1.000 ± 0.015. IHP within ± 0.3 of thumbwheel.

CHRYSLER

3) and 4) A weekly vehicle crosscheck is performed at 4000 lbs. inertia weight and 13.2 horsepower. Duplicate hot 505 tests are perforemd on each rolls. Any significant changes in dynamometer loading would be identified by changes in the CO_2 emissions.

TOYO KOGYO

A) Set the vehicle lifting device and the coastdown timer.
B) Check the power meter. C) Set the check car.
D) Select the inertia 2875lb or (Trim+500+250+125lb)
E) Warm up · F) Set the horsepower 8HP at 50mph (front roll speed)
F) Accelerate the dynamometer to 60mph and lift up the car from the roll Measure the coastdown time of 55-45mph
G) Verify that the coastdown time is within ±1 second of the latest calibration data.
H) Set the inertia 3000lb or (Trim+1000lb). Repeat step E) to G).
I) If the coastdown times differ by more than ±1sec., a new calibration is required.

NISSAN Torque meter check : warmup--30mph x 40min, check 0 & 50lb-ft point, dead weight tester 5, 15, 25, 40, 50, lb-ft (tolerance: 0.3lb-ft)

Speed meter check : by strobe, tolerance ± 0.3 mile/h.

EPA

See TP-302.

IHC and ERI replied N/A.

4) Describe any special correlation test performed to assure dynamometer correlation and stability.

GM

As a part of the dynamometer monthly calibration, hysteresis checks are made in the automatic mode and stability checks in the manual mode.

The stability check consists of an 18-cycle FTP schedule driven at an indicated horsepower of 12.0. The starting and ending indicated horsepower readings must agree within ± 0.5 hp.

The hysteresis check is performed by driving a vehicle at 50 mph front roll speed while the thumbwheel settings are cycled to obtain the following sequence of indicated horsepower readings, within the tolerance of ± 0.4 hp: 10.0, 12.0, 10.0, 8.0, and 10.0. The middle and last reading at the 10.0 hp setting must agree within ± 0.2 hp.

FORD

mometer correlation and stability. Loaded vehicle coastdowns weekly on every cell; breakaway force: front & rear rolls; rear roll coastdown.

Refer to Chrysler's answer to question #3 in this section.

EPA

mometer correlation and stability. Repeatable vehicle is used and loaded quickcheck times are also analyzed between sites. Alternate PAU determinations have been performed across dynos at times.

IHC, Toyo Kogyo, Nissan, and ERI do not perform any special tests.

5) What control charts or trend analyses are performed using the verification data?

GM

Tolerances are described in 4) above.

FORD

Plot all of above.

CHRYSLER

Weekly crosscheck data is kept for six months back, but no formal trend analysis is performed since any trends would likely be caused by vehicle changes.

EPA

See TP-302.

IHC, Toyo Kogyo, Nissan, and ERI replied N/A.

6) What techniques have you developed to correlate road load data to quick check coast down times?

GM

Quick check coast downs are not performed.

CHRYSLER

Road load data is correlated to rolls coastdown times as required by EPA in A/C #55B.

EPA

quick check coastdown times? If AHp is calculated from the quick check coastdown time and plotted versus PAU TW value, a tight band of data is observed and can be used to spot outliers.

Ford, IHC, Toyo Kogyo, Nissan, and ERI replied N/A.

SITE CORRELATION TESTS

1) If your facility uses a repeatable car, what is the variation in the coast down time across dynamometers?

GM

Repeatable car is not used to gather coast down data.

FORD

± 5.35%

CHRYSLER

Coastdowns are not normally done. Our tolerance is a coefficient of variation of 2.0% or less on CO_2 emissions. (This is sometimes exceeded.)

IHC

Maximum variation at 50 mph across dynamometers is 5 to 6%.

TOYO KOGYO

55-45mph coastdown time(repeatable car)

IW=2750lb AHP=9.9HP N=4

AVG TIME=11.36sec. SIGMA=0.075

ERI

the coastdown time across dynamometers? At IW = 2750 lbs. AHP = 9.9 HP,

#1 C/D; $\bar{x}$ = 12.765 sec. σ = 0.088 at recent 12 data.

#2 C/D; $\bar{x}$ = 12.842 sec. σ = 0.067 at recent 9 data.

EPA

the coastdown time across dynamometers? The vehicle used has an average time of 12.1 seconds. Overall variation from day to day is about ± 2%. Average differences dyno to dyno are normally less than ± 1.5%

Nissan replied N/A.

2) What is the variability in the road load determination from dyno to dyno?

GM

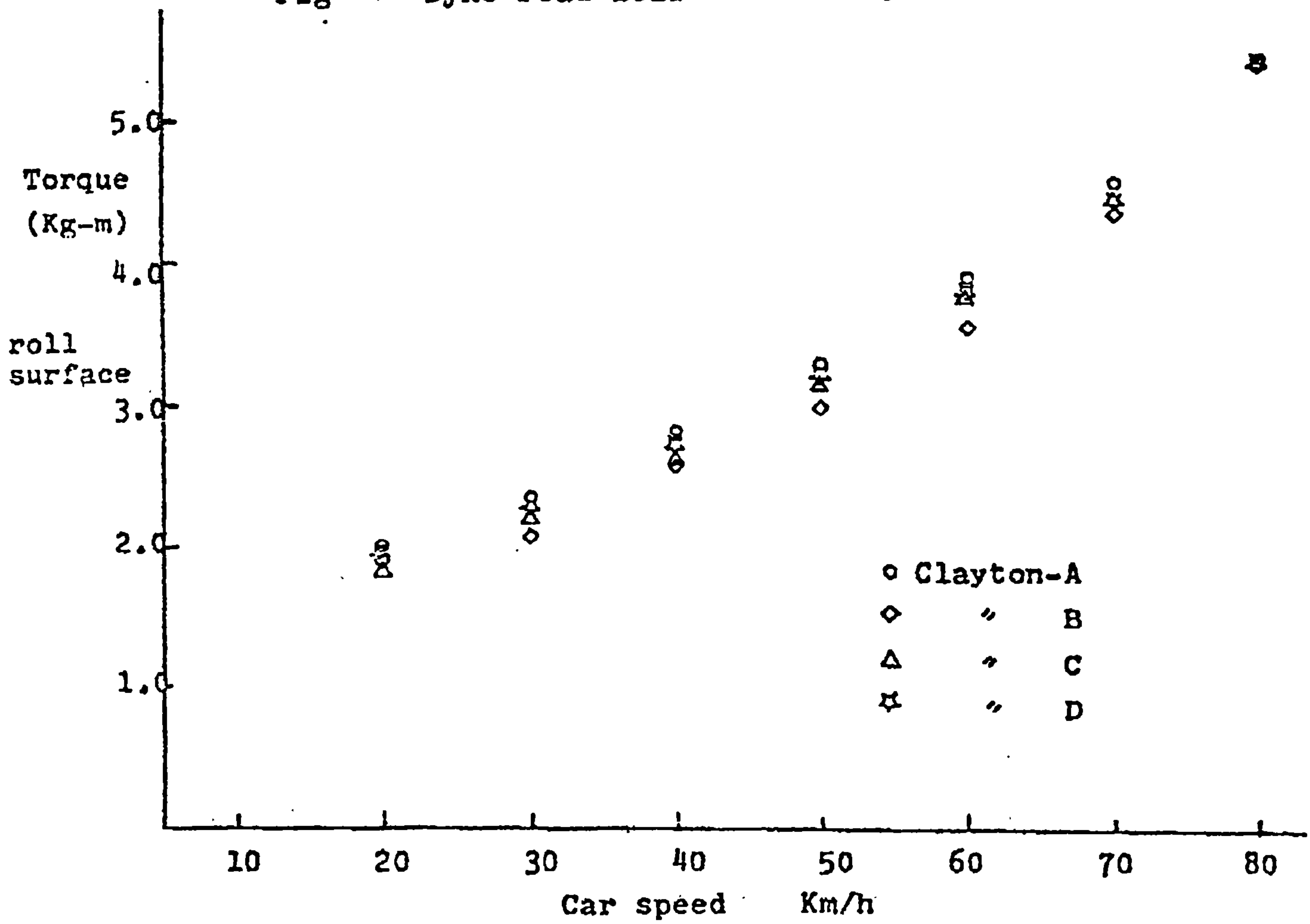
Investigations have indicated that the variability, defined as 2.77 times the standard deviation divided by the mean road load horsepower, is approximately 10 per cent.

CHRYSLER

The dynamometers vary slightly depending on the last calibration performed. Since they are recalibrated each month, the offset of a particular dynamometer is temporary, and they are all treated as being equal. Several dynamometers are averaged to generate road load horsepower settings.

TOYO KOGYO

Fig 5 Dyno road load variability



ERI The difference between two dynamometers is 1.7～3.3%,

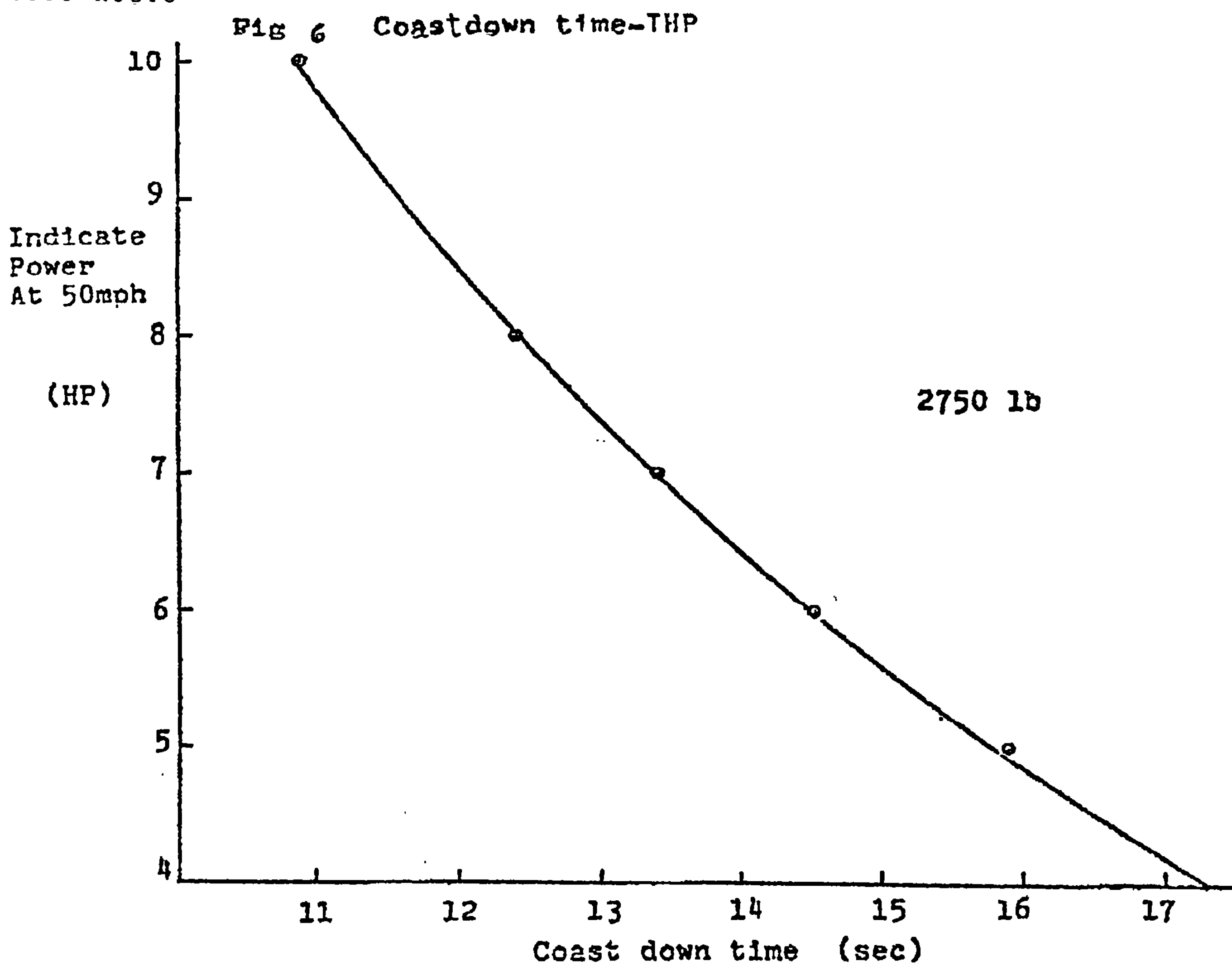
When IW = 2500～3000 lbs., AHP = 9～11 HP Class vehicles are used.

Ford, IHC, Nissan, and EPA replied N/A.

3) What is the sensitivity of the quick check coast down time to the horsepower setting?

CHRYSLER
This varies from vehicle to vehicle.

TOYO KOGYO



ERI
0.09 sec./0.1 HP at IW = 2750 lbs., AHP = 9.9 HP.

EPA A 7% change in coastdown time equates to a 15% change in PAU, since the FHp of the drive train is the dominant part of total FHp.

GM, Ford, IHC, and Nissan had no data on this.

4) How much friction in the quick check procedure is attributable to the vehicle dyno combination (i.e., dyno bearing friction vs total frictional)?

ERI About 15 % (i.e., dyno bearing friction is 2.0 HP, total friction is 13.2 HP).

EPA For one vehicle the total friction was 10 Hp = 2 FHp dyno + 8 FHp vehicle.

GM, Ford, Chrysler, IHC, Toyo Kogyo, and Nissan have no data on this subject.

SPEED SIGNAL FOR DRIVING TEST

1) Please describe any studies performed to characterize the effect of tire slip on test results.

No response to this question.

2) Has test variability been assessed using coupled rolls vs front or rear roll driving trace signal?

IHC

Front roll driving trace introduces large error due to tire slippage.

GM, Ford, Chrysler, Toyo Kogyo, Nissan, ERI, and EPA have not done any assessment of test variability.

AHP VS T*S
AHP VS TW

FRICTIONAL HORSEPOWER

DYNO	DATE	2000(2)	2000(1)	1000	500	250	125	TRIM	TRIM	REAR ROLL	
D001											
T*S	5/15/78	0.797	0.744	0.458	0.291	0.155	0.171		0.863	0.119	
TW	11/20/78	0.738	0.729	0.467	0.271	0.182	0.203	0.815	0.792	0.106	1750-5500
			.57	.51	.34	.19		1.19			← 250# IW CONFIG.
D002											
TW	5/17/78	0.623	0.674	0.485	0.288	0.208	0.201		0.611	0.159	
T*S	9/19/78	0.730	0.681	0.419	0.293	0.182	0.167		0.709	0.172	
T*S	12/15/78	0.646	0.654	0.439	0.325	0.189	0.187	0.722	0.734	0.166	
			.63	.45	.29	.25		1.32			
D003											
T*S	3/10/78	0.789	0.661	0.448	0.239	0.187	0.128		0.834	0.203	
TW	7/11/78	0.730	0.611	0.438	0.244	0.201	0.138		0.746	0.108	
T*S	7/11/78	0.853	0.697	0.455	0.250	0.192	0.151		0.680		
TW	7/13/79	0.650	0.515	0.434	0.282	0.165	0.150	0.641	0.609	0.128	
T*S	7/13/79	0.764	0.614	0.445	0.284	0.171	0.150	0.607	0.609		
			.60	.40	.23	.16		1.10			
D004											
T*S	3/7/78	0.980	0.702	0.508	0.290	0.225	0.127	N/A	1.040	0.077	
T*S	4/4/78	0.830	0.646	0.454	0.240	0.181	0.097		0.801	0.076	
TW	5/30/78	0.559	0.661	0.435	0.291	0.216	0.139		0.724	0.078	
			.47	.47	.31	.17		1.36			
D005											
T*S	1/13/78	0.759	0.795	0.509	0.325	0.259	0.136		0.696	0.144	
T*S	8/29/78	0.537	0.711	0.435	0.389	0.212	0.135		0.670	0.157	
T*S	3/16/79	0.822	0.682	0.463	0.432	0.225	0.148	0.715	0.721	0.159	
			.62	.41	.22	.20		1.19			
D006											
T*S	1/9/78	0.641	0.704	0.455	0.225	0.158	0.119		0.755	0.139	
T*S	8/22/78	0.675	0.555	0.560	0.247	0.204	0.124		0.843	0.120	
T*S	3/22/79	0.671	0.537	0.501	0.219	0.212	0.114	0.989	1.013	0.183	
			.48	.42	.23	.11		1.06			

ATTACHMENT III

08-22-79 JAH

AHP vs T*S
AHP vs TW

FRICTIONAL HORSEPOWER

DYNO	DATE	2000(2)	2000(1)	1000	500	250	125	TRIM	$\overline{TRIM}$	REAR Roll	11	12	13
D 007													
TW	8/29/78	0.632	0.646	0.498	0.306	0.258	0.234		1.346	0.114			
T*S	8/29/78	0.700	0.738	0.521	0.316	0.272	0.260		1.309				
D 207													
T*S	3/20/78	0.807	0.727	0.490	0.238	0.193	0.144		0.972	0.137			
TW	6/13/78	0.773	0.732	0.486	0.251	0.182	0.175		1.002	0.277			
T*S	6/13/78	0.861	0.819	0.528	0.261	0.182	0.165		0.911				
D 208													
T*S	10/30/78	0.651	0.631	0.452	0.196	0.163	0.173		1.347	0.159			
TW	10/30/78	0.573	0.619	0.463	0.192	0.170	0.206		1.285				
T*S	6/1/79	0.628	0.601	0.403	0.345	0.291	0.228	0.872	0.920	0.144			
D 209													
TW	9/13/78	0.796	0.674	0.460	0.287	0.156	0.232		0.869	0.331			
TW	8/9/79	0.685	0.576	0.482	0.237	0.177	0.218	1.208	1.236	0.358			
T*S	8/9/79	0.733	0.576	0.452	0.235	0.172	0.205	1.342	1.374				
MIN		0.537	0.515	0.403	0.192	0.155	0.097	0.607	0.609	0.076			
MAX		0.861	0.819	0.560	0.432	0.291	0.260	1.342	1.374	0.358			
MEAN		0.714	0.664	0.468	0.276	0.198	0.168	0.879	0.901	0.159			
TAN. DEV.		0.0886	0.0724	0.0349	0.0526	0.0350	0.0415	0.2552	0.2421	0.0719			
MEAN FOR 250# DYNO			.56	.44	.27	.18		1.20					

IW = 1750 - 5500

ATTACHMENT V

```
DYNAMOMETER SITE: D003
    DYNO EPA PID: N/A
CALIBRATION MODE: AUTO
CALIBRATION DATE: 07-12-79
CALIBRATION TIME: 09:00:00
     OPERATOR ID: 30900
       BAROMETER: 29.15
     SPEED RANGE: 55.00-45.00 MPH
   DATA LINE NO.:  2736.000
    CAL CHECK HP:  0.0
   REAR ROLL FHP:  0.128 HP
 AVG. RR DELTA T: 73.32
        COMMENTS: GENERAL 125LB. CALIB TO US FOR CERT WITH 600 CFM-CVS FR&RR RESET7-13 6500REDON
```

```
********************************************
****    DYNAMOMETER CALIBRATION AND     ****
****                                    ****
****    COASTDOWN   DATA   ANALYSIS     ****
********************************************
```

```
DDDDDD      000000     000000    33333333
 DD  DD   00     00  00     00  3       33
 DD   DD  0       0  0       0       33333
 DD  DD   00     00  00     00  3       33
DDDDDD      000000     000000    33333333
```

```
PROCESSED: AUG  9, 1979  08:44:35
  VERSION: 1.23
  OPTIONS:  TW VS AHP,FULL IW,CERT TBL
```

SPEED CALIBRATION / **TORQUE CALIBRATION** / **POWER ABSORBER CURVE DATA**

SPD MTR MPH	FRNT TACH VDC	REAR TACH VDC	TRUE SPEED RPM	FRNT SPD MPH	FRNT VDC% DIFF	DEAD WGHT FTLB	LOAD CELL VDC	TORQ CNTS HZ	TORQ VDC% DIFF	REAR SPD MPH	SPEED COUNTS	TORQUE COUNTS	TIME SECS	AVG SPEED MPH	AVG TORQ FTLB	CALC IHP	PAU CURVE FTLB	PAU CURVE %DIFF
10.	1.00	1.00	387.	10.0	0.29	5.0	0.189	189.	2.77	10.0	3971.	693.	10.250	9.97	1.87	0.14	0.99	89.47
15.	1.50	1.50	593.	15.0	-0.14	10.0	0.367	366.	-0.51	15.0	5828.	1219.	10.099	14.85	3.31	0.36	2.24	47.72
20.	2.00	2.00	776.	20.0	0.03	15.0	0.551	551.	-0.52	20.0	7764.	1776.	10.048	19.88	4.82	0.71	4.08	18.31
25.	2.50	2.50	968.	24.9	0.24	20.0	0.739	738.	0.01	25.0	9692.	2332.	10.064	24.78	6.31	1.16	6.41	-1.52
30.	2.99	3.00	1160.	29.9	0.04	25.0	0.919	917.	-0.53	30.0	11886.	3510.	10.279	29.76	9.28	2.04	9.33	-0.53
35.	3.48	3.50	1350.	34.7	0.05	30.0	1.109	1108.	0.01	35.0	13664.	4819.	10.160	34.61	12.87	3.30	12.72	1.20
40.	3.98	4.00	1540.	39.6	0.31	40.0	1.485	1483.	0.41	40.0	15706.	6371.	10.237	39.48	16.87	4.93	16.67	1.24
45.	4.46	4.50	1733.	44.6	-0.11	50.0	1.845	1843.	-0.21	45.0	17594.	8091.	10.203	44.37	21.49	7.06	21.18	1.44
50.	4.95	5.00	1921.	49.4	0.01	60.0	2.225	2223.	0.27	50.0	19473.	9935.	10.165	49.30	26.48	9.66	26.29	0.71
55.	5.44	5.50	2114.	54.4	-0.12	75.0	2.768	2766.	-0.21	55.0	21184.	11767.	10.087	54.04	31.59	12.63	31.75	-0.50
60.	5.91	6.00	2295.	59.1	-0.05	90.0	3.331	3329.	0.06	60.0	23454.	14049.	10.238	58.95	37.15	16.21	37.95	-2.10

```
FRONT VDC = 0.1001*MPH ( 5.006 @50)    FT-LB= 0.0270*HZ + 0.0450    PAU CURVE FOR 4000. LB INERTIA WEIGHT    TORQ= K * (N**M)
 REAR VDC = 0.1600*MPH ( 5.000 @50)     VDC = 0.0370*FTLB-0.0011    BETWEEN 25 AND 60 MPH                       K = 0.8817E-02
                                                                                                                M = 2.65
```

COASTDOWN DATA

IW LBS.	DELTA T	TW HP	CALC AHP	CALC FHP	LLSQ TW	%DIFF TW	LLSQ TW =M*AHP+B M	B
1000.	14.462	3.0	4.20	1.20	2.96	1.513	0.8333	-0.5440
	7.639	6.0	7.95	1.95	6.08	-1.328		
	4.816	10.0	12.61	2.61	9.96	0.361		
1125.	15.854	3.0	4.31	1.31	2.97	1.132	0.8589	-0.7350
	8.636	6.0	7.91	1.91	6.06	-0.991		
	5.480	10.0	12.47	2.47	9.97	0.266		
1250.	17.884	3.0	4.24	1.24	2.98	0.739	0.8705	-0.7171
	9.781	6.0	7.76	1.76	6.04	-0.649		
	6.176	10.0	12.29	2.29	9.98	0.171		

CERTIFICATION PRINTOUT FOR EXAMPLE USED IN TP-202 DATED 7/27/79.

IW LBS.	DELTA T	TW HP	CALC AMP	CALC FHP	LLSQ TW	%DIFF TW	LLSQ TW =M*AMP+B M	B
1375.	19.021	3.0	4.39	1.39	2.99	0.476	0.8848	-0.8985
	10.671	6.0	7.83	1.83	6.03	-0.419		
	6.786	10.0	12.31	2.31	9.99	0.109		
1500.	21.265	3.0	4.28	1.28	2.99	0.285	0.8855	-0.8017
	11.833	6.0	7.70	1.70	6.02	-0.249		
	7.472	10.0	12.19	2.19	9.99	0.065		
1625.	22.482	3.0	4.39	1.39	2.97	1.061	0.8947	-0.9587
	12.586	6.0	7.84	1.84	6.06	-0.930		
	8.075	10.0	12.22	2.22	9.98	0.249		
1750.	24.129	3.0	4.40	1.40	2.98	0.567	0.9078	-1.0154
	13.694	6.0	7.76	1.76	6.03	-0.497		
	8.769	10.0	12.12	2.12	9.99	0.131		
1875.	25.120	3.0	4.53	1.53	2.97	1.054	0.9115	-1.1629
	14.377	6.0	7.92	1.92	6.06	-0.925		
	9.318	10.0	12.22	2.22	9.98	0.247		
2000.	27.948	3.0	4.35	1.35	2.98	0.603	0.9133	-0.9871
	15.804	6.0	7.69	1.69	6.03	-0.529		
	10.109	10.0	12.02	2.02	9.99	0.139		
2125.	28.769	3.0	4.49	1.49	2.98	0.638	0.9211	-1.1508
	16.545	6.0	7.80	1.80	6.03	-0.558		
	10.674	10.0	12.09	2.09	9.99	0.147		
2250.	30.427	3.0	4.49	1.49	2.99	0.257	0.9209	-1.1431
	17.582	6.0	7.77	1.77	6.01	-0.225		
	11.298	10.0	12.09	2.09	9.99	0.059		
2375.	31.264	3.0	4.61	1.61	2.98	0.795	0.9319	-1.3228
	18.250	6.0	7.90	1.90	6.04	-0.697		
	11.890	10.0	12.13	2.13	9.99	0.184		
2500.	33.225	3.0	4.57	1.57	2.98	0.832	0.9302	-1.2755
	19.295	6.0	7.87	1.87	6.04	-0.729		
	12.547	10.0	12.10	2.10	9.98	0.193		
2625.	33.888	3.0	4.70	1.70	2.98	0.651	0.9385	-1.4345
	20.032	6.0	7.96	1.96	6.03	-0.571		
	13.102	10.0	12.17	2.17	9.98	0.151		
2750.	35.597	3.0	4.69	1.69	2.97	0.957	0.9399	-1.4379
	20.960	6.0	7.97	1.97	6.05	-0.840		
	13.750	10.0	12.15	2.15	9.98	0.224		
2875.	35.976	3.0	4.85	1.85	2.98	0.710	0.9396	-1.5815
	21.533	6.0	8.11	2.11	6.04	-0.623		
	14.186	10.0	12.31	2.31	9.98	0.165		
3000.	28.307	5.0	6.44	1.44	4.98	0.358	0.9513	-1.1405
	15.507	10.0	11.75	1.75	10.04	-0.359		
	10.750	15.0	16.95	1.95	14.98	0.121		

IW LBS.	DELTA T	TW HP	CALC AHP	CALC FHP	LLSQ TW	%DIFF TW	LLSQ TW =M*AHP+B M	B
3125.	28.648	5.0	6.62	1.62	4.99	0.101	0.9555	-1.3352
	15.984	10.0	11.87	1.87	10.01	-0.102		
	11.105	15.0	17.09	2.09	14.99	0.034		
3250.	29.728	5.0	6.64	1.64	4.99	0.173	0.9593	-1.3777
	16.616	10.0	11.88	1.88	10.02	-0.173		
	11.567	15.0	17.06	2.06	14.99	0.058		
3375.	30.063	5.0	6.82	1.82	4.99	0.219	0.9647	-1.5881
	17.031	10.0	12.03	2.03	10.02	-0.219		
	11.928	15.0	17.18	2.18	14.99	0.074		
3500.	31.628	5.0	6.72	1.72	4.99	0.261	0.9620	-1.4781
	17.774	10.0	11.96	1.96	10.03	-0.261		
	12.419	15.0	17.12	2.12	14.99	0.088		
3625.	31.921	5.0	6.90	1.90	4.98	0.404	0.9661	-1.6826
	18.141	10.0	12.14	2.14	10.04	-0.406		
	12.764	15.0	17.25	2.25	14.98	0.138		
3750.	32.974	5.0	6.91	1.91	4.99	0.138	0.9643	-1.6671
	18.801	10.0	12.11	2.11	10.01	-0.139		
	13.182	15.0	17.28	2.28	14.99	0.047		
3875.	33.560	5.0	7.01	2.01	4.98	0.491	0.9598	-1.7545
	19.134	10.0	12.30	2.30	10.05	-0.495		
	13.501	15.0	17.43	2.43	14.97	0.168		
4000.	35.503	5.0	6.84	1.84	4.99	0.191	0.9684	-1.6354
	20.184	10.0	12.04	2.04	10.02	-0.192		
	14.149	15.0	17.17	2.17	14.99	0.064		
4250.	36.888	5.0	7.00	2.00	4.99	0.142	0.9702	-1.7957
	21.204	10.0	12.17	2.17	10.01	-0.142		
	14.916	15.0	17.30	2.30	14.99	0.048		
4500.	38.464	5.0	7.10	2.10	4.99	0.182	0.9729	-1.9215
	22.268	10.0	12.27	2.27	10.02	-0.184		
	15.721	15.0	17.38	2.38	14.99	0.062		
4750.	40.061	5.0	7.20	2.20	4.97	0.511	0.9688	-2.0013
	23.186	10.0	12.44	2.44	10.05	-0.514		
	16.463	15.0	17.52	2.52	14.97	0.175		
5000.	24.879	10.0	12.21	2.21	10.01	-0.055	0.9856	-2.0244
	16.615	16.0	18.28	2.28	15.99	0.069		
	12.455	22.0	24.38	2.38	22.01	-0.025		
5250.	25.824	10.0	12.35	2.35	9.99	0.083	0.9841	-2.1579
	17.263	16.0	18.47	2.47	16.02	-0.105		
	12.992	22.0	24.54	2.54	21.99	0.038		
5500.	26.697	10.0	12.51	2.51	10.00	-0.032	0.9854	-2.3258
	17.967	16.0	18.59	2.59	15.99	0.039		
	13.529	22.0	24.69	2.69	22.00	-0.014		

4

IW LBS.	DELTA T	TW HP	CALC AHP	CALC FHP	LLSQ TW	%DIFF TW	LLSQ TW =M*AHP+B M	B
6000.	28.771	10.0	12.66	2.66	9.99	0.084	0.9896	-2.5412
	19.430	16.0	18.75	2.75	16.02	-0.105		
	14.698	22.0	24.79	2.79	21.99	0.039		
6500.	30.679	10.0	12.87	2.87	10.00	-0.021	0.9882	-2.7124
	20.850	16.0	18.93	2.93	16.00	0.026		
	15.783	22.0	25.01	3.01	22.00	-0.009		

****
****   SLOPE/SPEED STATISTICS   ****
****

	N	MIN	MAX	AVG	SIGMA	%CV
IMP SLOPES	33	0.8333	0.9896	0.9387	0.0404	4.3047
AVG SPEEDS	99	49.60	49.67	49.63	0.0109	0.0220

5

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DDDDD     00000    00000    33333
D    D   0     0  0     0  3     3
D     D  0     0  0     0        3
D     D  0     0  0     0     333
D     D  0     0  0     0        3
D    D   0     0  0     0  3     3
DDDDD     00000    00000    33333
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DYNAMOMETER CALIBRATION TABLE FOR D003
DERIVED FROM AMP VERSUS TW DATA
CALIBRATION DATE: 07-12-79

APPROVED BY:________________________

EFFECTIVE DATE:____/____/____

PROCESSED: AUG 9, 1979 08:44:35
VERSION: 1.23
NOTE:FOR AMP VALUES OTHER THAN THOSE GIVEN IN THE TABLE, USE THE FOLLOWING FORMULA, ROUNDING THE RESULT TO THE NEAREST TENTH:

TW= M * AMP + B

INERTIA	1000		1125		1250		1375		1500		1625		1750		1875	
** M **	0.833		0.859		0.871		0.885		0.885		0.895		0.908		0.911	
** B **	-0.54		-0.73		-0.72		-0.90		-0.80		-0.96		-1.02		-1.16	
	AMP	TW	AMP	TW	AMP	TW	AMP	TW	AMP	TW	AMP	TW	AMP	TW	AMP	TW
	3.0	2.0	3.0	1.8	3.0	1.9	3.0	1.8	3.0	1.9	3.0	1.7	3.0	1.7	3.0	1.6
	3.1	2.0	3.1	1.9	3.1	2.0	3.1	1.8	3.1	1.9	3.1	1.8	3.1	1.8	3.1	1.7
	3.2	2.1	3.2	2.0	3.2	2.1	3.2	1.9	3.2	2.0	3.2	1.9	3.2	1.9	3.2	1.8
	3.3	2.2	3.3	2.1	3.3	2.2	3.3	2.0	3.3	2.1	3.3	2.0	3.3	2.0	3.3	1.8
	3.4	2.3	3.4	2.2	3.4	2.2	3.4	2.1	3.4	2.2	3.4	2.1	3.4	2.1	3.4	1.9
	3.5	2.4	3.5	2.3	3.5	2.3	3.5	2.2	3.5	2.3	3.5	2.2	3.5	2.2	3.5	2.0
	3.6	2.5	3.6	2.4	3.6	2.4	3.6	2.3	3.6	2.4	3.6	2.3	3.6	2.3	3.6	2.1
	3.7	2.5	3.7	2.4	3.7	2.5	3.7	2.4	3.7	2.5	3.7	2.4	3.7	2.3	3.7	2.2
	3.8	2.6	3.8	2.5	3.8	2.6	3.8	2.5	3.8	2.6	3.8	2.4	3.8	2.4	3.8	2.3
	3.9	2.7	3.9	2.6	3.9	2.7	3.9	2.6	3.9	2.7	3.9	2.5	3.9	2.5	3.9	2.4
	4.0	2.8	4.0	2.7	4.0	2.8	4.0	2.6	4.0	2.7	4.0	2.6	4.0	2.6	4.0	2.5
	4.1	2.9	4.1	2.8	4.1	2.9	4.1	2.7	4.1	2.8	4.1	2.7	4.1	2.7	4.1	2.6
	4.2	3.0	4.2	2.9	4.2	2.9	4.2	2.8	4.2	2.9	4.2	2.8	4.2	2.8	4.2	2.7
	4.3	3.0	4.3	3.0	4.3	3.0	4.3	2.9	4.3	3.0	4.3	2.9	4.3	2.9	4.3	2.8
	4.4	3.1	4.4	3.0	4.4	3.1	4.4	3.0	4.4	3.1	4.4	3.0	4.4	3.0	4.4	2.8
	4.5	3.2	4.5	3.1	4.5	3.2	4.5	3.1	4.5	3.2	4.5	3.1	4.5	3.1	4.5	2.9
	4.6	3.3	4.6	3.2	4.6	3.3	4.6	3.2	4.6	3.3	4.6	3.2	4.6	3.2	4.6	3.0
	4.7	3.4	4.7	3.3	4.7	3.4	4.7	3.3	4.7	3.4	4.7	3.2	4.7	3.3	4.7	3.1
	4.8	3.5	4.8	3.4	4.8	3.5	4.8	3.3	4.8	3.4	4.8	3.3	4.8	3.3	4.8	3.2
	4.9	3.5	4.9	3.5	4.9	3.5	4.9	3.4	4.9	3.5	4.9	3.4	4.9	3.4	4.9	3.3
	5.0	3.6	5.0	3.6	5.0	3.6	5.0	3.5	5.0	3.6	5.0	3.5	5.0	3.5	5.0	3.4
	5.1	3.7	5.1	3.6	5.1	3.7	5.1	3.6	5.1	3.7	5.1	3.6	5.1	3.6	5.1	3.5
	5.2	3.8	5.2	3.7	5.2	3.8	5.2	3.7	5.2	3.8	5.2	3.7	5.2	3.7	5.2	3.6
	5.3	3.9	5.3	3.8	5.3	3.9	5.3	3.8	5.3	3.9	5.3	3.8	5.3	3.8	5.3	3.7
	5.4	4.0	5.4	3.9	5.4	4.0	5.4	3.9	5.4	4.0	5.4	3.9	5.4	3.9	5.4	3.8
	5.5	4.0	5.5	4.0	5.5	4.1	5.5	4.0	5.5	4.1	5.5	4.0	5.5	4.0	5.5	3.9
	5.6	4.1	5.6	4.1	5.6	4.2	5.6	4.1	5.6	4.2	5.6	4.1	5.6	4.1	5.6	3.9
	5.7	4.2	5.7	4.2	5.7	4.2	5.7	4.1	5.7	4.2	5.7	4.1	5.7	4.2	5.7	4.0
	5.8	4.3	5.8	4.2	5.8	4.3	5.8	4.2	5.8	4.3	5.8	4.2	5.8	4.2	5.8	4.1
	5.9	4.4	5.9	4.3	5.9	4.4	5.9	4.3	5.9	4.4	5.9	4.3	5.9	4.3	5.9	4.2
	6.0	4.5	6.0	4.4	6.0	4.5	6.0	4.4	6.0	4.5	6.0	4.4	6.0	4.4	6.0	4.3
	6.1	4.5	6.1	4.5	6.1	4.6	6.1	4.5	6.1	4.6	6.1	4.5	6.1	4.5	6.1	4.4
	6.2	4.6	6.2	4.6	6.2	4.7	6.2	4.6	6.2	4.7	6.2	4.6	6.2	4.6	6.2	4.5
	6.3	4.7	6.3	4.7	6.3	4.8	6.3	4.7	6.3	4.8	6.3	4.7	6.3	4.7	6.3	4.6
	6.4	4.8	6.4	4.8	6.4	4.9	6.4	4.8	6.4	4.9	6.4	4.8	6.4	4.8	6.4	4.7
	6.5	4.9	6.5	4.8	6.5	4.9	6.5	4.9	6.5	5.0	6.5	4.9	6.5	4.9	6.5	4.8
	6.6	5.0	6.6	4.9	6.6	5.0	6.6	4.9	6.6	5.0	6.6	4.9	6.6	5.0	6.6	4.9
	6.7	5.0	6.7	5.0	6.7	5.1	6.7	5.0	6.7	5.1	6.7	5.0	6.7	5.1	6.7	4.9
	6.8	5.1	6.8	5.1	6.8	5.2	6.8	5.1	6.8	5.2	6.8	5.1	6.8	5.2	6.8	5.0
	6.9	5.2	6.9	5.2	6.9	5.3	6.9	5.2	6.9	5.3	6.9	5.2	6.9	5.2	6.9	5.1
	7.0	5.3	7.0	5.3	7.0	5.4	7.0	5.3	7.0	5.4	7.0	5.3	7.0	5.3	7.0	5.2
	7.1	5.4	7.1	5.4	7.1	5.5	7.1	5.4	7.1	5.5	7.1	5.4	7.1	5.4	7.1	5.3
	7.2	5.5	7.2	5.4	7.2	5.6	7.2	5.5	7.2	5.6	7.2	5.5	7.2	5.5	7.2	5.4
	7.3	5.5	7.3	5.5	7.3	5.6	7.3	5.6	7.3	5.7	7.3	5.6	7.3	5.6	7.3	5.5
	7.4	5.6	7.4	5.6	7.4	5.7	7.4	5.6	7.4	5.8	7.4	5.7	7.4	5.7	7.4	5.6
	7.5	5.7	7.5	5.7	7.5	5.8	7.5	5.7	7.5	5.8	7.5	5.8	7.5	5.8	7.5	5.7
	7.6	5.8	7.6	5.8	7.6	5.9	7.6	5.8	7.6	5.9	7.6	5.8	7.6	5.9	7.6	5.8
	7.7	5.9	7.7	5.9	7.7	6.0	7.7	5.9	7.7	6.0	7.7	5.9	7.7	6.0	7.7	5.9
	7.8	6.0	7.8	6.0	7.8	6.1	7.8	6.0	7.8	6.1	7.8	6.0	7.8	6.1	7.8	5.9
	7.9	6.0	7.9	6.1	7.9	6.2	7.9	6.1	7.9	6.2	7.9	6.1	7.9	6.2	7.9	6.0

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DDDDD    00000    00000    33333
D    D   0    0   0    0   3    3
D     D  0    0   0    0        3
D     D  0    0   0    0      333
D     D  0    0   0    0        3
D    D   0    0   0    0   3    3
DDDDD    00000    00000    33333
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DYNAMOMETER CALIBRATION TABLE FOR D003
DERIVED FROM AHP VERSUS TW DATA
CALIBRATION DATE: 07-12-79

APPROVED BY:________________________

EFFECTIVE DATE:____/____/____

VERSION: 1.23
NOTE:FOR AHP VALUES OTHER THAN THOSE GIVEN IN THE TABLE, USE THE FOLLOWING FORMULA, ROUNDING THE RESULT TO THE NEAREST TENTH:

TW= M * AHP + B

INERTIA	2000		2125		2250		2375		2500		2625		2750		2875	
** M **	0.913		0.921		0.921		0.932		0.930		0.939		0.940		0.940	
** B **	-0.99		-1.15		-1.14		-1.32		-1.28		-1.43		-1.44		-1.58	
	AHP	TW	AHP	TW	AHP	TW	AHP	TW	AHP	TW	AHP	TW	AHP	TW	AHP	TW
	6.0	4.5	6.0	4.4	6.0	4.4	6.0	4.3	6.0	4.3	6.0	4.2	6.0	4.2	6.0	4.1
	6.1	4.6	6.1	4.5	6.1	4.5	6.1	4.4	6.1	4.4	6.1	4.3	6.1	4.3	6.1	4.2
	6.2	4.7	6.2	4.6	6.2	4.6	6.2	4.5	6.2	4.5	6.2	4.4	6.2	4.4	6.2	4.2
	6.3	4.8	6.3	4.7	6.3	4.7	6.3	4.5	6.3	4.6	6.3	4.5	6.3	4.5	6.3	4.3
	6.4	4.9	6.4	4.7	6.4	4.8	6.4	4.6	6.4	4.7	6.4	4.6	6.4	4.6	6.4	4.4
	6.5	4.9	6.5	4.8	6.5	4.8	6.5	4.7	6.5	4.8	6.5	4.7	6.5	4.7	6.5	4.5
	6.6	5.0	6.6	4.9	6.6	4.9	6.6	4.8	6.6	4.9	6.6	4.8	6.6	4.8	6.6	4.6
	6.7	5.1	6.7	5.0	6.7	5.0	6.7	4.9	6.7	5.0	6.7	4.9	6.7	4.9	6.7	4.7
	6.8	5.2	6.8	5.1	6.8	5.1	6.8	5.0	6.8	5.1	6.8	4.9	6.8	5.0	6.8	4.8
	6.9	5.3	6.9	5.2	6.9	5.2	6.9	5.1	6.9	5.1	6.9	5.0	6.9	5.0	6.9	4.9
	7.0	5.4	7.0	5.3	7.0	5.3	7.0	5.2	7.0	5.2	7.0	5.1	7.0	5.1	7.0	5.0
	7.1	5.5	7.1	5.4	7.1	5.4	7.1	5.3	7.1	5.3	7.1	5.2	7.1	5.2	7.1	5.1
	7.2	5.6	7.2	5.5	7.2	5.5	7.2	5.4	7.2	5.4	7.2	5.3	7.2	5.3	7.2	5.2
	7.3	5.7	7.3	5.6	7.3	5.6	7.3	5.5	7.3	5.5	7.3	5.4	7.3	5.4	7.3	5.3
	7.4	5.8	7.4	5.7	7.4	5.7	7.4	5.6	7.4	5.6	7.4	5.5	7.4	5.5	7.4	5.4
	7.5	5.9	7.5	5.8	7.5	5.8	7.5	5.7	7.5	5.7	7.5	5.6	7.5	5.6	7.5	5.5
	7.6	6.0	7.6	5.8	7.6	5.9	7.6	5.8	7.6	5.8	7.6	5.7	7.6	5.7	7.6	5.6
	7.7	6.0	7.7	5.9	7.7	5.9	7.7	5.9	7.7	5.9	7.7	5.8	7.7	5.8	7.7	5.7
	7.8	6.1	7.8	6.0	7.8	6.0	7.8	5.9	7.8	6.0	7.8	5.9	7.8	5.9	7.8	5.7
	7.9	6.2	7.9	6.1	7.9	6.1	7.9	6.0	7.9	6.1	7.9	6.0	7.9	6.0	7.9	5.8
	8.0	6.3	8.0	6.2	8.0	6.2	8.0	6.1	8.0	6.2	8.0	6.1	8.0	6.1	8.0	5.9
	8.1	6.4	8.1	6.3	8.1	6.3	8.1	6.2	8.1	6.3	8.1	6.2	8.1	6.2	8.1	6.0
	8.2	6.5	8.2	6.4	8.2	6.4	8.2	6.3	8.2	6.4	8.2	6.3	8.2	6.3	8.2	6.1
	8.3	6.6	8.3	6.5	8.3	6.5	8.3	6.4	8.3	6.4	8.3	6.4	8.3	6.4	8.3	6.2
	8.4	6.7	8.4	6.6	8.4	6.6	8.4	6.5	8.4	6.5	8.4	6.4	8.4	6.5	8.4	6.3
	8.5	6.8	8.5	6.7	8.5	6.7	8.5	6.6	8.5	6.6	8.5	6.5	8.5	6.6	8.5	6.4
	8.6	6.9	8.6	6.8	8.6	6.8	8.6	6.7	8.6	6.7	8.6	6.6	8.6	6.6	8.6	6.5
	8.7	7.0	8.7	6.9	8.7	6.9	8.7	6.8	8.7	6.8	8.7	6.7	8.7	6.7	8.7	6.6
	8.8	7.0	8.8	7.0	8.8	7.0	8.8	6.9	8.8	6.9	8.8	6.8	8.8	6.8	8.8	6.7
	8.9	7.1	8.9	7.0	8.9	7.1	8.9	7.0	8.9	7.0	8.9	6.9	8.9	6.9	8.9	6.8
	9.0	7.2	9.0	7.1	9.0	7.1	9.0	7.1	9.0	7.1	9.0	7.0	9.0	7.0	9.0	6.9
	9.1	7.3	9.1	7.2	9.1	7.2	9.1	7.2	9.1	7.2	9.1	7.1	9.1	7.1	9.1	7.0
	9.2	7.4	9.2	7.3	9.2	7.3	9.2	7.3	9.2	7.3	9.2	7.2	9.2	7.2	9.2	7.1
	9.3	7.5	9.3	7.4	9.3	7.4	9.3	7.3	9.3	7.4	9.3	7.3	9.3	7.3	9.3	7.2
	9.4	7.6	9.4	7.5	9.4	7.5	9.4	7.4	9.4	7.5	9.4	7.4	9.4	7.4	9.4	7.3
	9.5	7.7	9.5	7.6	9.5	7.6	9.5	7.5	9.5	7.6	9.5	7.5	9.5	7.5	9.5	7.3
	9.6	7.8	9.6	7.7	9.6	7.7	9.6	7.6	9.6	7.7	9.6	7.6	9.6	7.6	9.6	7.4
	9.7	7.9	9.7	7.8	9.7	7.8	9.7	7.7	9.7	7.7	9.7	7.7	9.7	7.7	9.7	7.5
	9.8	8.0	9.8	7.9	9.8	7.9	9.8	7.8	9.8	7.8	9.8	7.8	9.8	7.8	9.8	7.6
	9.9	8.1	9.9	8.0	9.9	8.0	9.9	7.9	9.9	7.9	9.9	7.9	9.9	7.9	9.9	7.7
	10.0	8.1	10.0	8.1	10.0	8.1	10.0	8.0	10.0	8.0	10.0	8.0	10.0	8.0	10.0	7.8
	10.1	8.2	10.1	8.2	10.1	8.2	10.1	8.1	10.1	8.1	10.1	8.0	10.1	8.1	10.1	7.9
	10.2	8.3	10.2	8.2	10.2	8.2	10.2	8.2	10.2	8.2	10.2	8.1	10.2	8.1	10.2	8.0
	10.3	8.4	10.3	8.3	10.3	8.3	10.3	8.3	10.3	8.3	10.3	8.2	10.3	8.2	10.3	8.1
	10.4	8.5	10.4	8.4	10.4	8.4	10.4	8.4	10.4	8.4	10.4	8.3	10.4	8.3	10.4	8.2
	10.5	8.6	10.5	8.5	10.5	8.5	10.5	8.5	10.5	8.5	10.5	8.4	10.5	8.4	10.5	8.3
	10.6	8.7	10.6	8.6	10.6	8.6	10.6	8.6	10.6	8.6	10.6	8.5	10.6	8.5	10.6	8.4
	10.7	8.8	10.7	8.7	10.7	8.7	10.7	8.6	10.7	8.7	10.7	8.6	10.7	8.6	10.7	8.5
	10.8	8.9	10.8	8.8	10.8	8.8	10.8	8.7	10.8	8.8	10.8	8.7	10.8	8.7	10.8	8.6
	10.9	9.0	10.9	8.9	10.9	8.9	10.9	8.8	10.9	8.9	10.9	8.8	10.9	8.8	10.9	8.7

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DDDDD    00000    00000    33333
D   D   0     0  0     0  3     3
D    D  0     0  0     0        3
D    D  0     0  0     0      333
D    D  0     0  0     0        3
D   D   0     0  0     0  3     3
DDDDD    00000    00000    33333
```

DYNAMOMETER CALIBRATION TABLE FOR 0003
DERIVED FROM AHP VERSUS TW DATA
CALIBRATION DATE: 07-12-79

APPROVED BY:________________________

EFFECTIVE DATE:____/____/____

VERSION: 1.23
NOTE: FOR AHP VALUES OTHER THAN THOSE GIVEN IN THE TABLE, USE THE FOLLOWING FORMULA, ROUNDING THE RESULT TO THE NEAREST TENTH:

TW= M * AHP + B

INERTIA	3000		3125		3250		3375		3500		3625		3750		3875	
** M **	0.951		0.956		0.959		0.965		0.962		0.966		0.964		0.960	
** B **	-1.14		-1.34		-1.38		-1.59		-1.48		-1.68		-1.67		-1.75	
	AHP	TW	AHP	TW	AHP	TW	AHP	TW	AHP	TW	AHP	TW	AHP	TW	AHP	TW
	8.0	6.5	8.0	6.3	8.0	6.3	8.0	6.1	11.0	9.1	11.0	8.9	11.0	8.9	11.0	8.8
	8.1	6.6	8.1	6.4	8.1	6.4	8.1	6.2	11.1	9.2	11.1	9.0	11.1	9.0	11.1	8.9
	8.2	6.7	8.2	6.5	8.2	6.5	8.2	6.3	11.2	9.3	11.2	9.1	11.2	9.1	11.2	9.0
	8.3	6.8	8.3	6.6	8.3	6.6	8.3	6.4	11.3	9.4	11.3	9.2	11.3	9.2	11.3	9.1
	8.4	6.9	8.4	6.7	8.4	6.7	8.4	6.5	11.4	9.5	11.4	9.3	11.4	9.3	11.4	9.2
	8.5	6.9	8.5	6.8	8.5	6.8	8.5	6.6	11.5	9.6	11.5	9.4	11.5	9.4	11.5	9.3
	8.6	7.0	8.6	6.9	8.6	6.9	8.6	6.7	11.6	9.7	11.6	9.5	11.6	9.5	11.6	9.4
	8.7	7.1	8.7	7.0	8.7	7.0	8.7	6.8	11.7	9.8	11.7	9.6	11.7	9.6	11.7	9.5
	8.8	7.2	8.8	7.1	8.8	7.1	8.8	6.9	11.8	9.9	11.8	9.7	11.8	9.7	11.8	9.6
	8.9	7.3	8.9	7.2	8.9	7.2	8.9	7.0	11.9	10.0	11.9	9.8	11.9	9.8	11.9	9.7
	9.0	7.4	9.0	7.3	9.0	7.3	9.0	7.1	12.0	10.1	12.0	9.9	12.0	9.9	12.0	9.8
	9.1	7.5	9.1	7.4	9.1	7.4	9.1	7.2	12.1	10.2	12.1	10.0	12.1	10.0	12.1	9.9
	9.2	7.6	9.2	7.5	9.2	7.4	9.2	7.3	12.2	10.3	12.2	10.1	12.2	10.1	12.2	10.0
	9.3	7.7	9.3	7.6	9.3	7.5	9.3	7.4	12.3	10.4	12.3	10.2	12.3	10.2	12.3	10.1
	9.4	7.8	9.4	7.6	9.4	7.6	9.4	7.5	12.4	10.5	12.4	10.3	12.4	10.3	12.4	10.1
	9.5	7.9	9.5	7.7	9.5	7.7	9.5	7.6	12.5	10.5	12.5	10.4	12.5	10.4	12.5	10.2
	9.6	8.0	9.6	7.8	9.6	7.8	9.6	7.7	12.6	10.6	12.6	10.5	12.6	10.5	12.6	10.3
	9.7	8.1	9.7	7.9	9.7	7.9	9.7	7.8	12.7	10.7	12.7	10.6	12.7	10.6	12.7	10.4
	9.8	8.2	9.8	8.0	9.8	8.0	9.8	7.9	12.8	10.8	12.8	10.7	12.8	10.7	12.8	10.5
	9.9	8.3	9.9	8.1	9.9	8.1	9.9	8.0	12.9	10.9	12.9	10.8	12.9	10.8	12.9	10.6
	10.0	8.4	10.0	8.2	10.0	8.2	10.0	8.1	13.0	11.0	13.0	10.9	13.0	10.9	13.0	10.7
	10.1	8.5	10.1	8.3	10.1	8.3	10.1	8.2	13.1	11.1	13.1	11.0	13.1	11.0	13.1	10.8
	10.2	8.6	10.2	8.4	10.2	8.4	10.2	8.3	13.2	11.2	13.2	11.1	13.2	11.1	13.2	10.9
	10.3	8.7	10.3	8.5	10.3	8.5	10.3	8.3	13.3	11.3	13.3	11.2	13.3	11.2	13.3	11.0
	10.4	8.8	10.4	8.6	10.4	8.6	10.4	8.4	13.4	11.4	13.4	11.3	13.4	11.3	13.4	11.1
	10.5	8.8	10.5	8.7	10.5	8.7	10.5	8.5	13.5	11.5	13.5	11.4	13.5	11.4	13.5	11.2
	10.6	8.9	10.6	8.8	10.6	8.8	10.6	8.6	13.6	11.6	13.6	11.5	13.6	11.4	13.6	11.3
	10.7	9.0	10.7	8.9	10.7	8.9	10.7	8.7	13.7	11.7	13.7	11.6	13.7	11.5	13.7	11.4
	10.8	9.1	10.8	9.0	10.8	9.0	10.8	8.8	13.8	11.8	13.8	11.6	13.8	11.6	13.8	11.5
	10.9	9.2	10.9	9.1	10.9	9.1	10.9	8.9	13.9	11.9	13.9	11.7	13.9	11.7	13.9	11.6
	11.0	9.3	11.0	9.2	11.0	9.2	11.0	9.0	14.0	12.0	14.0	11.8	14.0	11.8	14.0	11.7
	11.1	9.4	11.1	9.3	11.1	9.3	11.1	9.1	14.1	12.1	14.1	11.9	14.1	11.9	14.1	11.8
	11.2	9.5	11.2	9.4	11.2	9.4	11.2	9.2	14.2	12.2	14.2	12.0	14.2	12.0	14.2	11.9
	11.3	9.6	11.3	9.5	11.3	9.5	11.3	9.3	14.3	12.3	14.3	12.1	14.3	12.1	14.3	12.0
	11.4	9.7	11.4	9.6	11.4	9.6	11.4	9.4	14.4	12.4	14.4	12.2	14.4	12.2	14.4	12.1
	11.5	9.8	11.5	9.7	11.5	9.7	11.5	9.5	14.5	12.5	14.5	12.3	14.5	12.3	14.5	12.2
	11.6	9.9	11.6	9.7	11.6	9.8	11.6	9.6	14.6	12.6	14.6	12.4	14.6	12.4	14.6	12.3
	11.7	10.0	11.7	9.8	11.7	9.8	11.7	9.7	14.7	12.7	14.7	12.5	14.7	12.5	14.7	12.4
	11.8	10.1	11.8	9.9	11.8	9.9	11.8	9.8	14.8	12.8	14.8	12.6	14.8	12.6	14.8	12.5
	11.9	10.2	11.9	10.0	11.9	10.0	11.9	9.9	14.9	12.9	14.9	12.7	14.9	12.7	14.9	12.5
	12.0	10.3	12.0	10.1	12.0	10.1	12.0	10.0	15.0	13.0	15.0	12.8	15.0	12.8	15.0	12.6
	12.1	10.4	12.1	10.2	12.1	10.2	12.1	10.1	15.1	13.0	15.1	12.9	15.1	12.9	15.1	12.7
	12.2	10.5	12.2	10.3	12.2	10.3	12.2	10.2	15.2	13.1	15.2	13.0	15.2	13.0	15.2	12.8
	12.3	10.6	12.3	10.4	12.3	10.4	12.3	10.3	15.3	13.2	15.3	13.1	15.3	13.1	15.3	12.9
	12.4	10.7	12.4	10.5	12.4	10.5	12.4	10.4	15.4	13.3	15.4	13.2	15.4	13.2	15.4	13.0
	12.5	10.8	12.5	10.6	12.5	10.6	12.5	10.5	15.5	13.4	15.5	13.3	15.5	13.3	15.5	13.1
	12.6	10.8	12.6	10.7	12.6	10.7	12.6	10.6	15.6	13.5	15.6	13.4	15.6	13.4	15.6	13.2
	12.7	10.9	12.7	10.8	12.7	10.8	12.7	10.7	15.7	13.6	15.7	13.5	15.7	13.5	15.7	13.3
	12.8	11.0	12.8	10.9	12.8	10.9	12.8	10.8	15.8	13.7	15.8	13.6	15.8	13.6	15.8	13.4
	12.9	11.1	12.9	11.0	12.9	11.0	12.9	10.9	15.9	13.8	15.9	13.7	15.9	13.7	15.9	13.5

8

```
DDDDD    00000    00000    33333
D   D  0     0  0     0  3     3
D    D 0     0  0     0        3
D    D 0     0  0     0      333
D    D 0     0  0     0        3
D   D  0     0  0     0  3     3
DDDDD    00000    00000    33333
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DYNAMOMETER CALIBRATION TABLE FOR D003
DERIVED FROM AHP VERSUS TW DATA
CALIBRATION DATE: 07-12-79

APPROVED BY:______________________

EFFECTIVE DATE:____/____/____

VERSION: 1.23
NOTE:FOR AHP VALUES OTHER THAN THOSE GIVEN IN THE TABLE, USE THE FOLLOWING FORMULA, ROUNDING THE RESULT TO THE NEAREST TENTH:

TW= M * AHP + B

INERTIA	4000		4250		4500		4750		5000		5250		5500		6000		6500	
** M **	0.968		0.970		0.973		0.969		0.986		0.984		0.985		0.990		0.988	
** B **	-1.64		-1.80		-1.92		-2.00		-2.02		-2.16		-2.33		-2.54		-2.71	
	AHP	TW	AHP	TW	AHP	TW	AHP	TW	AHP	TW	AHP	TW	AHP	TW	AHP	TW	AHP	TW
	10.0	8.0	10.0	7.9	10.0	7.8	10.0	7.7	13.0	10.8	14.0	11.6	15.0	12.5	15.0	12.3	16.0	13.1
	10.2	8.2	10.2	8.1	10.2	8.0	10.2	7.9	13.2	11.0	14.2	11.8	15.2	12.7	15.2	12.5	16.2	13.3
	10.4	8.4	10.4	8.3	10.4	8.2	10.4	8.1	13.4	11.2	14.4	12.0	15.4	12.8	15.4	12.7	16.4	13.5
	10.6	8.6	10.6	8.5	10.6	8.4	10.6	8.3	13.6	11.4	14.6	12.2	15.6	13.0	15.6	12.9	16.6	13.7
	10.8	8.8	10.8	8.7	10.8	8.6	10.8	8.5	13.8	11.6	14.8	12.4	15.8	13.2	15.8	13.1	16.8	13.9
	11.0	9.0	11.0	8.9	11.0	8.8	11.0	8.7	14.0	11.8	15.0	12.6	16.0	13.4	16.0	13.3	17.0	14.1
	11.2	9.2	11.2	9.1	11.2	9.0	11.2	8.8	14.2	12.0	15.2	12.8	16.2	13.6	16.2	13.5	17.2	14.3
	11.4	9.4	11.4	9.3	11.4	9.2	11.4	9.0	14.4	12.2	15.4	13.0	16.4	13.8	16.4	13.7	17.4	14.5
	11.6	9.6	11.6	9.5	11.6	9.4	11.6	9.2	14.6	12.4	15.6	13.2	16.6	14.0	16.6	13.9	17.6	14.7
	11.8	9.8	11.8	9.7	11.8	9.6	11.8	9.4	14.8	12.6	15.8	13.4	16.8	14.2	16.8	14.1	17.8	14.9
	12.0	10.0	12.0	9.8	12.0	9.8	12.0	9.6	15.0	12.8	16.0	13.6	17.0	14.4	17.0	14.3	18.0	15.1
	12.2	10.2	12.2	10.0	12.2	9.9	12.2	9.8	15.2	13.0	16.2	13.8	17.2	14.6	17.2	14.5	18.2	15.3
	12.4	10.4	12.4	10.2	12.4	10.1	12.4	10.0	15.4	13.2	16.4	14.0	17.4	14.8	17.4	14.7	18.4	15.5
	12.6	10.6	12.6	10.4	12.6	10.3	12.6	10.2	15.6	13.4	16.6	14.2	17.6	15.0	17.6	14.9	18.6	15.7
	12.8	10.8	12.8	10.6	12.8	10.5	12.8	10.4	15.8	13.5	16.8	14.4	17.8	15.2	17.8	15.1	18.8	15.9
	13.0	11.0	13.0	10.8	13.0	10.7	13.0	10.6	16.0	13.7	17.0	14.6	18.0	15.4	18.0	15.3	19.0	16.1
	13.2	11.1	13.2	11.0	13.2	10.9	13.2	10.8	16.2	13.9	17.2	14.8	18.2	15.6	18.2	15.5	19.2	16.3
	13.4	11.3	13.4	11.2	13.4	11.1	13.4	11.0	16.4	14.1	17.4	15.0	18.4	15.8	18.4	15.7	19.4	16.5
	13.6	11.5	13.6	11.4	13.6	11.3	13.6	11.2	16.6	14.3	17.6	15.2	18.6	16.0	18.6	15.9	19.6	16.7
	13.8	11.7	13.8	11.6	13.8	11.5	13.8	11.4	16.8	14.5	17.8	15.4	18.8	16.2	18.8	16.1	19.8	16.9
	14.0	11.9	14.0	11.8	14.0	11.7	14.0	11.6	17.0	14.7	18.0	15.6	19.0	16.4	19.0	16.3	20.0	17.1
	14.2	12.1	14.2	12.0	14.2	11.9	14.2	11.8	17.2	14.9	18.2	15.8	19.2	16.6	19.2	16.5	20.2	17.2
	14.4	12.3	14.4	12.2	14.4	12.1	14.4	11.9	17.4	15.1	18.4	15.9	19.4	16.8	19.4	16.7	20.4	17.4
	14.6	12.5	14.6	12.4	14.6	12.3	14.6	12.1	17.6	15.3	18.6	16.1	19.6	17.0	19.6	16.9	20.6	17.6
	14.8	12.7	14.8	12.6	14.8	12.5	14.8	12.3	17.8	15.5	18.8	16.3	19.8	17.2	19.8	17.1	20.8	17.8
	15.0	12.9	15.0	12.8	15.0	12.7	15.0	12.5	18.0	15.7	19.0	16.5	20.0	17.4	20.0	17.3	21.0	18.0
	15.2	13.1	15.2	13.0	15.2	12.9	15.2	12.7	18.2	15.9	19.2	16.7	20.2	17.6	20.2	17.4	21.2	18.2
	15.4	13.3	15.4	13.1	15.4	13.1	15.4	12.9	18.4	16.1	19.4	16.9	20.4	17.8	20.4	17.6	21.4	18.4
	15.6	13.5	15.6	13.3	15.6	13.3	15.6	13.1	18.6	16.3	19.6	17.1	20.6	18.0	20.6	17.8	21.6	18.6
	15.8	13.7	15.8	13.5	15.8	13.5	15.8	13.3	18.8	16.5	19.8	17.3	20.8	18.2	20.8	18.0	21.8	18.8
	16.0	13.9	16.0	13.7	16.0	13.6	16.0	13.5	19.0	16.7	20.0	17.5	21.0	18.4	21.0	18.2	22.0	19.0
	16.2	14.1	16.2	13.9	16.2	13.8	16.2	13.7	19.2	16.9	20.2	17.7	21.2	18.6	21.2	18.4	22.2	19.2
	16.4	14.2	16.4	14.1	16.4	14.0	16.4	13.9	19.4	17.1	20.4	17.9	21.4	18.8	21.4	18.6	22.4	19.4
	16.6	14.4	16.6	14.3	16.6	14.2	16.6	14.1	19.6	17.3	20.6	18.1	21.6	19.0	21.6	18.8	22.6	19.6
	16.8	14.6	16.8	14.5	16.8	14.4	16.8	14.3	19.8	17.5	20.8	18.3	21.8	19.2	21.8	19.0	22.8	19.8
	17.0	14.8	17.0	14.7	17.0	14.6	17.0	14.5	20.0	17.7	21.0	18.5	22.0	19.4	22.0	19.2	23.0	20.0
	17.2	15.0	17.2	14.9	17.2	14.8	17.2	14.7	20.2	17.9	21.2	18.7	22.2	19.6	22.2	19.4	23.2	20.2
	17.4	15.2	17.4	15.1	17.4	15.0	17.4	14.9	20.4	18.1	21.4	18.9	22.4	19.7	22.4	19.6	23.4	20.4
	17.6	15.4	17.6	15.3	17.6	15.2	17.6	15.0	20.6	18.3	21.6	19.1	22.6	19.9	22.6	19.8	23.6	20.6
	17.8	15.6	17.8	15.5	17.8	15.4	17.8	15.2	20.8	18.5	21.8	19.3	22.8	20.1	22.8	20.0	23.8	20.8
	18.0	15.8	18.0	15.7	18.0	15.6	18.0	15.4	21.0	18.7	22.0	19.5	23.0	20.3	23.0	20.2	24.0	21.0
	18.2	16.0	18.2	15.9	18.2	15.8	18.2	15.6	21.2	18.9	22.2	19.7	23.2	20.5	23.2	20.4	24.2	21.2
	18.4	16.2	18.4	16.1	18.4	16.0	18.4	15.8	21.4	19.1	22.4	19.9	23.4	20.7	23.4	20.6	24.4	21.4
	18.6	16.4	18.6	16.3	18.6	16.2	18.6	16.0	21.6	19.3	22.6	20.1	23.6	20.9	23.6	20.8	24.6	21.6
	18.8	16.6	18.8	16.4	18.8	16.4	18.8	16.2	21.8	19.5	22.8	20.3	23.8	21.1	23.8	21.0	24.8	21.8
	19.0	16.8	19.0	16.6	19.0	16.6	19.0	16.4	22.0	19.7	23.0	20.5	24.0	21.3	24.0	21.2	25.0	22.0
	19.2	17.0	19.2	16.8	19.2	16.8	19.2	16.6	22.2	19.9	23.2	20.7	24.2	21.5	24.2	21.4	25.2	22.2
	19.4	17.2	19.4	17.0	19.4	17.0	19.4	16.8	22.4	20.1	23.4	20.9	24.4	21.7	24.4	21.6	25.4	22.4
	19.6	17.3	19.6	17.2	19.6	17.1	19.6	17.0	22.6	20.3	23.6	21.1	24.6	21.9	24.6	21.8	25.6	22.6
	19.8	17.5	19.8	17.4	19.8	17.3	19.8	17.2	22.8	20.4	23.8	21.3	24.8	22.1	24.8	22.0	25.8	22.8

ATTACHMENT VI

Calibration of Chassis Dynamometers for Emissions- and Fuel Economy Testing of Passenger Cars

Recommended by

CEC-CF 22 project group

FOREWORD

A general calibration method for chassis dynamometers is an important prerequisite for obtaining comparable emissions- and fuel economy mean test results with reasonable confidence.

Recent studies of several investigators had shown that the methods currently in use for calibrating the load behaviour of chassis dynamometers are unsatisfactory in some respects.

As an example, the power absorption unit is calibrated in only one point of velocity. In practice, it is quite possible for two dynamometers which absorb the same amount of power at the calibration point to differ markedly at any speed below this point, giving rise to non-comparable emissions- or fuel economy test results.

In order to provide recommendations for a more suitable chassis dynamometer calibration procedure a project group was established in January 1977. This group - called CF 22 - is integrated into the C.E.C. Engine Fuels Technical Committee.

As a result of our studies within the CF 22 group, we present in this report a proposal which is in our opinion suitable for a standardized chassis dynamometer calibration procedure.

November 1978 D. Schürmann

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Introduction

According to e.g. the European procedure the load of the dynamometer is adjusted at a speed of 50 km/h, in the US at 80 km/h. This does not furnish any data regarding the behaviour of the power absorber below 50 km/h or 80 km/h, respectively, giving rise to a grave problem, because the major proportion of the ECE test is run in the speed range below 50 km/h, that of the US test below 80 km/h (s. Fig. 1).

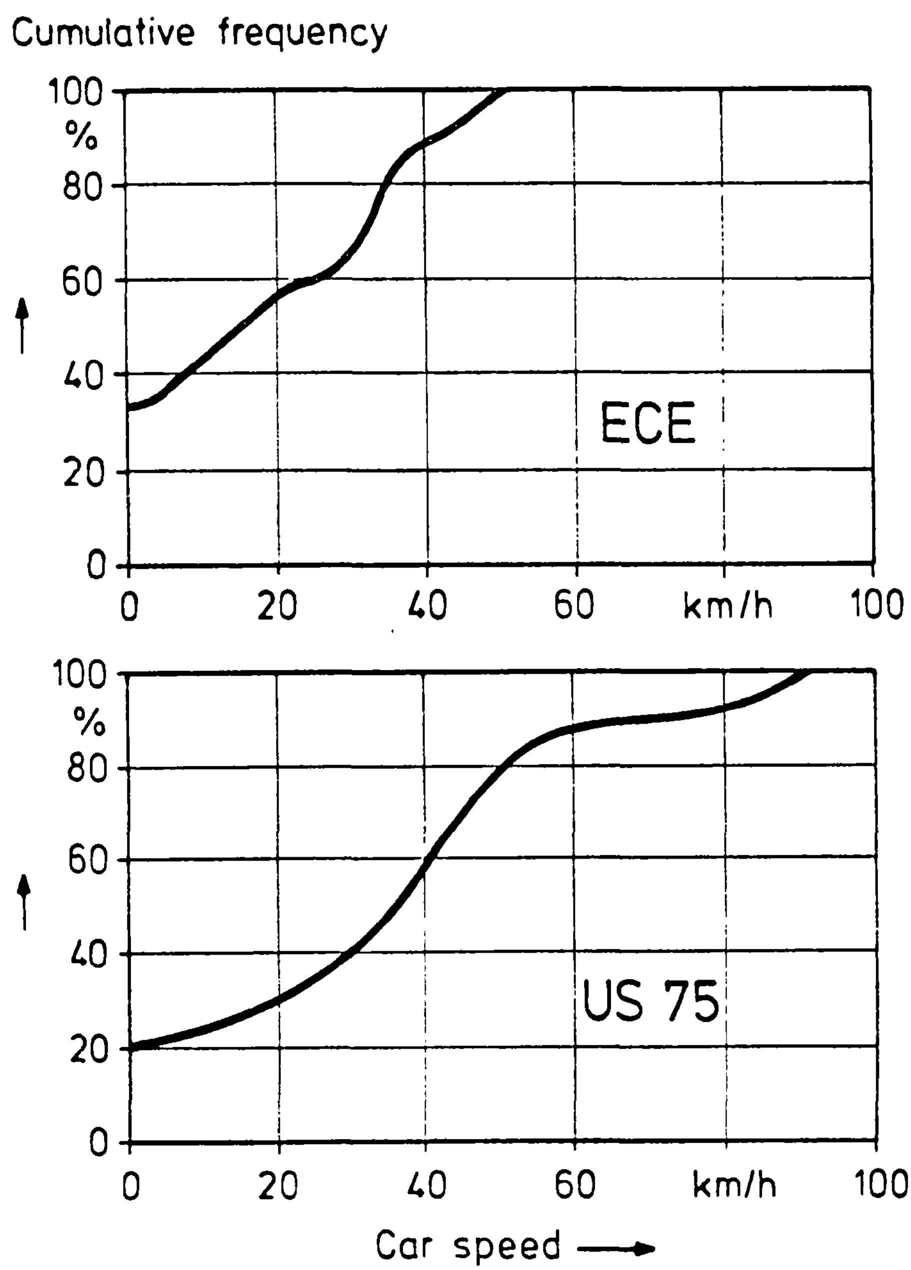


Fig. 1: Cumulative Frequency of Speeds in the European- and US Driving Cycles.
Nearly 90 % of all driven speeds are below 40 km/h (ECE) or below 60 km/h (US).

According to Fig. 1, 90 % of all driving modes involve speeds below 40 km/h in the case of the European test and below 60 km/h in the case of the US test.

Therefore, the way in which the power absorption unit behaves in these ranges is of decisive influence on the exhaust emissions- and fuel economy test results of the vehicle under test.

Special problems are raised if, according to the European regulation (and also to the US regulation in the past time) the load resistance of the dynamometer is adjusted by using the manifold vacuum pressure value measured when driving on the road at 50 km/h (ECE) or 80 km/h (US). With this method precision and reproducibility are insufficient since generally the intake manifold vacuum values are widely dispersed between 20 km/h and 70 km/h, being strongly dependent on such factors as engine type, emission control system, driver. A typical example for this situation is displayed in Fig. 2.

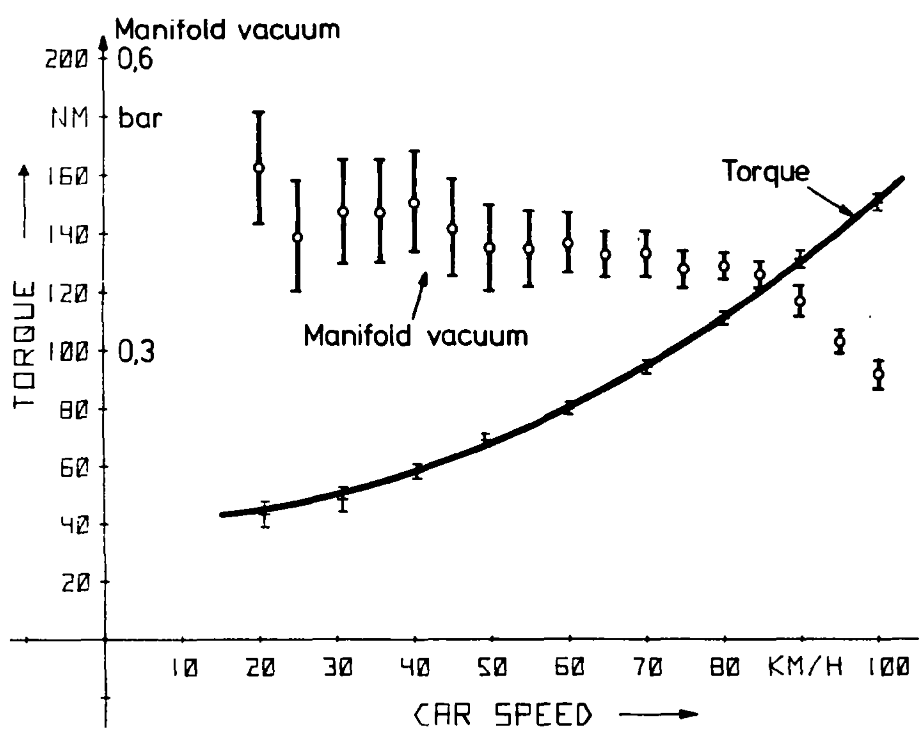


Fig. 2: Manifold Vacuum as a Function of Car Speed.
The spread of the data below 70 km/h is considerable.
For Comparision the torque on the drive shafts of the test vehicle is also plotted, showing much less dispersion.

As a consequence, the power absorber loading of the dynamometer cannot accurately be determined by using the vacuum pressure method.

In recognition of the fact that the legal procedures described above suffer from a variety of defects, several investigators began to develop an alternative method for calibrating chassis dynamometers aiming at an improved procedure. Some members of the CEC-CF 22 project group carried out an extensive measuring program in order to back up the procedure with more data on various types of vehicles.

1. Description of Method

1.1 General

Starting from the deliberation that a procedure should be developed which allows the measurement of the forces (driving resistances) experienced by the vehicle on the road as directly as possible the CEC-CF 22 group agreed to study torque methods. This decision was supported by promising test results already available at that time through some group members. To enable an accurate adjustment of a dynamometer to road load conditions it is necessary to measure the load behaviour of the vehicle on the road. This is done by recording the road torque measured at the drive shafts or in the wheels of the test car at constant speeds in the range from 20 km/h to 100 km/h in increments of 10 km/h (s. Fig. 3). If then - for calibration purposes - the car is placed on the dynamometer and the same torque curve over the above speed range is reproduced as determined on the road, it is ensured that the power absorption unit is adjusted to actual road load. This completes the constant speed (stationary) portion of the adjustment procedure.

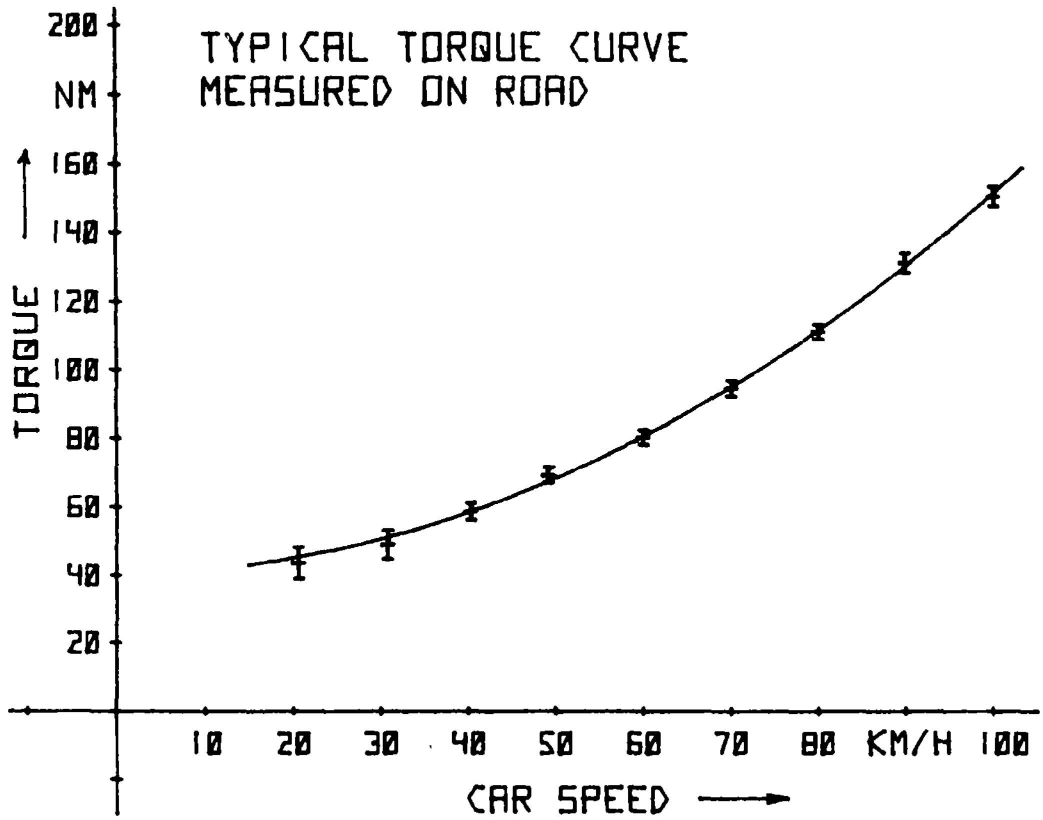


Fig. 3: Torque Measured as a Function of Stepwise Constant Speed of a Test Vehicle.
The data were reduced to standard ambient conditions and "zero grade" of the road. The error bars mean the standard deviations. The curve is a second order polynominal of the form $M = a_o + a_2 v^2$ and computed by means of the least squares method giving the numerical values of a_o and a_2.
M = torque, v = vehicle velocity.

$\frac{a_o}{r}$ can be interpreted as the rolling resistance under the assumption that the coefficient of rolling resistance is independent of velocity which is justified as an approximation in the velocity range studied here.

Then $\frac{a_o}{r} = f_R mg$, with r = rolling radius of tire, f_R = coefficient of rolling resistance, m = vehicle mass, g = acceleration of gravity.

Under the above assumption for f_R, $\frac{a_2}{r} v^2$ is the air resistance

$\frac{a_2}{r} v^2 = \frac{\rho}{2} c_a A v^2$ with ρ = air density at standard conditions, c_a = coefficient of air resistance, A = projected vehicle area.

In practice - due to the lack of flexibility in their characteristics - for water brake and older eddy current dynamometer types it is not possible to adjust the dynamometer load curves over the whole speed range (20 km/h to 100 km/h) to actual road load. In the CF 22 group it was therefore agreed to recommend - as the best practicable approximation to the complete road load curve - a compromise for those dynamometers consisting of adjusting the load curve in two points with defined tolerances. In this way, the current method of using only one adjustment point is substantially improved. A perfect reproduction of actual road load conditions can be achieved with DC-machine and modern eddy current type dynamometers. Here, the CF 22 group recommends the adjustment of the whole load curve. In addition, with these types of dynamometers all influences of the dynamometer geometry such as roll diameters, spacing of rolls and also the bearing friction can be compensated.

Going back to the basic philosophy that the dynamometer should reflect "what the vehicle sees on the road" (EPA) it is also necessary to look at the dynamic behaviour of the vehicle during the driving cycles. This dynamic behaviour can also be characterized very exactly by using the torque method. For this purpose the CF 22 group recommends the use of the integrated (mean) torque $\overline{M}$ determined by the integral

$$\overline{M} = \frac{1}{t_2-t_1} \int_{t_1}^{t_2} M(t)\, dt \qquad (\text{with } M(t) > 0)$$

measured over a variable driving pattern, preferably the first 505 s of the US driving cycle (LA-4) excluding the idle period at the beginning (t_1 = 20 s; t_2 = 505 s). The first phase of the European test can of course also be used.

In order to adjust the dynamic behaviour of the dynamometer to actual road conditions this mean torque value $\overline{M}$ first has to be determined on the road.

Leaving the already adjusted load curve unchanged this "dynamic" $\overline{M}$-value is reproduced on the dynamometer by varying its flywheel masses, the precision of adjustment being only dependent on the size of the flywheel increments available at the dynamometer.

1.2 Physical Background

In the following a short outline of the physical background of the torque calibration procedure is given.

We start from the basic equation of motion. To move a vehicle the tractive force F or torque M acting on the driving wheels is needed to overcome the various driving resistances.

$$F = \frac{M}{r} = R_r + R_g + R_a + R_i \qquad (1)$$

<u>Nomenclature</u>

F	=	Tractive force on driving wheels
M	=	Torque on driving wheels
R_r	=	Rolling resistance
R_g	=	Grade resistance
R_a	=	Air resistance
R_i	=	Inertia resistance
r	=	Distance of drive axle to road surface
R_r	=	$f_R mg$
f_R	=	Coefficient of rolling friction (in general: $f_R = f_R(v)$)
m	=	Vehicle mass, g = Acceleration of gravity
v	=	Vehicle velocity
R_g	=	$mg\sin\alpha$
$\sin\alpha \simeq$		$\tan\alpha$ = grade [%] (valid for small grades)
R_a	=	$\frac{\rho}{2} c_a A v^2$
ρ	=	Air density at standard ambient conditions
c_a	=	Coefficient of air resistance
A	=	Projected vehicle area
R_i	=	$m\dot{v} + \lambda m \dot{v}$
R_i	=	$m(1 + \lambda)\dot{v} = m^* \dot{v}$
$\dot{v}$	-	$\frac{dv}{dt}$
λ	-	vehicle parameter which accounts for rotating masses.

With these expressions and for the case of "zero grade", i.e. $R_g = 0$ we rewrite equation (1) as a function of time:

$$M(t)_{road} = r\,(f_R\, mg + \frac{\rho}{2}\, c_a\, A\, v^2\,(t) + m^*\, \dot{v}\,(t))$$

This equation governs the adjustment of the dynamometer load behaviour to that of actual road driving.

The equation of motion for the vehicle on the dynamometer reads:

$$M(t)_{dyno} = M(t)_{PAU} + \Theta\,\dot{\omega}(t)$$

with

$M(t)_{dyno}$ = Torque on the driving wheels of the vehicle on the dyno

$M(t)_{PAU}$ = Resistance Torque provided by the power absorption unit of the dyno.

and

$$\theta\,\dot{\omega}\,(t) = \theta_{roll}\,\dot{\omega}\,(t) + \theta_{sim}\;k\,\dot{\omega}(t)$$

$$= (\theta_{roll} + k\,\theta_{sim})\;\dot{\omega}\,(t)$$

with

$\dot{\omega}$ = angular acceleration

θ_{roll} = Inertia of the dynamometer roll

θ_{sim} = Inertia simulated either by discrete flywheels or electrically.

k = Factor accounting for a possible transmission between roll and flywheel axle.

In most cases $k = 1$ holds true. Therefore, in the following $k = 1$ is assumed.

$$\dot{\omega} = \frac{d\omega}{dt} = \frac{\dot{v}}{R}$$

with

R = Radius of dynamometer roll

v = Circumferential velocity at the surface of the roll and at the rolling radius of the vehicle tires (neglecting slip).

Our recommended procedure is performed in two steps:

(1) Constant speed (stationary) procedure: $\dot{v} = 0$, $\dot{\omega} = 0$, i.e. v = const., ω = constant, and M = constant (acceleration effects are not taken into account).

Then: $M_{road} = r(R_r + R_a)$ and $M_{dyno} = M_{PAU}$.

M is measured on the road for constant speeds in the range of 20 km/h to 100 km/h, in increments of Δv = 10 km/h, resulting in constant M-values: $M_{20}, M_{30}, \ldots M_{100}$.

The dynamometer torque M_{PAU} is then adjusted as good as possible to these torque values so that $M_{PAU} = M_{road}$, thus simulating rolling and air resistances.
The geometrical dimensions of the dynamometer such as the roll diameter etc. no longer have an influence. The same holds true for the friction losses of the dyno. Therefore, coast down runs for determining these losses are no longer necessary.

(2) Integrated torque (dynamic) procedure: $\dot{v}(t) \neq 0$, $\dot{\omega}(t) \neq 0$, i.e. acceleration is taken into account (driving cycle).
The torque difference ΔM between the vehicle on the road and on the dyno reads:

$$\Delta M(t) = M(t)_{road} - M(t)_{dyno}$$

$$\Delta M(t) = r(R_r+R_a) + rm^* \dot{v}(t) - M_{PAU} - (\theta_{roll}+\theta_{sim}) \frac{\dot{v}(t)}{R}$$

The term $r(R_r+R_a)$ is equal to M_{PAU} because this part of adjustment has already been accomplished in step (1).

Thus,

$$\Delta M(t) = rm^* \dot{v}(t) - (\theta_{roll}+\theta_{sim}) \frac{\dot{v}(t)}{R}$$

$$= (rm^* - \frac{\theta_{roll} + \theta_{sim}}{R}) \dot{v}(t)$$

The only remaining term still to be adjusted, i.e. simulated by the dyno is therefore rm^*.

From this follows the condition for perfect adjustment of the dynamometer load to road load.

i.e. $\Delta M \rightarrow 0$ or $M(t)_{dyno} \approx M(t)_{road}$:

$$\frac{\theta_{roll} + \theta_{sim}}{R} = rm^* \quad \text{(by adjusting } \theta_{sim}\text{)}.$$

This at the same time also simulates the effect of the non-driven vehicle wheels. In order to simplify the measuring and evaluation procedures we use the same relation but integrate both sides of the equation and multiply by $\frac{1}{t_2 - t_1}$ as is done in our recommended procedure:

$$\frac{1}{t_2 - t_1}\left[\int_{t_1}^{t_2} M(t)_{road} dt - \int_{t_1}^{t_2} M(t)_{dyno}\, dt\right] = \left(rm^* - \frac{\theta_{roll}+\theta_{sim}}{R}\right) \frac{1}{t_2 - t_1} \int_{t_1}^{t_2} \dot{v}(t)\, dt$$

or

$$\Delta \overline{M} = \overline{M}_{road} - \overline{M}_{dyno} = \left(rm^* - \frac{\theta_{roll} + \theta_{sim}}{R}\right) \bar{\dot{v}}$$

with the condition for adjustment that $\overline{M}_{road} \approx \overline{M}_{dyno}$.

To sum up the dynamic procedure we note that

(i) the parameter for dynamically adjusting the dynamometer load behaviour to road conditions is θ_{sim}, i.e. the inertia of the flywheel mass of the dynamometer system.
By varying for instance the flywheel mass of the dynamometer so that $\frac{\theta_{roll} + \theta_{sim}}{R} = rm^*$ the difference $\Delta \overline{M}$ between road and dynamometer vanishes and the dyno is completely matched.

(ii) the driving cycle and the length of the time period, i.e. the integration limits (t_1, t_2) can be chosen arbitrarily.
For reasons of a considerable measuring effect it is advisable, however, to take a driving cycle with rather high accelerations such as the US cycle or parts of it. Both steps - the stationary and dynamic one - for calibrating dynamometers must be seen as a whole, taken together they constitute the complete calibration

2. Types of Chassis Dynamometers

2.1 General

Motor vehicle emission testing is mainly performed on three types of dynamometers: Waterbrake dyno, eddy current brake dyno and the DC-machine dyno. Basically, these three types differ from one another in their transformation of the mechanical energy of the vehicle being tested into another form of energy (see table 1).

Dynamometer types	Conversion of mechanical energy into:	Simulation of driving resistance torque $M=r(R_r+R_a)$ by	simulation of inertia resistance (θ_{sim}) by
water brake	heat energy (by internal friction of the whirling water)	adjustable water charges of the brake $M_{PAU} = B + Av^{\beta}$ *) $1 \leq \beta < 2$ (B = mechanical friction losses)	discrete flywheels in steps of 125 lbs or 250 lbs
eddy current brake	heat energy (by induction of eddy currents in the stator of the brake)	$M_{PAU} = B' + A' v^{\gamma}$ $2 \leq \gamma < 3$ (B' = mechanical friction losses)	discrete flywheels in steps of 125 lbs or 250 lbs
direct current (DC-Machine)	electrical energy (generator principle)	adjustable DC-voltages so that $M_{PAU} = a_0 + a_1 v + a_2 v^2$ *)M_{PAU} = torque of power absorption unit, v = velocity	(i) discrete flywheels in steps of 125 lbs or 250 lbs (ii) electrical simulation

Table 1: Characteristic Properties of Chassis Dynamometers.
$M_{road}=a_0+a_2v^2$ (for $\dot{v}=0$).
($a_0=rf_Rmg$, $a_2=r\frac{\rho}{2}c_aA$, a_1 is for compensation of friction)
$\theta_{sim}=R\ rm^* - \theta_{roll}$ (s. section 1.2)
B, B', a_0 also depend on the temperature of the dyno bearings. Therefore, a warming up of the dyno - for all types of dynos - is necessary.

2.2 Characteristic Properties

In the following we discuss the essential properties of the different types of dynamometers.

(i) Water Brake Dynamometer:

The parameter A (s. table 1) can be varied by the amount of water within the brake. β-which should be equal to 2 in order to simulate the air resistance (s. section 1.2) - is given mainly by the geometry of the brake. However, β is not exactly a constant due to hydraulic losses. To some extent it is possible to correct the deviation of β from the required value of 2 by a control circuit.

(ii) Eddy Current Brake Dynamometer:

Here, the torque is approximately proportional to v . Again $M \sim v^2$ is required in order to simulate the air resistance. This is accomplished by controlling the current i through the exciter coil in such a way that $i \sim v$, resulting in $M \sim v^{\gamma}$ with $2 \leq \gamma < 3$, thus approximating air resistance. This approximation-which is quite common for most dynamometers currently in use - is not sufficient because of the fact that the parameter γ is not equal to 2. New designs of this dynamometer type use electronic control circuits for the exciter current and enable a more realistic reproduction of the road resistances.

(iii) Direct Current Brake Dynamometer:

With this type of dynamometer the rolling and air resistances a_0 and $a_2\ v^2$, respectively, can be reproduced nearly perfectly on the dynamometer. In addition, the friction losses ($a_1 v$) can be compensated in some cases. Positive and negative grades can be simulated, too. Moreover, the inertia resistance can also be simulated electrically allowing a continuous variation of inertia.

In general, the usefulness of a dynamometer must be measured by the degree of its capability to reproduce the road driving resistances of a given vehicle, i.e. what the vehicle sees on the road.

The different types of dynamometers (s. table 1) can be compared by using the general equation of motion which applies to the system "vehicle on the road" on the one side and "vehicle on the dyno" on the other.

With the nomenclature of the foregoing section we write the equations of motion:

$$M_{road} = r(R_r + R_a) + rm^* \dot{v} \quad (\text{assuming } R_g = 0)$$

$$M_{dyno} = M_{PAU} + \frac{\theta_{roll} + \theta_{sim}}{R} \dot{v}$$

Condition for adjustment: $M_{road} = M_{dyno}$

The more important difference between the types of dynamometer refers to M_{PAU}, i.e. the power absorption unit.

Most of the water brake dynos currently in use have a fixed shape of the characteristic curve expressed by torque as a function of car speed. By changing the absorbed power (changing A) in a given point of dyno speed the whole curve is thereby already established. Fig. 4 shows examples for different values of A (here: A = a_2, see also table 1).

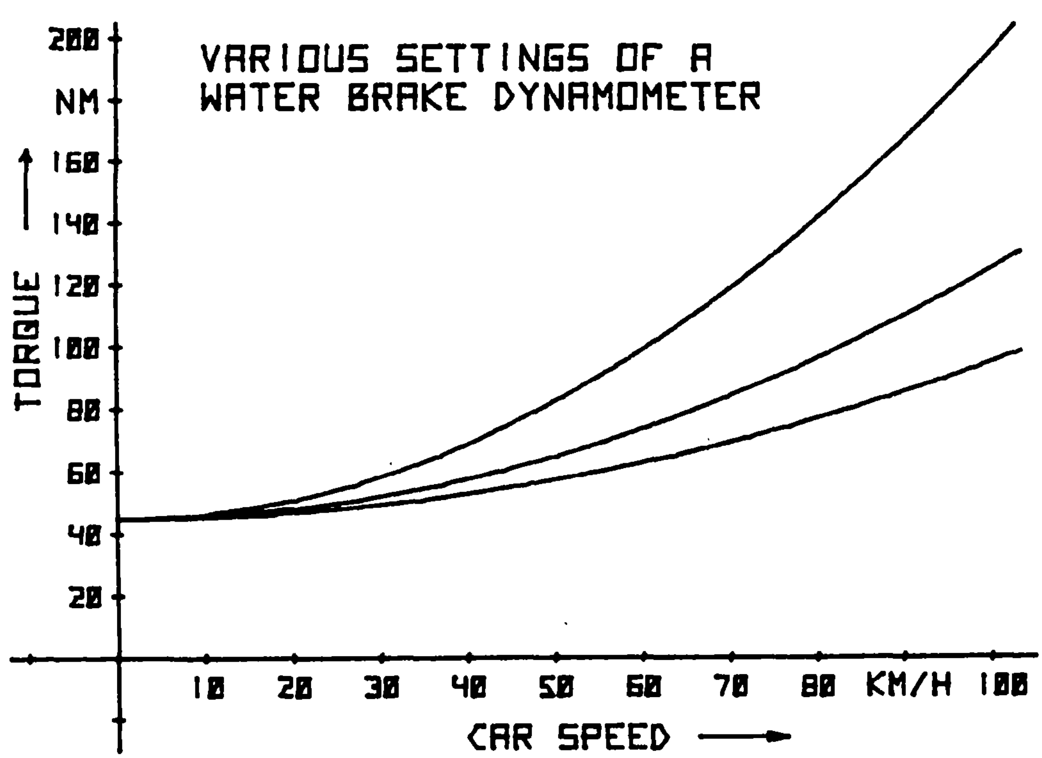


Fig. 4: Torque vs Car Speed of a Water Brake Dynamometer.
An (idealized) functional dependence of $M = a_o + a_2 v^2$ was assumed. Only the parameter a_2 can be varied. The three curves represent a_2-values of 0.015, 0.008 and 0.005 (from top to bottom); a_o = 45 was assumed for each curve. The disadvantage of the water brake principle is obvious, because it is not possible to vary the parameter a_o (rolling resistance + internal friction). Therefore, an acceptable adjustment to road load is not possible in many cases.

In contrast to this, the DC machine dyno and the modern eddy current dyno are much more flexible. Here, the road torque can be exactly reproduced on the dynos by adjusting the simulation for the coefficients of rolling resistance and air resistance resulting in $M_{PAU} = r\ (R_r + R_a)$.

This is especially true for the DC machine type dynamometer. Even when compared to the eddy current brake dynamometer the DC-dynamometer has some considerable advantages.

(i) Possibility of continuous inertia simulation.

(ii) Possibility of operating as a motor.
This motor operation is a rather important property for daily use, because e.g. the warming up of the dynamometer does not require an additional vehicle or other equipment for driving the dyno.

(iii) In general, the DC dyno has a higher dynamic response to varying loads.

(iv) The mechanical energy of the vehicle on the dynamometer is converted into useful electrical energy, which is returned to the main system during operation of the dyno as a power absorber (generator).

(v) Furthermore, in contrast to the two other types of dynos no cooling water is required.

3. Comparison of Torque Method to Other Methods.

The purpose of this section is to describe the various calibration procedures for chassis dynamometers currently in use and to point out their advantages and disadvantages compared to the torque method.

3.1 Review of Existing Methods

3.1.1 Inertia Weight Classes

First, we have the well known table values of inertia weight classes for the dynamometer load at one given speed point. These values are based on the corresponding loaded vehicle weights. Such tables are used in both Europe and the US. Necessarily, these table values are only very rough figures due to the fact that only one vehicle parameter, namely its weight is considered.

Such an important quantity like the particular shape of a vehicle type - i.e. its particular air resistance - is not taken into account. This can easily be seen from the basic equation of motion (with Rg = 0):

$$F = R_r + R_i + R_a = f_R mg + m^* \dot{v} + \frac{\varrho}{2} c_a A v^2$$

Only two of the three resistance forces - namely the rolling resistance R_r and the inertia resistance R_i - are proportional to the vehicle mass. This is not the case for the air resistance R_a. Here, the product of the air drag coefficient c_a and the frontal area A is the decisive quantity. Therefore, the classification according to the inertia weight classes alone is obviously not sufficient.

The calibration of the dynamometer load at the given one point of speed is carried out by a coast down procedure. From the coast down - which has to be performed for each individual inertia weight because of varying amounts of friction losses - the internal friction of the dynamometer can be calculated.

This internal friction is due to the friction in the bearings of the clutches, rollers, and flywheels and due to the aerodynamic drag associated with the spinning rotating parts. The total amount of power - which is required for the test vehicle at the roller surface to drive the dynamometer - is given by the sum of the so-called indicated load and the internal friction. The indicated load means the output of the power meter of the dynamometer. Here we have the problem that the inertia of the idle roll is not considered, because the vehicle is lifted from the rolls.

3.1.2 Intake Manifold Pressure

The still remaining problem which is basically inherent in all vehicle testing on dynamometers, is the question of correlating the vehicle operation on the dyno with that on the road. The table values are only a rough approximation for this.

As a first alternative to the table values, the manufacturer was allowed to take the intake manifold vacuum pressure value of a representative vehicle driven at constant speed on the road as a basis for adjusting the dynamometer at that same speed.

Meanwhile, for 1979 and later model year light-duty vehicles the intake manifold vacuum procedure was removed from the US regulations as a requirement. Therefore, this adjustment method is no longer automatically accepted by EPA but only if a manufacturer requests its use. In Europe, the manifold vacuum procedure is - as well - no longer considered as the only adjustment procedure and emphasis was changed to use it as an alternative method among others.

The problems arising with this procedure have been already discussed in section 1. Generally, this procedure is very unsatisfactory and is therefore no longer considered by the group members as a useful method.

3.1.3 Aerodynamical Assumptions

In the US, a further adjustment procedure for 1979 and later model year light-duty vehicles is based on calculations of the dynamometer power absorber setting from vehicle parameters such as frontal area, body shape and tire type. The assumption about the body shape of the vehicle-including protruding parts such as mirrors etc. - is only a more or less accurate approximation to reality. This is due to the fact that the coefficient of air drag c_a is not explicitely considered in the calculations. In this procedure, again the dynamometer power absorber is adjusted in only one point of speed.

3.1.4 Coast Down

A further attempt to solve the essential problem of reproducing the road behaviour of the vehicle on a dynamometer is the application of the well known coast down technique to measurements on the road. This procedure is valid in the US for 1979 model year light-duty vehicles as an alternative. The measuring quantity of this procedure is the time versus speed behaviour of the vehicle when freely decelerating between two given velocities. The main task which must be accomplished is to extract from these speed vs time data the deceleration vs velocity function. From this function the forces acting on the vehicle during deceleration can be calculated. First, we have to assume a model of the deceleration ($-\dot{v}$) vs speed (v) equation.

$$-\dot{v} = a_0 + g \sin\alpha + a_2 (v-w)^2$$

with a_0 = rolling resistance term

$g \sin\alpha$ = grade resistance term

$a_2 (v-w)^2$ = aerodynamic term, w = wind velocity.

Whenever possible the wind velocity should be constant and the direction should be the same as the vehicle translation. The most reliable results are obtained with $w=0$.

The above equation must be integrated to obtain a function for the speed vs time behaviour. The result of this integration is of the form:

$$v = \mathcal{f}(a_0, a_2) \cdot \tan\left[\chi(a_0, a_2)\right] \quad ; \quad \mathcal{f}, \chi = \text{functions of } a_0, a_2$$

The measured coast down data $v = v(t)$ have to be fitted to this non-linear equation by a least squares technique .

As a result the coefficients a_o and a_2 can be obtained. The total road force (F) acting on the vehicle can be written as

$-F = m^* \dot{v}$ or $-F = m^* (a_o + a_2 v^2) = f_o + f_2 v^2$ with m^* = total effective vehicle mass and assuming $\alpha = 0$, $w=0$.

The values f_o and f_2 must still be corrected to standard ambient conditions by the well known equations.

From the corrected f_o and f_2 values finally the time interval ΔT for the vehicle coasting down from 55 mph to 45 mph can be calculated. This ΔT-extracted from the road data as described-is then reproduced on the dynamometer. So, finally this ΔT is the decisive quantity for adjusting the dyno to road conditions.

3.2 Torque vs Coast Down Method

In the following only the coast down method is compared to the torque method. The other procedures discussed in the preceding sections are no more considered here, because in the opinion of the group it is evident that these procedures cannot compete with the torque- or the coast down method.

There are several reasons for preferring the torque method rather than the coast down method.

(i) As was seen in section 1 it is very important to adjust the whole load curve of the dynamometer or at least two representative points to road conditions. This is not accomplished - however possible - with the coast down procedure as recommended by EPA, where again only one point of velocity is adjusted to road conditions.

(ii) The internal friction losses of the dynamometer are also automatically taken into account by the torque method. Therefore, coast down checks for determining the friction losses are no longer necessary.

(iii) If the driving resistance on the road is measured in the vehicle drive wheels by the torque method, possible influences on the measured value - e.g. by sticking brakes, variation of the friction losses in the drive train (oil temperature variation) - are automatically eliminated. This is not the case with a coast down method.

(iv) A fundamental drawback of the coast down technique is the fact that it is a very indirect method, i.e. the measured speed vs time data cannot be taken rightaway to calibrate the dyno. In contrast, some non-elementary mathematical operations like fitting data to a non-linear function are necessary. In some cases this might eventually cause convergency problems in determining the required coefficients a_o and a_2, perhaps, especially if the areodynamic drag coefficient (i.e. a_2) is very small.

These problems are not present with the torque method because the needed load curve is measured directly. The rolling resistance and aerodynamic drag can be extracted directly from the measured data by performing a simple linear least squares fit using a polynomial of second degree (see Fig. 3)

This can be done with the aid of simple calculations. Moreover, the torque method allows the matching of the dynamometer load curve and the dynamic behaviour to road conditions, operating the dyno in the same way as in emissions- or fuel consumption testing.

Finally, the time required to prepare the vehicle with the torque equipment is now considerably reduced due to the fact that the torque wheels can be easily removed from one vehicle and attached to another using special adapters. Such adapters can be produced for each type of vehicle.

4. Apparatus

4.1 Torque Meters

The CF 22 group recommends a torque measuring principle which is based on determining the overall tractive torque. So far, there are experiences with two different concepts of applying the strain gauges technique.

4.1.1 Torque in the Drive Shafts

This concept makes use of the torsion in the drive shafts when the vehicle engine is working against the driving resistances. Thus, the (sum of the) torque in the drive shaft(s) is the measuring quantity. A certain disadvantage is the fact that the drive shafts have to be removed and reinstalled for the torque measurements and at present also have to be specially machined.

4.1.2 Torque in the Drive Wheels

In this case the torque is generally measured between the vehicle brake and the tire. The torque device may consist of two discs which are welded together at specified locations. One of these discs is provided with spokes to which strain gauges are applied. The measuring effect is the flexure of the spokes representing the forces experienced by the car wheels when running on the road or on a dyno. In order to give a brief impression of the technique used, a schematic diagram of such a device is displayed in Fig. 5.

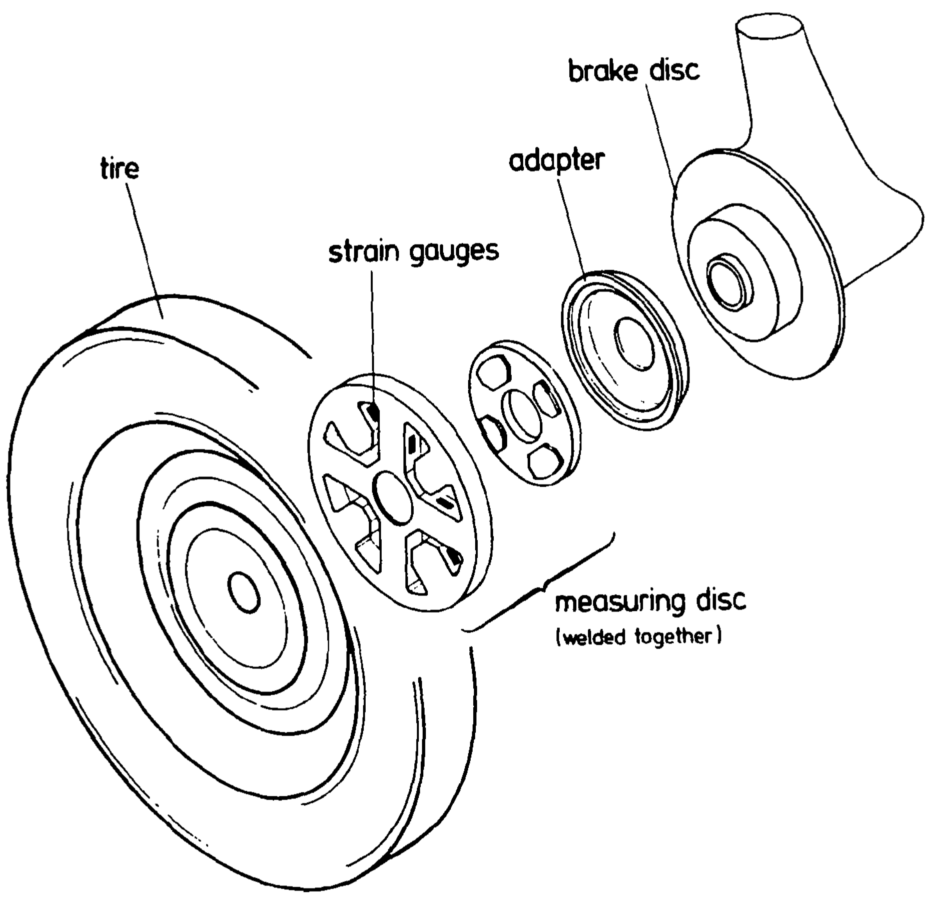


Fig. 5: Schematic Diagram of One Possible Way for Realizing a Torque Measuring Device. The torque is measured in the wheel between tire and brake. Such a torque device is mounted to each driven wheel.

This concept allows an easy installation of the measuring device because it is mounted to the wheels.
An application of this device to the various types of vehicles can be done be using adapters.

4.2 Electronic Equipment

The signals coming from the torque meter are amplified and processed into readable quantities. In the case of measuring the load curve - i.e. torque as a function of stepwise constant car speed - the amplified signals can be recorded for example on magnetic tape or strip chart recorder.

In the case of measuring the dynamic behaviour expressed as $\overline{M}$ the torque vs time function has to be integrated over the chosen driving cycle. One possible method is to store the analog signals on magnetic tape and integrate the torque function by a computer after the measurements.

A more convenient way of obtaining the desired $\overline{M}$-value is to use an electronic device for integrating the torque function. This can be done by a voltage to frequency converter which converts the torque signals (voltages) into pulses, the number of which per time unit is proportional to the height of the torque signals. These pulses are summed up by a pulse counting device. The final number of pulses is then divided by the time length of the driving cycle. The result is $\overline{M}$.

This technique has the big advantage that the numerical value of $\overline{M}$ is available immediately after the measurement. A block diagram of the equipment is shown in Fig. 6.

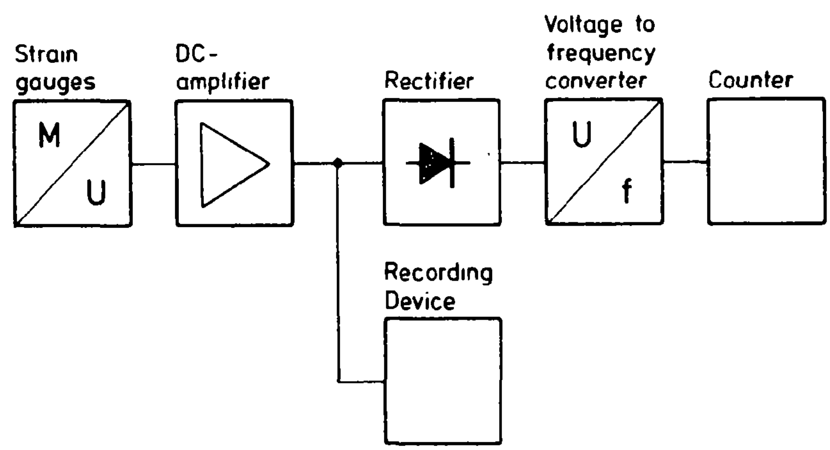


Fig. 6: Block Diagram of Typical Electronic Equipment for Torque Measurements.
For constant speed torque measurements the torque signal coming from the strain gauges bridge is amplified and directed to a recording device.
For integrated torque measurements the amplified and rectified torque signal is converted into pulses by the voltage to frequency converter.
Then the pulses are counted. The total number of pulses devided by the time interval (e.g. 485 s) represent the integrated torque $\overline{M}$.

5. Test Procedure

As described in Section 1 the recommended test procedure consists of measurements on the road and on the dyno. The driving resistances for the vehicle on the road should be reproduced on the dyno.

5.1 Measurements on the Road

The road should be straight and the grade - if any - must be known precisely so that the data can be corrected to zero grade. The grade must be constant within $\pm$ .05 % throughout the test section. The road surface shall be hard, smooth and dry.

During the measurements no interference from other vehicles should occur.

If any wind is present, the wind velocity should not exceed 2 m/s and it should be constant within $\pm$ 0.5 m/s. A reliable correction of the data can only be performed for steady winds in the direction of the test track. Nevertheless, the most reliable data are obtained without any wind being present. The tire pressure should be as recommended by the manufacturer.

5.1.1 Constant Speed Torque

A test vehicle which is representative for the vehicle type under study is driven on the road at constant speeds. Beginning at 20 km/h the speed is increased in constant intervals of 10 km/h covering a range of 20 km/h to 100 km/h. At each speed interval the torque must be kept constant. The torque pattern at the drive shaft(s) or in the drive wheel(s) is recorded.

This procedure should be repeated three times alternately in each direction of the road resulting in 6 sets of data points. Each set has to be corrected separately for possible grade and wind. The corrected data are combined and mean values as well as standard deviations of torque and speed are calculated for each data point. The result is a mean road load curve of the vehicle under study. This curve should be finally corrected to standard ambient conditions.

5.1.2 Integrated Torque over Variable Driving Pattern

In this dynamic procedure the mean torque value $\overline{M}$ is determined. This is accomplished by integrating the actual torque values with respect to time during operation of the test vehicle with a defined driving cycle. For this purpose the CF 22 group recommends e.g. the first 505 s of the US 75 cycle excluding the 20 s idle period at the beginning. The total time interval is therefore 20 s ... 505 s = 485 s. The integrated torque - obtained by one of the devices described in Section 4 - is finally divided by 485 s. The result is

$$\overline{M} = \frac{1}{485} \int_{20}^{505} M(t)\, dt.$$

5.2 Adjusting the Dynamometer

Here, it should be pointed out once more that the recommended torque procedure is applicable to all known types of chassis dynamometers. This method can substantially improve the adjustment of a dyno.

5.2.1 Constant Speed Adjustment

The dynamometer is warmed up according to the manufacturers' recommendation. The tire pressure of the test car is increased to a value of 3 bar at ambient test conditions and the car is placed on the dynamometer. Now we have to distinguish between the two classes of dynamometers - the one which allows a perfect reproduction of the road load curve and the one which does not.

5.2.1.1 Dynos Adjustable to Road Load

Generally, with this class of dynos we can match road load curves. The (corrected) averaged torque curve measured on the road is reproduced on the dynamometer by varying its adjustment parameters (s. Fig. 7).

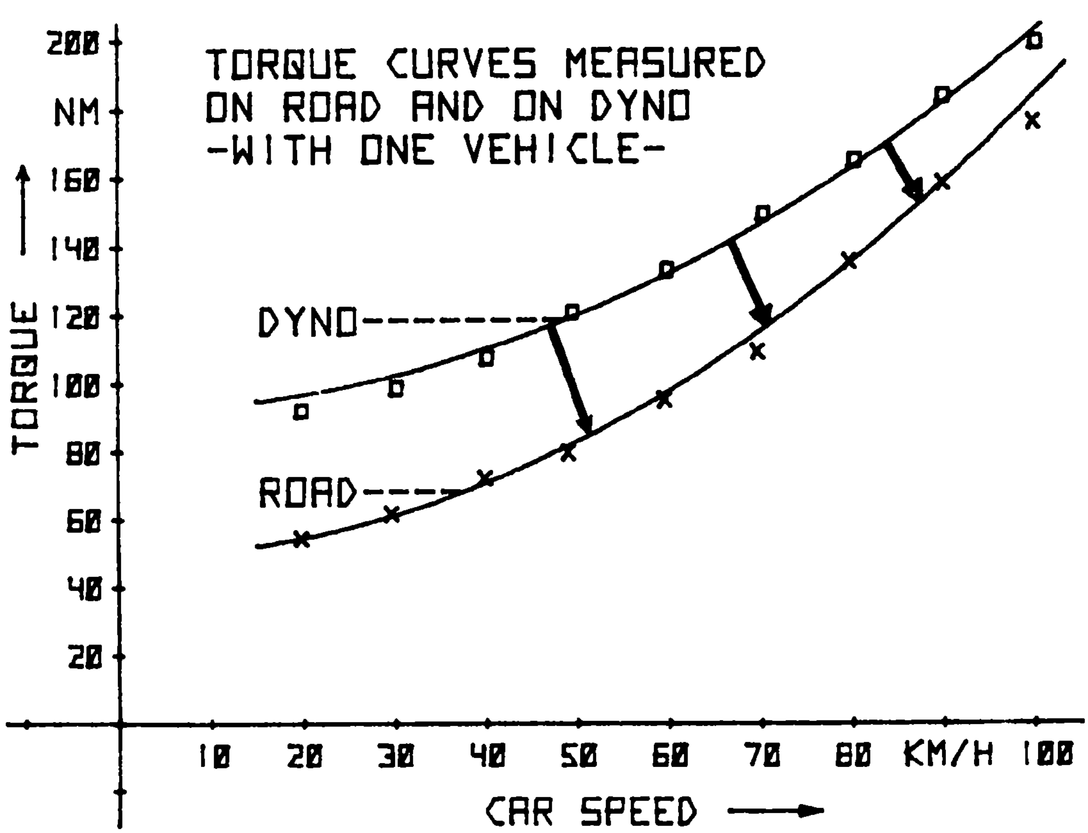


Fig. 7: Adjusting the Dynamometer Torque Curve to the Road Torque Curve. One and the same test vehicle was used in both cases. The adjustment is carried out by varying the rolling resistance + internal friction term (a_0) and the air resistance term (a_2) of the dynamometer PAU control circuit. For DC machine brakes and modern eddy current brakes a perfect adjustment can be achieved.

5.2.1.2 Dynos with Limited Adjustment

Generally, with this class of dynamometers we cannot achieve an adjustment to road load, because only the air resistance term can be varied. Therefore, the CF 22 group recommends a compromise for this case. The compromise consists of adjusting the dynamometer in two points of velocity - at 40 km/h and at 80 km/h. The remaining deviations between road torque and dyno torque in these two points should not exceed 15 %, i.e. $\Delta M \leq 15$ % (s. Fig. 8).

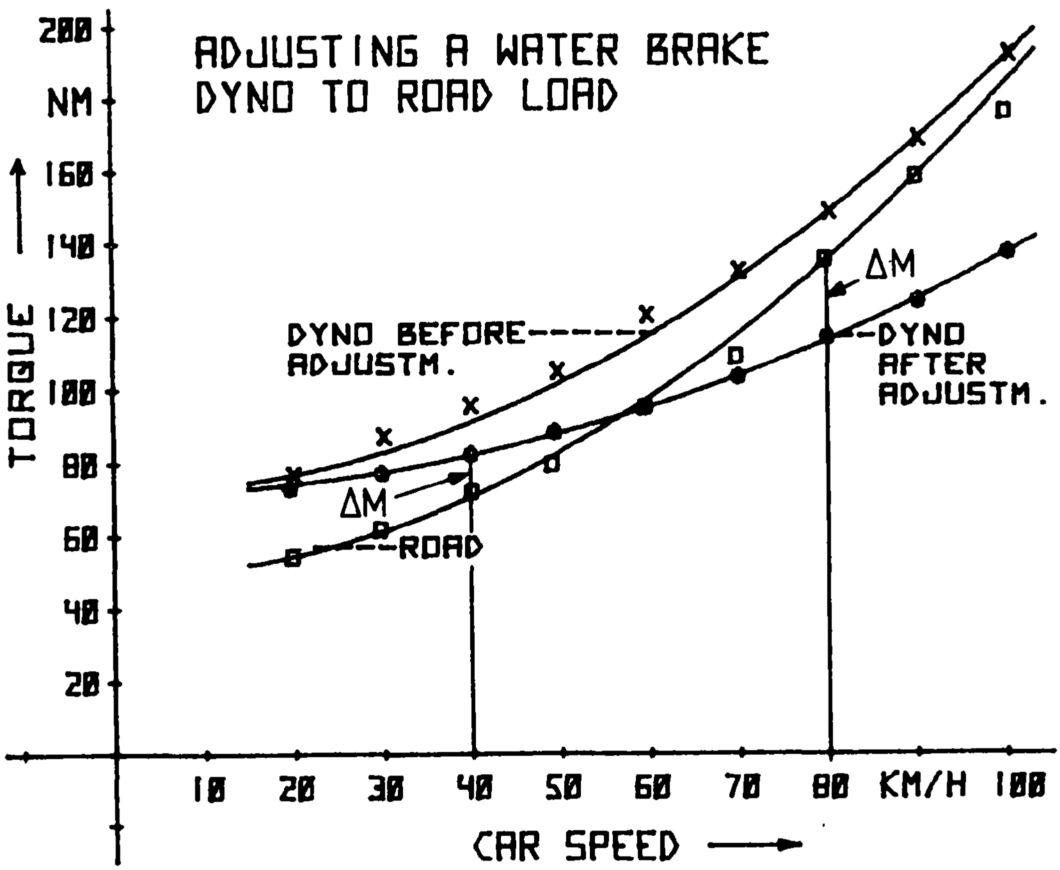


Fig. 8: Compromise for Adjusting the Dynamometer Torque Curve to the Road Torque Curve.
The dyno curve is adjusted in two points of velocity (40 km/h and 80 km/h). For these points ΔM should not exceed 15 %. In this case the $\overline{M}$-values of the two curves (dyno after adjustm. and road) were then in perfect agreement.

5.2.2 Integrated Torque Adjustment

After the stationary adjustment of the dynamometer the dynamic procedure is carried out. This means the adjustment of $\overline{M}_{dyno}$ to $\overline{M}_{road}$.

Following the derivations of Section 1

$$\frac{\theta_{roll} + \theta_{sim}}{R} = rm^* \qquad \text{, the adjustment parameter of the dyno}$$

Using this equation, θ_{sim} can be calculated from the effective mass of the test vehicle and from the dynamometer parameters (θ_{roll}, R). In principle, it is possible to achieve any precision with respect to $\overline{M}_{dyno} = \overline{M}_{road}$ provided that the increments of flywheel masses available at the dynamometer are small enough and provided that a matching of the dyno load curve to road load curve was carried out. As discussed in section 2 this is generally only possible with DC and eddy current dynamometers.

5.2.2.1 Dynos Adjustable to Road Load

In the case of such dynamometers the CF 22 group recommends the following steps to achieve $\overline{M}_{dyno} = \overline{M}_{road}$:

(1) Constant speed adjustment of dyno load curve to road load curve (s. 5.2.1.1).

(2) Calculation of vehicle equivalent inertia by using

$$\frac{\theta_{roll} + \theta_{sim}}{R} = rm^*$$

(3) Normally, the calculated inertia implies $\overline{M}_{dyno} = \overline{M}_{road}$ - within measurement precision (≈ 1 %). This, however, should be checked by the torque method. If the measurement yields a slight deviation - possibly due to dynamometer tolerances - from $\overline{M}_{dyno} = \overline{M}_{road}$, a fine adjustment of the dynamometer inertia should be carried out. In practice it should be checked whether a better agreement could be achieved by varying the flywheel masses available. In most cases a variation of $\pm$ 125 lbs = $\pm$ 56.7 kg will be sufficient, so that the flywheel increments available should be 125 lbs. A fault in the dynamometer performance - e.g. a flywheel disengaged or a faulty bearing of the non-coupled flywheels - generally causes a considerable discrepancy between $\overline{M}_{dyno}$ and $\overline{M}_{road}$. Such a case is immediately detected by the torque method and that under realistic conditions, which means that the dynamometer is loaded and operated (driving cycle) in the same way as during normal exhaust emissions - or fuel consumption testing, <u>i.e. under equal dynamic conditions.</u>

5.2.2.2 Dynos with Limited Adjustment

In the case of a dynamometer with limited adjustment (e.g. water brake) the CF 22 group recommends the following procedure:

(1) Constant speed adjustment within 15 % of the dynamometer load curve to road load curve in two points of velocity (s. 5.2.1.2).

(2) Calculation of vehicle equivalent inertia θ_{sim} by using

$$\frac{\theta_{roll} + \theta_{sim}}{R} = rm^*.$$

(3) Now, the calculated inertia generally does not imply $\overline{M}_{dyno}$ = $\overline{M}_{road}$ - due to the more or less good agreement between the dyno load curve and the road load curve. In practice, the deviation between $\overline{M}_{dyno}$ and $\overline{M}_{road}$ is not very high, provided that the dyno curve has been adjusted in the two points. In principle, there are two possibilities for achieving $\overline{M}_{dyno} = \overline{M}_{road}$:

(i) If, for example, $\overline{M}_{dyno} > \overline{M}_{road}$, the dyno curve in Fig. 8 could be lowered until $\overline{M}_{dyno} = \overline{M}_{road}$. However, this would normally result in inadequate fuel consumption test results because the load curve setting of the dyno has a considerable effect on fuel consumption. This is due to the fact that e.g. the "Highway Driving Cycle" consists mainly of more or less constant speed driving.

(ii) Therefore, the CF 22 group recommends a final adjustment of $\overline{M}_{dyno}$ to $\overline{M}_{road}$ by varying the inertia of the dyno. Again, flywheel increments of 125 lbs = 56,7 kg should be sufficient.

6. Evaluation and Reporting of Results

6.1 Constant Speed Torque

According to the stationary procedure the main task in the data analysis of the road measurements is the extraction of the rolling resistance and the air resistance of the vehicle under test. The three data sets in each direction of the test track have to be corrected for possible grade and wind. This must be done prior to combining the 6 data sets to one mean road torque curve.

(i) If the test track has a non-zero grade the term $rmg \cdot \sin\alpha$ (s. section 1) has to be added/subtracted to/from each data point (torque value). If the 6 different curves are combined to a mean curve prior to correcting each curve separately this would also result in the same curve but the standard deviation for each data point would be considerably higher.

(ii) Only steady winds of less than 2 m/s in the direction of the test track are allowed during data collection (s. section 5). The wind effect cannot be eliminated by simply combining the data sets for the different directions. This can easily be seen.
The resulting velocities which are decisive for the air resistances in both directions are:

$v_{1res}^2 = (v + w)^2$ and $v_{2res}^2 = (v - w)^2$, where w is the wind velocity.

Thus, the mean velocity for both directions becomes

$\frac{1}{2}(v_{1res}^2 + v_{2res}^2) = v^2 + w^2$, i.e. not equal to v^2.

So, each car velocity must be corrected separately for the wind influence.
The still remaining problem is the constancy of the wind conditions, i.e. its velocity and direction. Therefore, it is best to perform the measurements without any wind being present if that is possible.

After having corrected the data for wind and grade a least squares fit of the form $M = a_o + a_2 v^2$ is carried out. The coefficients a_o, a_2 and their uncertainties are obtained through this fit. a_o/r represents the rolling resistance and $a_2 v^2/r$ the air resistance. Here, it is assumed that in the given velocity range (20 km/h to 100 km/h) the rolling resistance is independent of car velocity (s. section 1).

Finally, the rolling and air resistances must be still corrected to standard ambient conditions. Particularly, this is important for the air resistance $R_a = \frac{\rho}{2} c_a A v^2$ because of the temperature and pressure dependence of the air density ρ. As standard ambient conditions a temperature of T_o = 293.2 K = 20 °C and a barometric pressure of p_o = 101.32 kPa = 760 Torr is usually presumed. In order to reduce "arbitrary" ambient conditions - given by T, p - we have to apply the well known formula:

$$\rho_{T_o,p_o} = \rho_{T,p} \frac{T}{T_o} \frac{p_o}{p} \quad (\mathrm{kg/m^3})$$

ρ_{T_o,p_o} = air density at T_o = 293.2 K, p_o = 101.32 kPa

$\rho_{T,p}$ = air density at T , p

For example, let us assume ambient conditions of T = 273.2 K and p = 103.99 kPa (= 780 Torr) during the measurements on the road. In this case the correction for air resistance amounts to approximately 10 %. This example was chosen in such a way that both corrections - for temperature and pressure - go into the same direction.

For the rolling resistance only a temperature correction is necessary. The correction is carried out by applying the formula for reducing the rolling resistance $R_{R,T}$ (at ambient temperature T) to the rolling resistance R_{R,T_o} at T_o = 293.2 K:

$$R_{R,T_o} = R_{R,T} \; (1 + 0.0018 \; (T - 293.2)), \text{ i.e.}$$

0.18 % per ΔT = 1 K.
This correction formula is also recommended by EPA (AC 55 A).
For our example of T = 273.2 K we obtain therefore a correction of approximately 4 % with regard to the rolling resistance $R_{R,T}$, drawn from the measured data; i.e. the measured $R_{R,T}$ is 4 % higher than R_{R,T_o}.

Having done all the above corrections we obtain the rolling and air resistances and their uncertainties reduced to zero wind, zero grade and standard ambient conditions.
With these values a reduced average torque curve - which is the mean of 6 single test runs - can be plotted. This curve is the reference curve for adjusting a chassis dynamometer to road conditions.

6.2 Integrated Torque over Variable Driving Pattern
The various corrections discussed under 6.1 are much less important for the dynamic measurements. This can easily be seen from the equation of motion (s. section 1):

$$\frac{M}{r} = F = R_R + R_a + R_i$$

When driving the 505 s cycle R_i dominates over R_R and R_a.
To get an impresssion of the order of magnitude let us assume for example a typical car with the following technical data:

m = 1 000 kg
m*= 1 040 kg
r = 0.3 m
A = 2 m²
c_a= 0.4
f_R= 0.015

Furthermore, we need the average speed and average acceleration during the 505 s cycle.
The values are: $\bar{v}$ = 49.9 km/h, $\bar{\dot{v}}$ = 0.62 m/s²
(excluding the standstills).
With these data we can now calculate an approximate average value for the force (or torque) over the 505 s cycle.

As an example let us determine the influence of the ambient conditions - in a reasonable range. Again, we assume T = 273.2 K and p = 103.99 kPa.

$$F_{T_o,p_o} = 9810 \cdot 0.015 + \frac{1.202}{2} \; 0.8 \cdot 13.86^2 + 1040 \cdot 0.62 \; (N)$$
$$= 884 \; (N)$$

$$F_{T,p} = 152.7 + \frac{1.324}{2} \; 0.8 \cdot 13.86^2 + 645 \; (N)$$
$$= 899 \; (N)$$

The difference between these two values of F for the assumed ambient conditions amounts to Δ = 1.7 %. The effect is so small that it usually can be neglected, but if desired the correction can be carried out.

From 6 mean torque values $\bar{M}$ an average value and its standard deviation is computed. This average mean torque value is then used for the dynamic calibration of the chassis dynamometer.

7. Precision of Torque Method

7.1 Constant Speed Torque

The precision of determining the coefficients a_o and a_2 of the rolling resistance and air resistance from the least squares fit applied to the corrected measurement data is better than $\pm$ 3 % in the case of rolling resistance and better than $\pm$ 2 % in the case of air resistance.
This situation is displayed in fig. 9 where typical measurement data are plotted in the usual way. In addition to the "least squares fit" curve two further curves

$M_+ = (a_o + \Delta a_o) + (a_2 + \Delta a_2)\, v^2$ and

$M_- = (a_o - \Delta a_o) + (a_2 - \Delta a_2)\, v^2$ are plotted, where Δa_o and Δa_2 represent the uncertainties of the coefficients. In this case the uncertainties amount to Δa_o = 2.2 % and Δa_2 = 1.5 %.

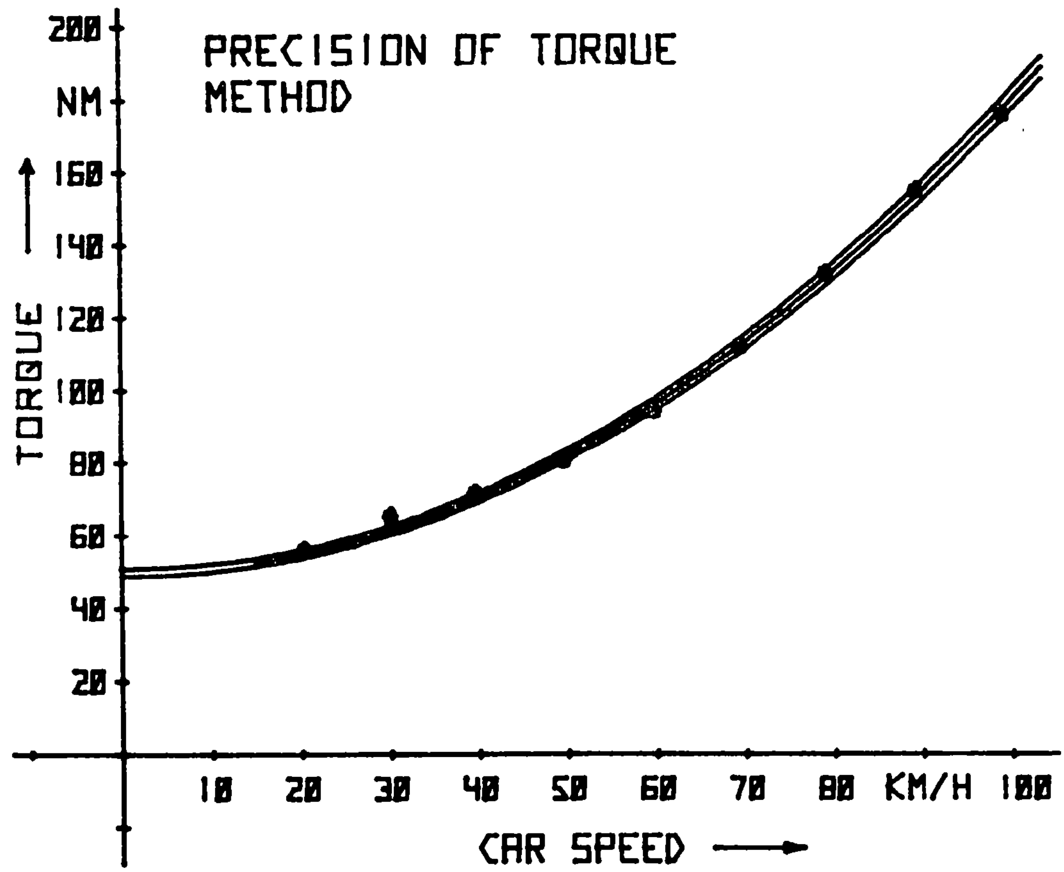


Fig. 9: Typical Torque Measurement Data.
The curve ($M = a_o + a_2v^2$) in the middle of the three curves is drawn from a least squares fit applied to the corrected data. The two other curves are:

$M_+ = (a_o + \Delta a_o) + (a_2 + \Delta a_2) v^2$ and $M_- = (a_o - \Delta a_o) + (a_2 - \Delta a_2) v^2$

Here: $\Delta a_o = \pm 2.2$ %, $\Delta a_2 = \pm 1.5$ %, which are resulting from the least squares fit.

Figure 10 demonstrates the precision which can be achieved when adjusting an electric brake dynamometer (upper curve) to road load. The two lower curves are the road load curve and the dyno curve after adjustment.
A difference between the two curves is hardly to be seen (Difference in a_o: 1 % , difference in a_2: 3.7 %)

Preliminary results for one car in the case of exhaust emissions and for four different types of cars in the case of fuel consumption indicate that the differences in mass emissions and fuel consumption measured in the city driving cycle when using the two different adjustments (upper curve and lower curves) are considerable (Δ fuel consumption ≈ 8 %, Δ NOx ≈ 18 %). For other types of cars the situation might be different. Here, still further results are needed.

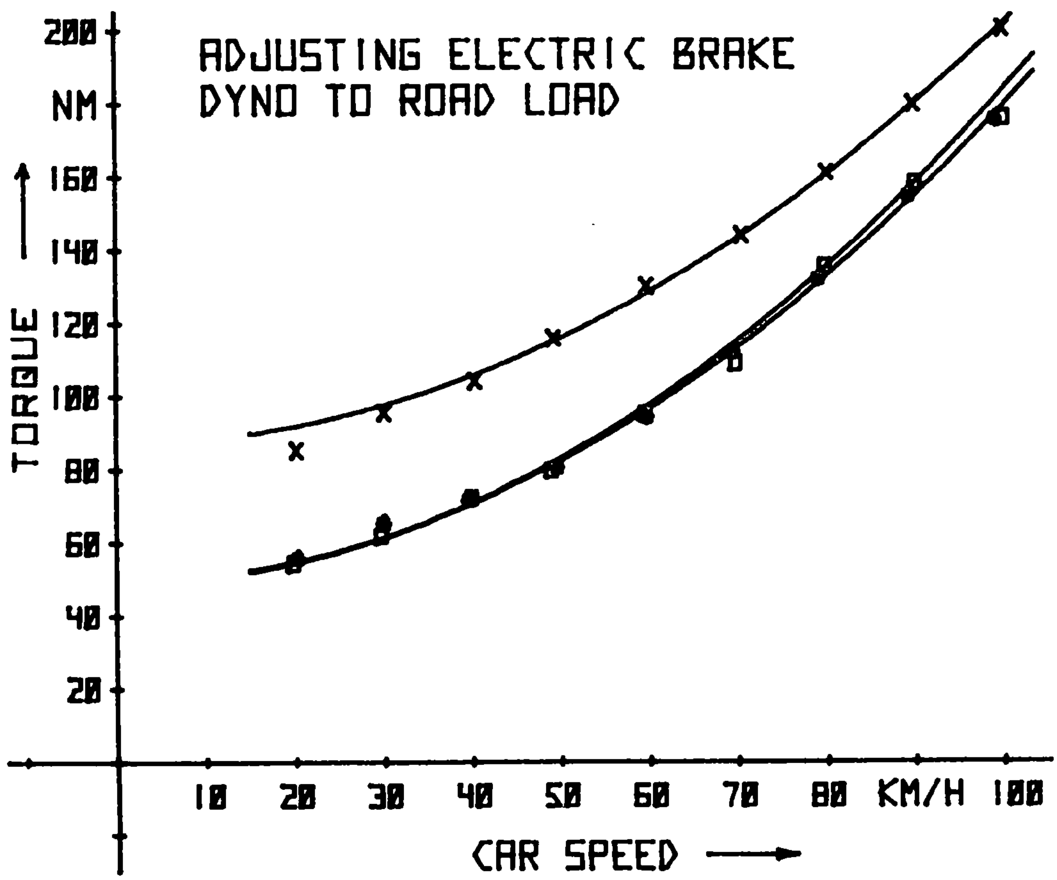


Fig. 10: Precision of Adjusting Dyno Load Curve.
Upper curve: Before adjustment.
Lower curves: Road load curve and dyno curve after adjustment.

7.2 Integrated Torque over Variable Driving Pattern

The precision of the dynamic measurements can be expressed in form of the standard deviation of $\overline{M}$-values found in repeated measurements. This was done for several sets of 10 typical measurements. The average standard deviation amounted to $\pm$ 1 %.

8. Summary and Conclusion

It is recommended by the CF 22 group that the calibration of chassis dynamometers for fuel consumption and exhaust emission measurements should be performed in such a way that the load behavior of the vehicle on the road is reproduced on the dynamometer.

- The best reproduction of actual road load conditions can be achieved only with electric brake dynamometers. Therefore, if possible, the use of this type of dynamometer is recommended. In addition, the dyno should be equipped with discrete flywheels of 125 lbs increments.

- In the case of water brake dynos a compromise which consists of adjusting these dynos in two points of velocity with defined tolerances can be performed. In this way, the current method of using only one point of adjustment is substantially improved.

- The current ECE procedure of setting the dyno load according to inertia weight classes has some fundamental disadvantages because it does not take into account the particular air resistance of a special vehicle type. The weight classes are only a rough approximation to reality. Even the driving resistance (i.e. PAU loading) calculation method,based mainly on the frontal area of the vehicle (as now used by the US-EPA) cannot really reflect the specific aerodynamic conditions of individual vehicle types.

- Concerning the coast down procedure recommended by EPA it has to be pointed out that this method does not yield the desired calibration quantities - i.e. the coefficients a_o and a_2 - in a direct way. Certain non-elementary mathematical operations such as fitting the measured speed vs time data to a non-linear function must be carried out. In contrast to this, the torque method yields the load behavior of the vehicle as a direct result of the measurements.

- Finally, the CF 22 group recommends that the use of the torque measuring principle is widely accepted by government as an independent alternative for dynamometer calibration.
  Furthermore, the group recommends the acceptance of this proposal as a basis for a standardized procedure.

ATTACHMENT VII

DAIMLER - BENZ

SUBMISSION NO. 3 TO EPA

"Detailed description of Daimler-Benz torque measuring device (torque disc) with instrumentation for road load determination in the vehicle's driving wheels and description of latest on board measurement evaluation system."

V1MA
8-16-79

Wheel Torque Measuring Device for Adjustment of Chassis Dynamometer used in Exhaust Emission and Fuel Economy Testing.

Summary:
Under the impact of steadily increasing demands in the field of vehicle exhaust emissions and fuel economy testing, continuous further refinement of existing or development of new and sophisticated instrumentation and measurement technology are necessary. In the chain of measurement uncertainties, chassis dynamometers have gained increased importance, since they have to simulate road load conditions, i.e. rolling and air resistance of vehicles during the above testing.
A special device was developed which allows direct measurement of the force exerted from the driving wheels of a vehicle to the road and reproduction of this force - measured as torque - on a chassis dynamometer.

After a test run on the road, the power absorption unit of the chassis dynamometer can be so adjusted to precisely simulate "what the vehicle sees on the road".

1. Introduction

The steadily increasing demands in the field of exhaust emission control and fuel economy testing require the application of more and more qualified measurement-, instrumentation- and evaluation techniques in order to obtain reliable test results.

Realizing that the torque in a vehicle's drive train represents a suitable physical means for realistic adjustments of chassis dynamometers for exhaust emission- and fuel economy testing and taking into account that the precision of torque measurements in the drive shaft (s) may be adversely affected by undefined friction losses in the differential or wheel assembly, Daimler-Benz developed a torque disc for application to the vehicle's drive wheels. Fig: 01 Fig: 02

The following criteria were set up for the development of this torque disc:

- contact-free measurement signal transmission
- possibility of electrical calibration
- integrated rpm pick-up
- resolution of low torque values
- high precision
- acceptance of high radial- and side forces without influencing the torque signal
- compact design with minimal increase of tread width
- applicability of standard rims
- applicability to non-Daimler-Benz vehicles
- easy installation to vehicle

Daimler-Benz's experience from earlier design work and testing with similar devices in the field of brake torques for different kinds of vehicles could be advantageously used in the development work of the new wheel torque disc.

- 3 -

A prototype of the new device was displayed on a demonstration test stand on the Hannover Fair in 1978.

2. Torque Measurement Device

2.1 Design and working principle

The torque measuring device mainly consists of the torque disc and the signal transmitter. The torque disc itself is formed by two portions, one connected to the drive shaft, the other connected to the rim. Both parts are welded together at the circumference by an electronical beam. The disc portion connected to the drive shaft has 5 bores for mounting the device to the brake disc and 32 measuring bores resulting in 32 webs acting like spokes of a wheel. Of these 32 webs, 16 are equipped on both sides with strain gauges, which are located at the place of maximum bending force, when forces in circumferential direction are applied. Fig.: 03

In axial direction, the cross section of these webs is substantially larger than in circumferential direction. The webs are therefore capable to stand high radial and axial forces. On the other hand, the webs are relatively flexible in circumferential direction as intended.

All 32 strain gauges are electrically combined to a bridge (R = 240 Ω), the connection of which leads to a rotation transmitter located in the center of the disc. Fig.: 04

Axle load, side forces and temperature related influences are largely eliminated by design-, construction- and electrical means, so that only torque related forces are measured and transmitted. Besides the earlier mentioned length to width ratio of the webs, the number of equally distributed measuring webs should be large and divisible by 4 in order to eliminate signal oscillations.

The dimensions and surfaces of the bores/webs require defined conditions. Further, a precise application of the strain gauges, optimal arrangement of each individual strain gauge within the bridge and - if needed - temperature compensation elements have to be considered.

Due to the very compact design of the torque disc Fig.:05
only a minor tread width increase (32 mm) occurs Fig.:06
and standard rims can be used.

The inductively working rotation transmitter Fig.: 04
consists of an input (feed) transmitter (coil 1= stationary and coil 2 = rotating) and the output (measurement signal) transmitter (coil 3 = rotating) and coil 4 = stationary). It transmits feed voltage and measurement signal to or from the strain gauge bridge located on the rotating torque disc. By means of a special computer program, the geometry and electrical outlay of each individual inductive rotating transmitter are optimized for the required measurement range.

This optimization, which is done at 5 kHz carrier frequency, considers transmission behavior and individual bridge resistance of the torque sensing device.

An important part for the electrical calibration of the measuring chain (sensor -amplifier- recorder, etc.) is the calibration switch incorporated in the inductive rotation transmitter. This device, coil 5 (stationary) and Reed contact (rotating) by exciting of coil 5 (12 V=) switches a certain resistor RK parallel to a quarter branch of the bridge and thus electrically decalibrates the full bridge. This calibration signal in relation to the output signal s~~t~~imulates a defined torque (A) when a certain torque (B) is mechanically applied to the disc. The defined torque (A), which may be 30 % to 100 % of the nominal torque (C) (in our case 50 %), can be achieved by choosing the value of resistor RK.

For wheel rpm pick-up, a light electric rpm counter, delivering 60 impulses per wheel revolution, is additionally incorporated.

2.2 Production

Most of the parts are manufactured out of stainless steel with the disc itself consisting of high alloy spring steel which is tempered to a strength of 2200 N/mm^2.

Precise production with intermediate checks and tests of the mechanical and electrical components guarantees a high quality of the device.

2.3 Calibration

After production, each device is subjected to a final test and calibration. For this torque calibration, a specially developed calibration test bench with aerostatic bearings ranging to 1 kNm was set up. This test stand is connected to an automatic data recording system and a computer, which evaluates and immediately plots non-linearity, sensitivity and calibration value. All test data are kept in a protocol and archive.

2.4 Technical Data

Nominal Torque (C)	:	$M_t = \pm$ 500 Nm
Overload	:	M_{tmax} = 100 %
Hysteresis	:	$<$ 0.3 % of (C)
Sensitivity	:	$\approx$ 0.01 mV/Nm (at 5 V feed voltage)
Calibration Value	:	50 % of (C)
Rpm Sensor	:	60 imp/wheel rev.
Carrier Frequency	:	5 kHz
Weight	:	8.4 kg

3. Measuring and Evaluation System

3.1 Measurement Set-Up

Since the torque discs are designed for application in chassis dynamometer adjustments by reproduction of actual road load forces, the complete measurement set-up has to be mountable into the vehicle with on board electrical power supply. Fig.: 10

Since further a subjective evaluation of rpm/ speed or torque strip charts includes many uncertainties, the evaluation should be done by the system itself. This is possible by application of a computer.

Fig. 07 shows a set-up which has proven high reliability during many tests on track or dynamometer. The wheel revolution impulses are transmitted via a frequency/voltage converter with attached low-pass amplifier to the inlet of a multiprogrammer. The torque discs in the vehicle's driving wheels are connected to 5 KHz carrier frequency amplifiers,the outlet voltages of which are summed up in an addition-subtraction amplif r directed via a low-pass amplifier to another inlet of the multiprogrammer as the sum of 2 torque values. The multiprogrammer is governed by a computer which takes care of all required evaluations of the measured values. Finally, the results of these evaluations are transmitted to a plotter: An immediate, complete on board test evaluation has taken place.

3.2 Adjustment of measurement system

The adjustment starts before each measurement with the vehicle lifted up (tension-free) and consists of the earlier mentioned defined electrical decalibration of the strain gauge bridges on the torque discs by means of the calibration switch.

The so simulated torque with its known value serves as a mean for adjusting the whole measuring chain from the disc through the computer to the plotter.

Correspondingly, a defined frequency is introduced before the frequency/voltage converter of the rpm channel, which simulates a certain wheel rpm value. Since in a preliminary test with loaded car and adjusted tire pressure the amount of impulses over a well known distance on the road was determined, each impulse frequency can be correlated to a certain vehicle speed.

4. Measurement and Evaluation

According to Daimler-Benz's experience, gained over the last two years during road load determination testing by constant speed driving and simultaneously computerized evaluation, it is essential to define the "constant speed" condition. Fig.: 09

It finally was decided to define this term first as a condition where as well the vehicle speed as the (sum of) torque do not exceed a certain spread. This definition represents a desired quality/acceptance criteria for the measurement points. Second, a limit was set for the drift of vehicle speed and (sum of) torque over a certain measurement time.

Both limits were set according to experience as follows:

Max. spread of vehicle speed for tests on the road/test track:

$\pm$ 0.2 Kph

max. spread of (sum of) torque for tests on the road/test track:

$\pm$ 7.5 Nm.

The latter spread allowance is increased at very low vehicle speeds.

Max. allowable drift of vehicle speed for tests on the road/test track:

0.05 Kph/8s

Max. allowable drift of (sum of) torque for tests on the road/test track

1 Nm/8s

For tests performed on the chassis dynamometer, these tolerances can be further narrowed.

During the trial, the computer tests each measurement and shifts the detected momentary values of vehicle speed and torque in a pre-programmed slide storage. With every new pair of data available, the computer checks whether the above described tolerances are met for all stored data pairs.

If this is not the case, the data pair, which was stored longest, will be shifted out of the storage when another data pair becomes available.

The check for meeting the tolerances then starts again and this routine continues until all criteria are met, i.e. until the desired constant speed is reached. At this moment, the average speed $\bar{V}$ and average torque $\bar{M}$ as well as the standard deviation s_V and s_M are calculated for all data pairs in the storage, and the results are transferred into the storage for final results.

The slide storage content is then erased, and a new search run starts.

In practice, it has proven useful to start this search run only above 5 Kph constant speed. Further, there is the possibility to interrupt the search run for a time, if necessary. The constant speed points of a road load curve can be approached in free order.

As soon as enough valid measurements are available in the result storage, pushing a button terminates the search runs and switches the program over to calculation. According to the least square method, the best fit curve is calculated and plotted through the measurement points using the formula

$$M = a \cdot v^b + c \cdot v + d$$

If needed, the formula

$$M = a \cdot v^2 + c \cdot v + d$$

can be used as well.

The computer prints the coefficients of these equations together with the corresponding average quadratic error.

The plotter delivers a graph of the road load curve together with the test conditions, which have to be given into the computer by hand.

If desired, the numeric data of the individual speed points (V and M) together with their standard deviations are printed out as well.

An additional program allows for immediate determination of average road load curves calculated e.g. from several test runs in one and several test runs in the other direction of a test track or road.

Further, all driving resistance curves as plotted after the test run can be corrected to standard conditions by giving track grade, barometric pressure ambient temperature and wind conditions into the computer. It must be noted, however, that Daimler-Benz restricts test runs with the system to practically zero wind conditions, if at all possible.

In order that this correction can be performed, knowledge of the vehicle's ($c_w \cdot A$)-term is essential if not a mere quadratic characteristic of the driving resistance curve is assumed.

Fig. 08 shows an example for a road load resistance curve established with the above described system from driving at constant speed points in two track directions. An excerpt from the computer print out is shown as well.

5. Other possibilities of the system

The Daimler-Benz torque disc and evaluation system can be used for further testing purposes as already proven in practice, e.g.:

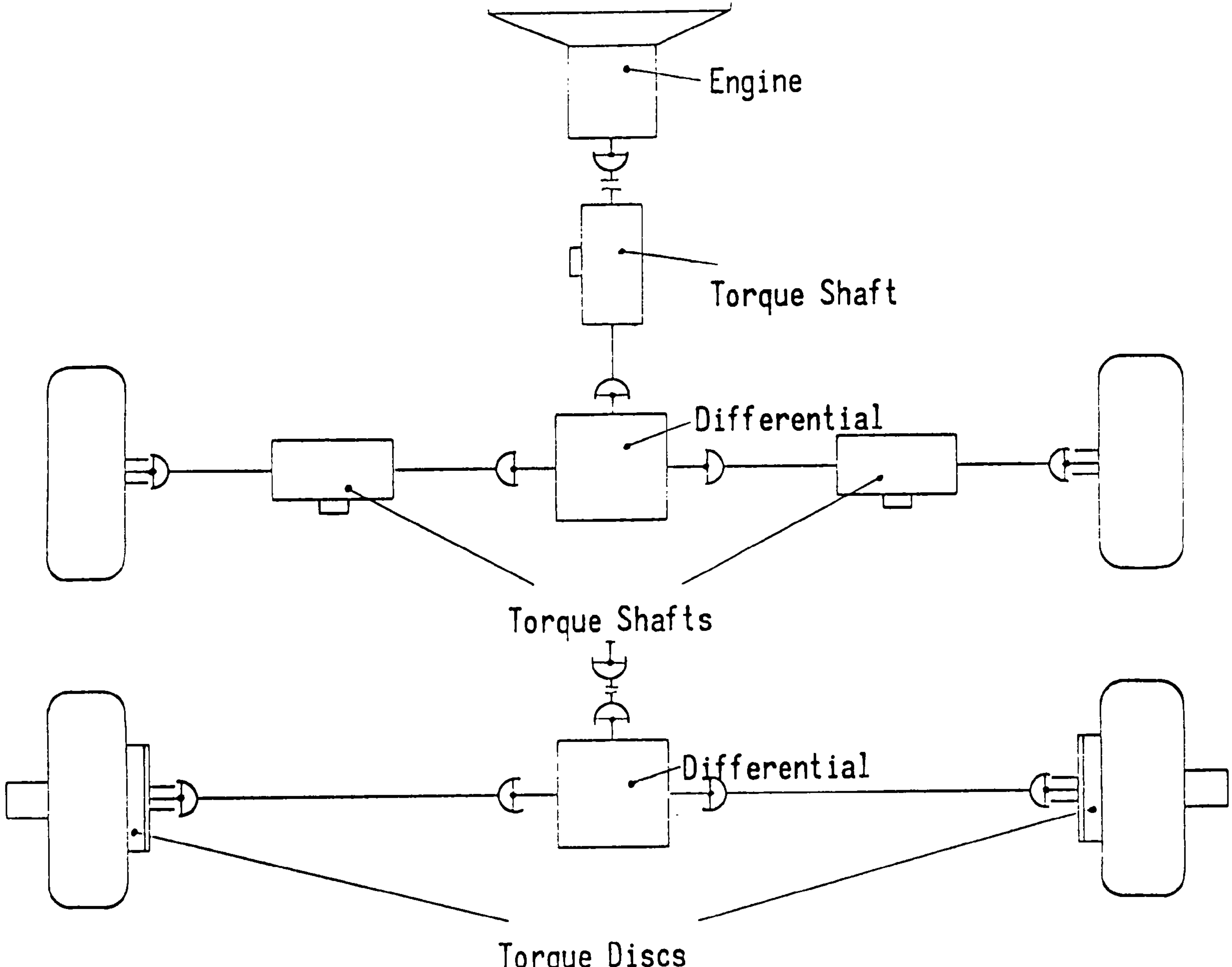
- determination and check of the chassis dynamometer's inertia mass(es),

- determination of the sum of positive torque or energy transferred via the vehicle's wheels during a complete test cycle or portions of it,

- separation of rolling resistance from acceleration/ deceleration portions of driving forces at constant speed (on board evaluation),

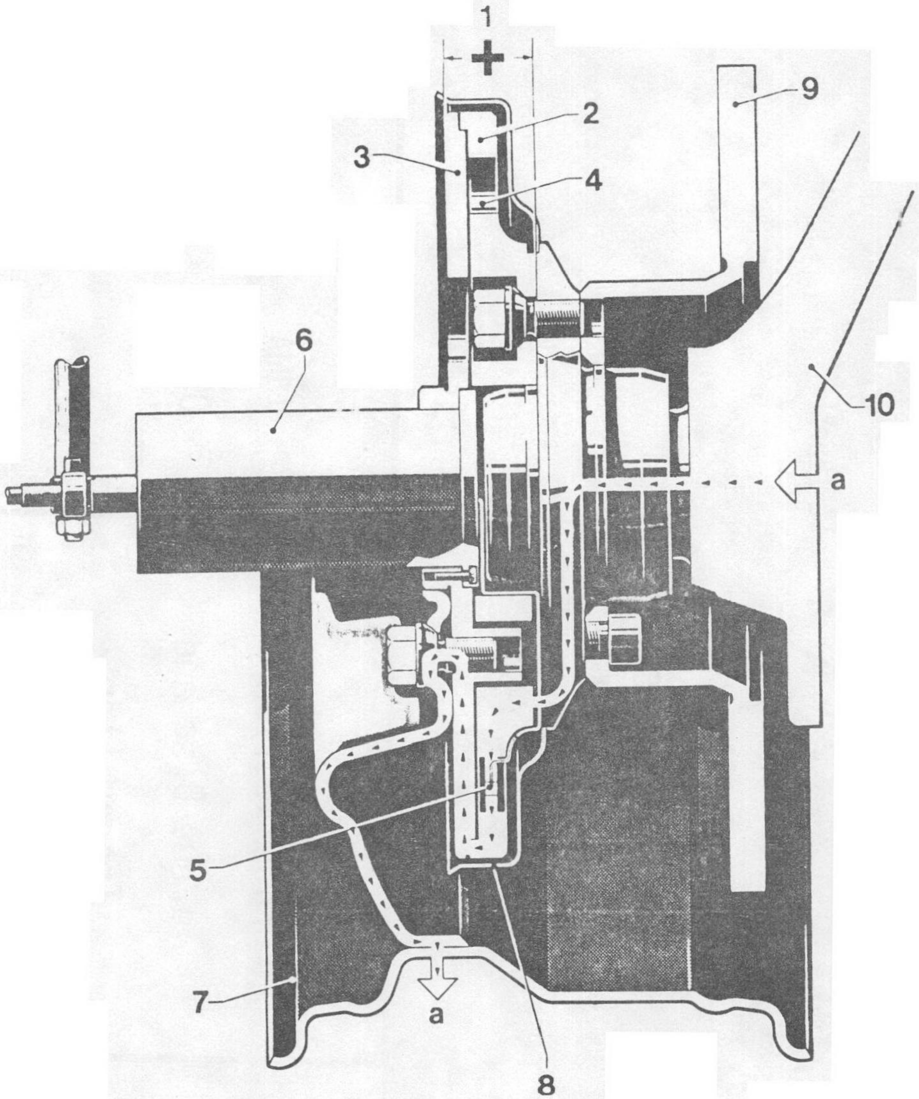
- comparison of dynamic behavior of different dynamometer types or routine check of the same dynamometers over time (on board evaluation)

Details and results of these investigations will be published in the near future.



Possibilities of Torque Measurements
in a Vehicle's Drive Train

V1MA 0879
01



1 Increase of Tread Width by Installation of Torque Disc

2 Torque Disc (portion connected to drive shaft)

3 Torque Disc (portion connected to wheel)

4 Bolt Circle with 32 Bores (32 webs, 16 webs with strain gauges)

5 Measuring Web with Strain Gauge

6 Inductive Rotation Transmitter

7 Standard Rim

8 Protection Cover

9 Brake Disc

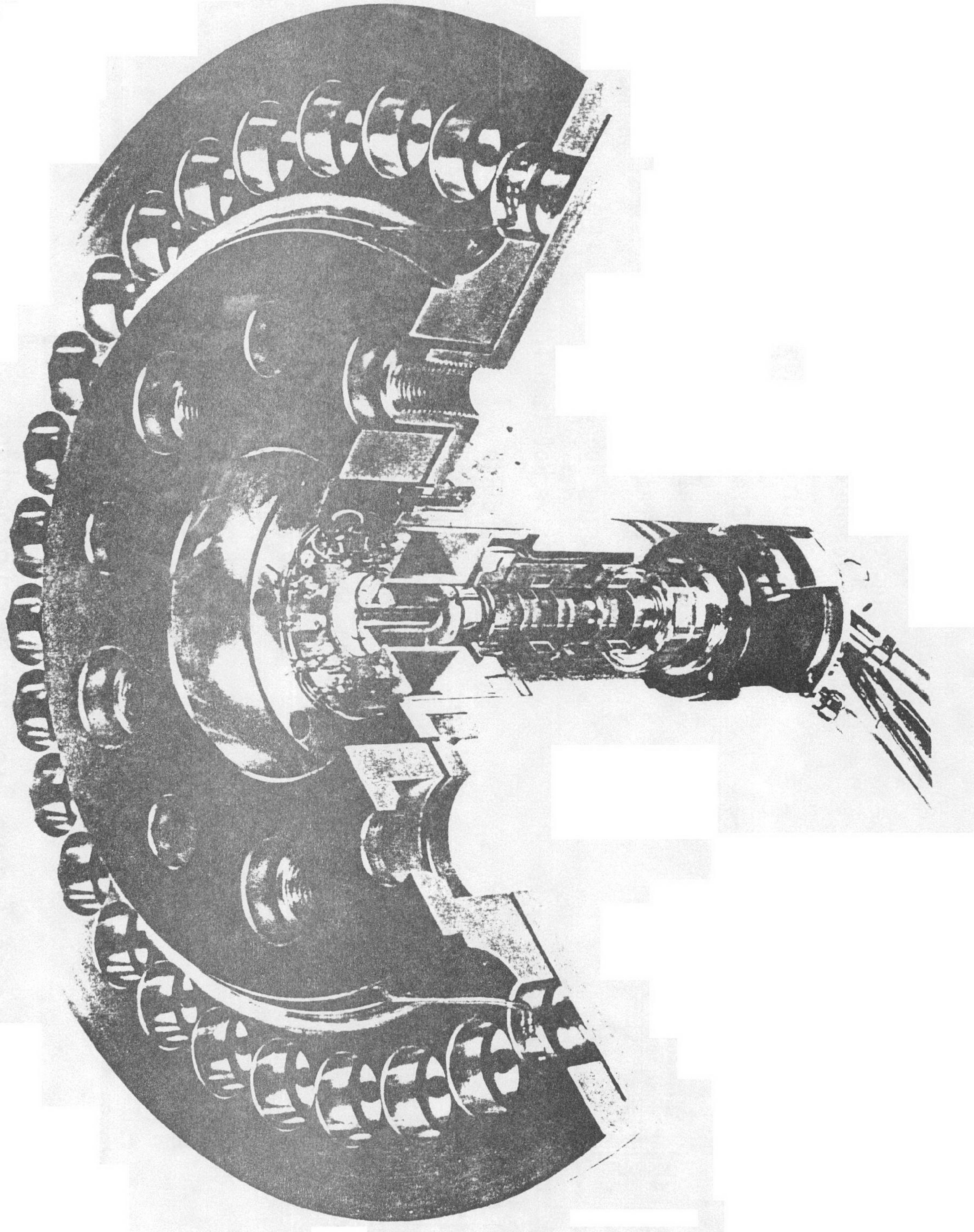
10 Wheel Suspension

a-a Flow of Driving Force

Schematic of Torque Disc and its Connection to the Wheel of a Passenger Car

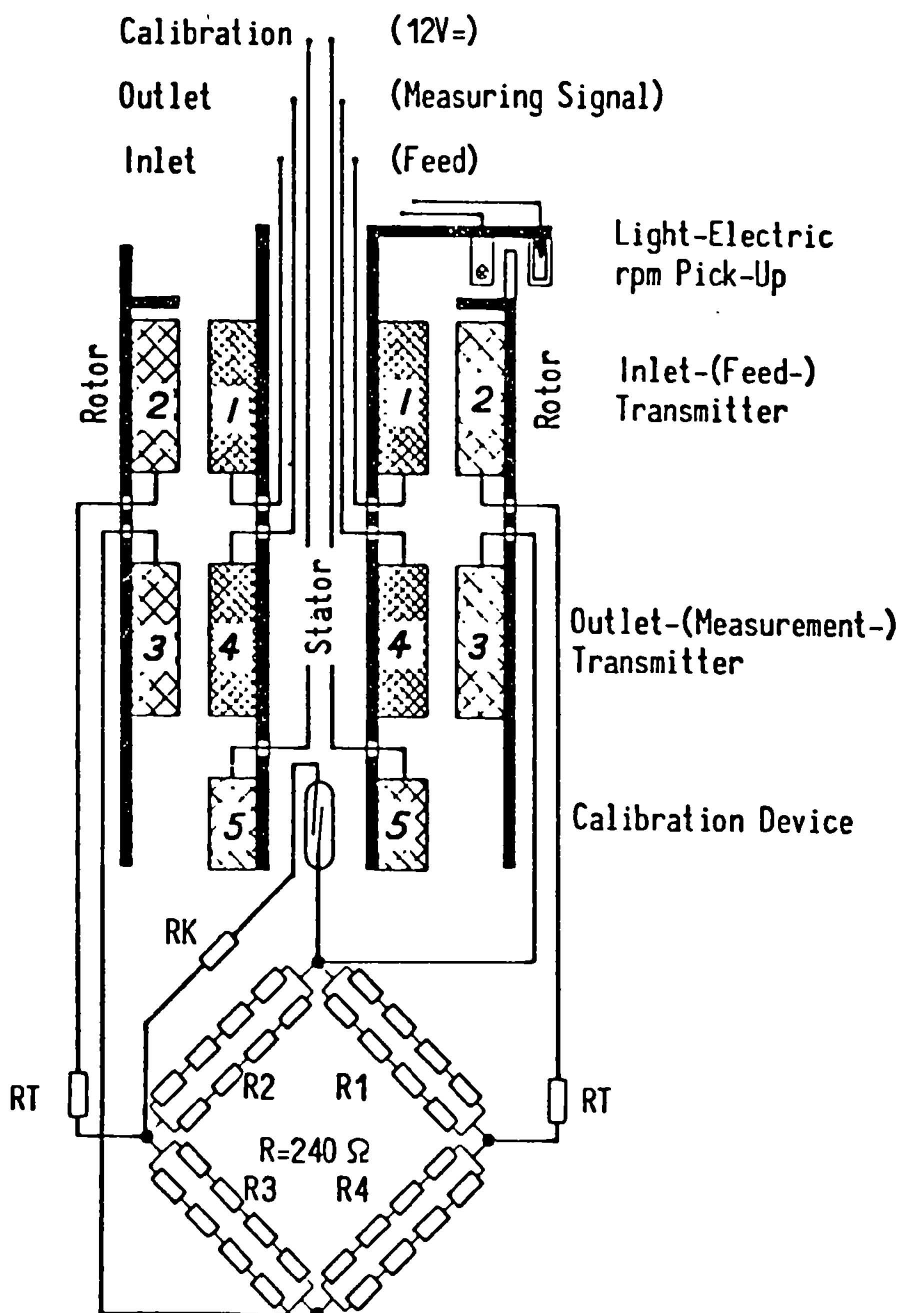
V1MA 0879

02

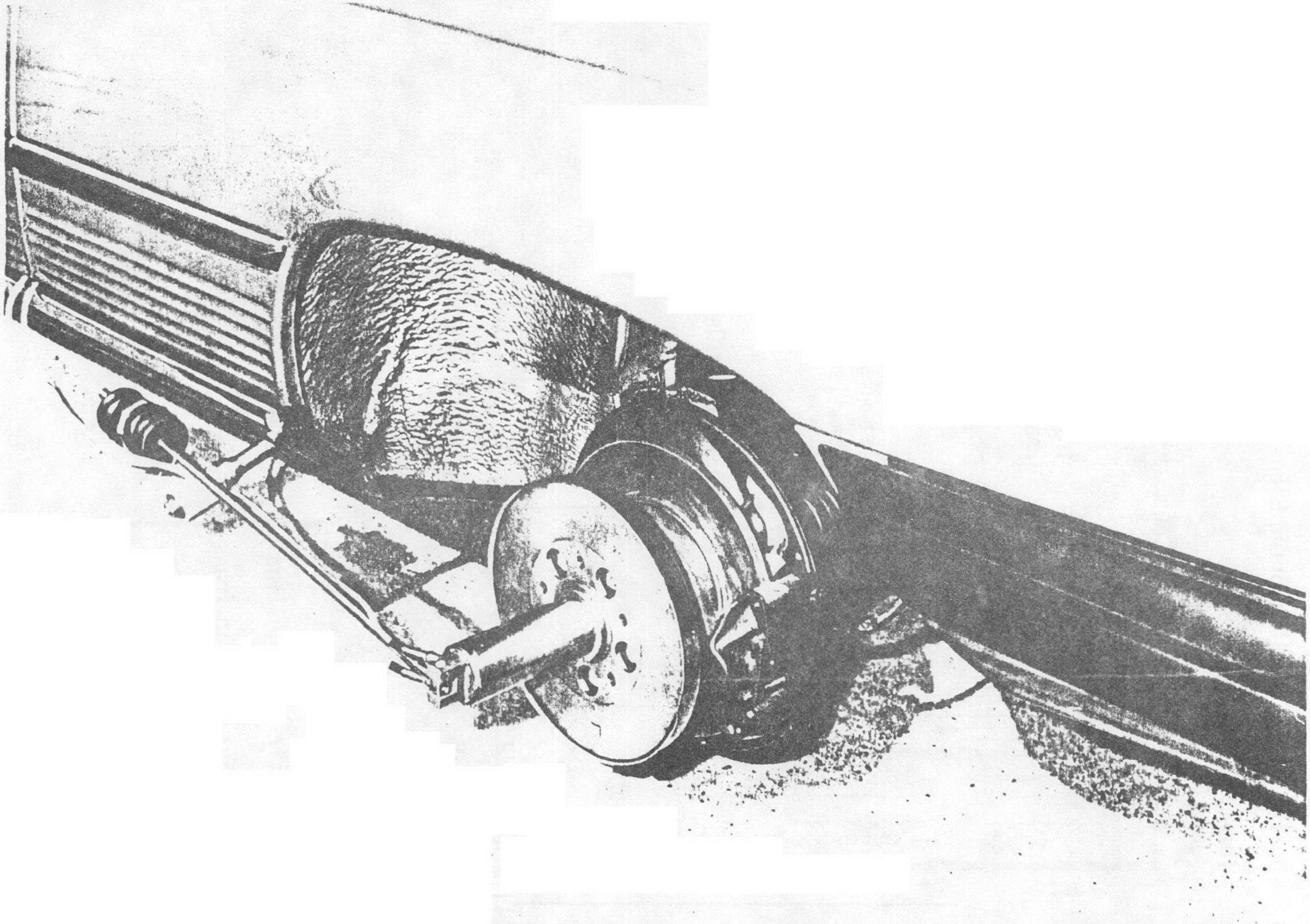


Cross Section of
Torque Disc and
Transmitter

V 1MA 0879
03



Schematic Outlay of the Inductive Rotation Transmitter with rpm Pick-Up, Electrical Calibration Device and Strain Gauge Bridge	V1MA 0879 04	



Detail of Torque Disc Installation on Vehicle

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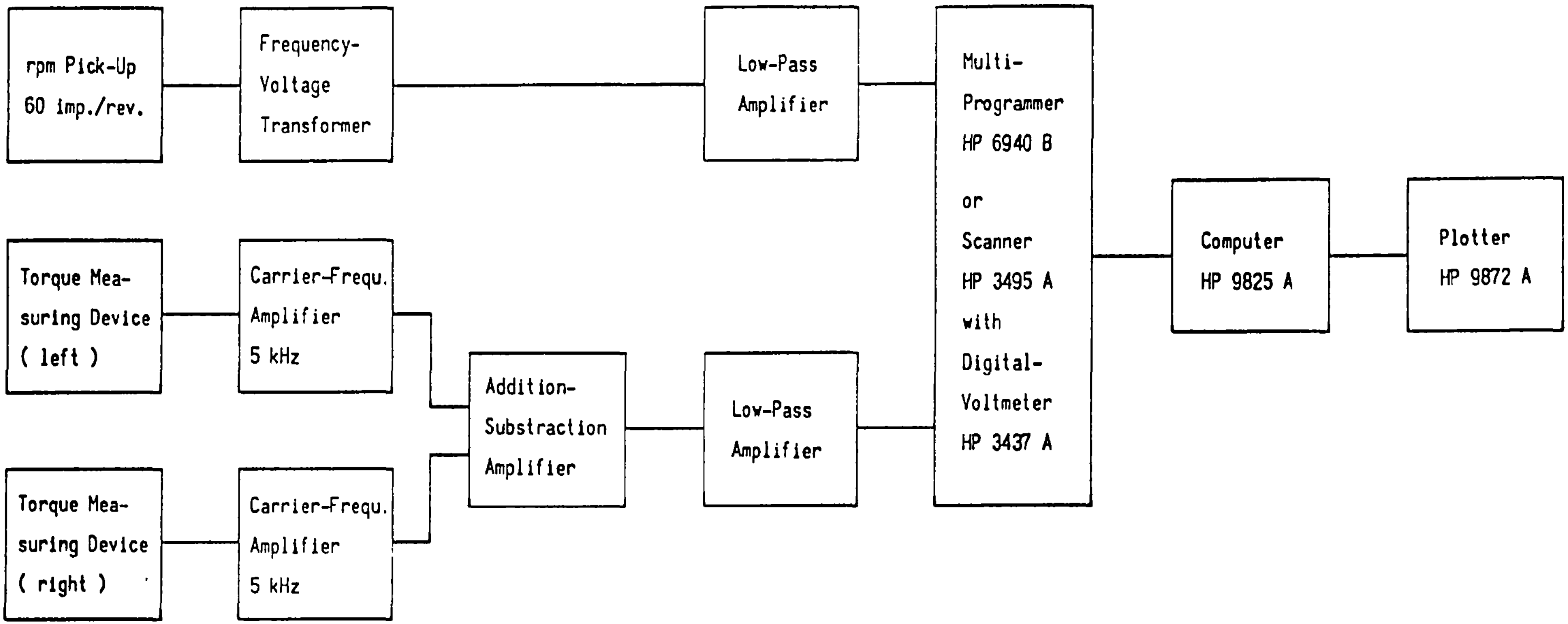
05

View of Vehicle with Torque Disc

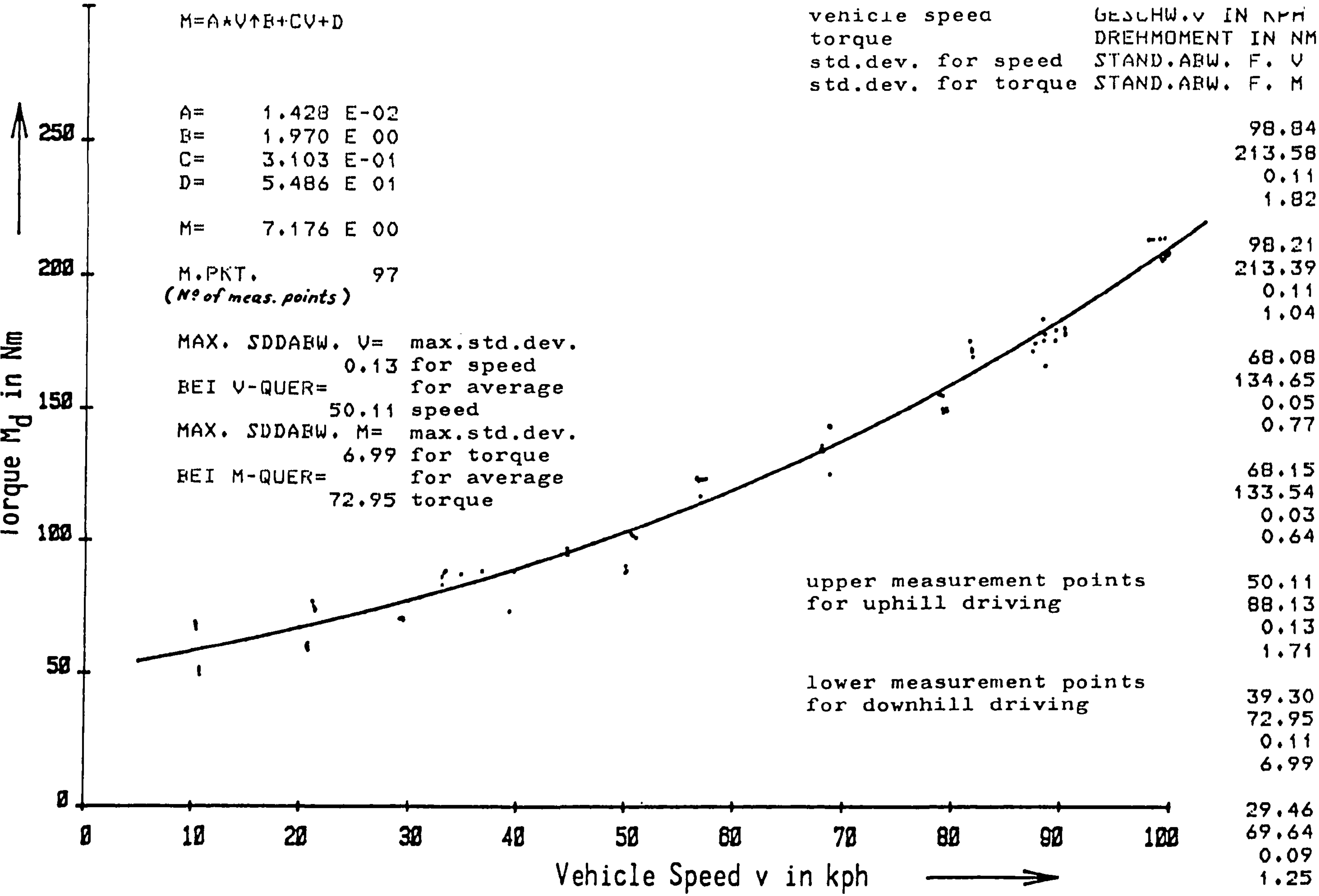
V1MA 0879

06



Block Diagram of Measuring-,
Instrumentation- and Evaluation System

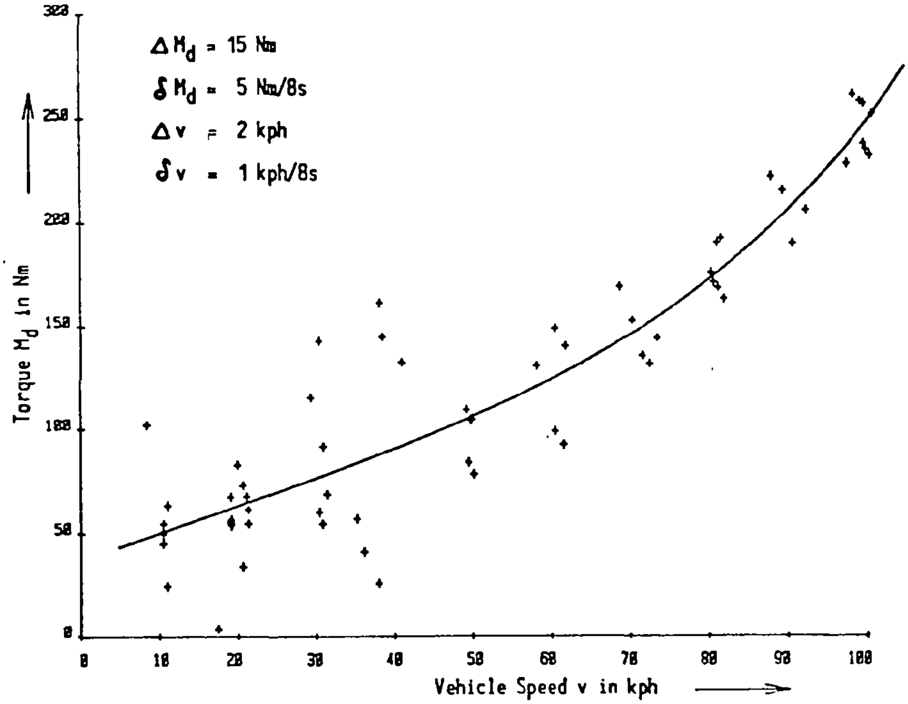
V1MA 0879
07

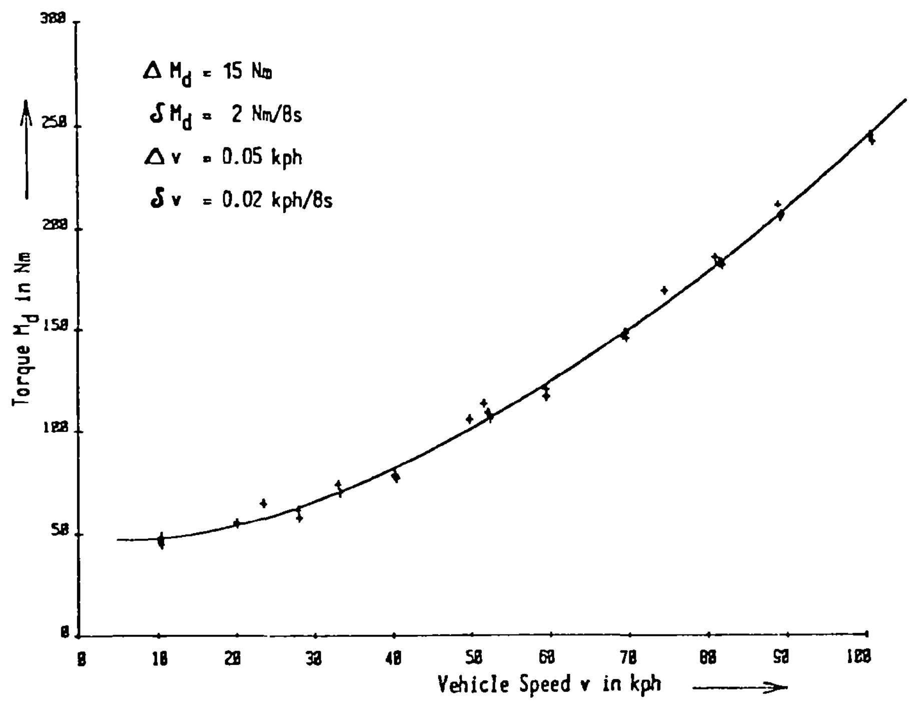


Evaluation Example of Road Load Determination on Test Track (Average of Uphill and Downhill Driving)

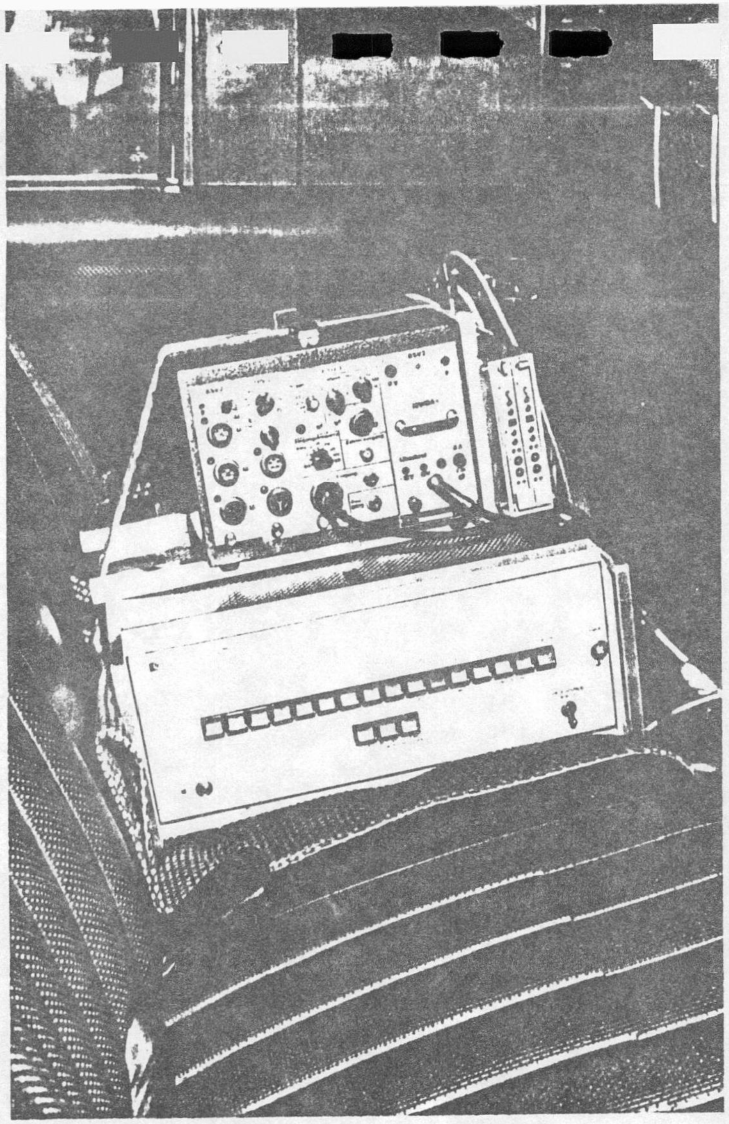
V1MA 0879

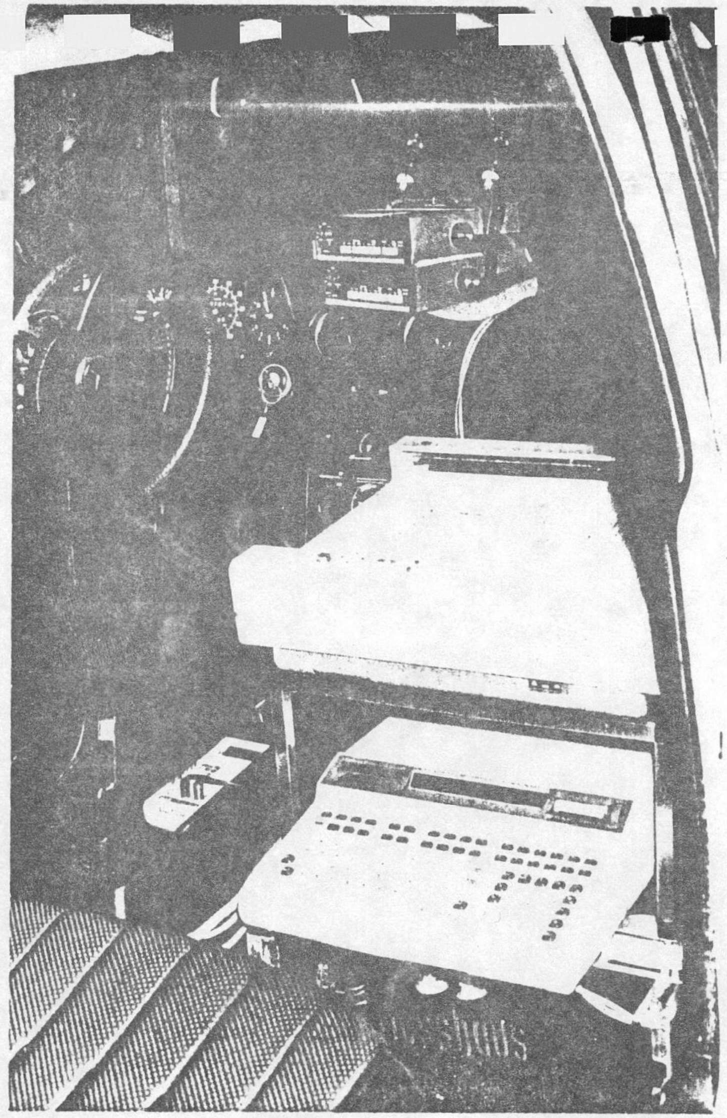
08





	Influence of Speed Drift at "Constant Speed"-Driving on Spread of Road Load Measuring Points	V1MA 0879 09





View of Electronic Measurement and Evaluation System for Road Load Determination and Chassis Dynamometer Adjustment

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10

ATTACHMENT VIII

Technical Report

A Track to Twin Roll Dynamometer Comparison of Several Different Methods of Vehicle Velocity Simulation

by

John Yurko

June 1979

NOTICE

Standards Development and Support Branch
Emission Control Technology Division
Office of Mobile Source Air Pollution Control
Office of Air, Noise and Radiation
U.S. Environmental Protection Agency

ABSTRACT

The current EPA test procedure for fuel economy and emissions testing uses a twin roll dynamometer, obtaining a speed signal from the rear roll and simulating the forces at the front roll. With the rolls coupled only by the drive wheels of the vehicle, the front roll travels approximately 2% slower than the rear roll at steady-state 50 mph, resulting in approximately a 4% overprediction of fuel economy. Coupling the rolls externally equalizes the roll speeds at a value which better simulates the road velocity and therefore better predicts the fuel economy. This report describes the test program and data analysis which led to these conclusions.

FOREWORD

The EPA has conducted a test program in order to determine the most representative method for simulating the road velocity of a vehicle on a Clayton twin-roll dynamometer. The three methods of simulating the road velocity on the twin-roll dynamometer are:

(1) Using the velocity of the rear roll, which is the current method,

(2) Using the velocity of the front roll,

(3) Operating with the rolls coupled.

To determine which of these three methods most closely represents the road experience of a vehicle, steady-state tests were conducted on a track and compared to dynamometer tests using each speed simulation method. The same vehicle was used for all phases of the test program. This report describes the test program, reports the results, and recommends the most appropriate method of velocity simulation on a twin-roll dynamometer.

SUMMARY

The results of the road to dynamometer comparison show that the road velocity is best simulated when the front and rear rolls of the dynamometer are coupled. With the rolls coupled, the simulated velocity was within 0.025% of actual road velocity. With the rolls uncoupled, the rear roll velocity over credited the vehicle speed by approximately 1.0% while the front roll under credited the speed by about 1.0%. Coupling the rolls reduced measured fuel economy by approximately 4% in comparison with the current method of using the rear roll speed. This is consistent with the 1% speed errors in each roll, since the force is proportional to the velocity squared. In conclusion, coupling the rolls is technically the best method of simulating the vehicle velocity and should improve EPA fuel economy predictions.

I. INTRODUCTION

When a vehicle is tested for fuel economy and emissions on a Clayton twin-roll dynamometer, there is a difference between the velocities of the front and rear rolls of the dynamometer.

Therefore, the speed sensor location can have a significant effect on fuel economy and emissions testing. Steady-state tests have shown that the rear roll travels approximately 1.0 mph faster than the front roll at 50 mph (1). This occurs because the drive wheels of the vehicle, which are cradled between the two rolls, act as the only coupling between the two rolls when a vehicle is driven on the dynamometer. The power absorber and inertia flywheels, which simulate the road force experienced by a vehicle, are connected to the front roll. This causes a greater tangential force at the tire/front-roll interface than at the rear-roll interface, resulting in a smaller effective rolling radius in the tire with respect to the front roll as opposed to the rear roll.

Externally coupling the rolls eliminates the difference in velocities of the two rolls. Therefore, this has been considered as an alternative method for simulating the vehicle speed. Locating the speed sensor on the front roll has also been suggested, since the forces and the velocity would then be associated with the same surface. To determine which method would best simulate the actual road velocity of a vehicle a test program was conducted. The following discussion describes the track tests, the dynamometer tests, and the road to dynamometer comparison which were used to determine the optimum method for measuring the simulated velocity of a vehicle.

II. DISCUSSION

The test program consisted of three portions: 1) track portion 2) dynamometer portion, and 3) data analysis. The track portion was conducted at the Transportation Research Center of Ohio (TRC). The dynamometer portion was conducted at the EPA laboratory in Ann Arbor. One vehicle, a 1978 Mercury Montego, was used for all testing. Steady-state tests were conducted on both the track and the dynamometer, for four different sets of radial tires which are listed in Appendix A-1.

A. Track Portion

Prior to each test, the vehicle was weighed with a full tank of indolene test fuel, complete instrumentation, and two operators. After a 20-minute warm up at 50 mph around an oval track, data were collected during one lap of the track for approximately 10 minutes at steady state 50 mph. Both left and right rear wheel speeds, left and right rear wheel torques and a fifth wheel speed were recorded at a once/second rate. Total fuel flow and distance traveled were also measured. Ambient temperature, barometric pressure, wind velocity and wind direction were monitored during the tests. Tire temperatures were recorded before and after each test. Immediately following the steady state test, 10 coastdowns were conducted in accordance with the EPA recommended practice for determination of road load for light-duty vehicles. A detailed description of all the equipment used is given in Appendix A-2.

B. Dynamometer Portion

The goal was to reproduce the exact road torque and speed conditions for each test on the dynamometer. In order to obtain the necessary precision, we instead chose to use a 9-point speed/torque test matrix, and then to interpolate the dynamometer data to the road datum.

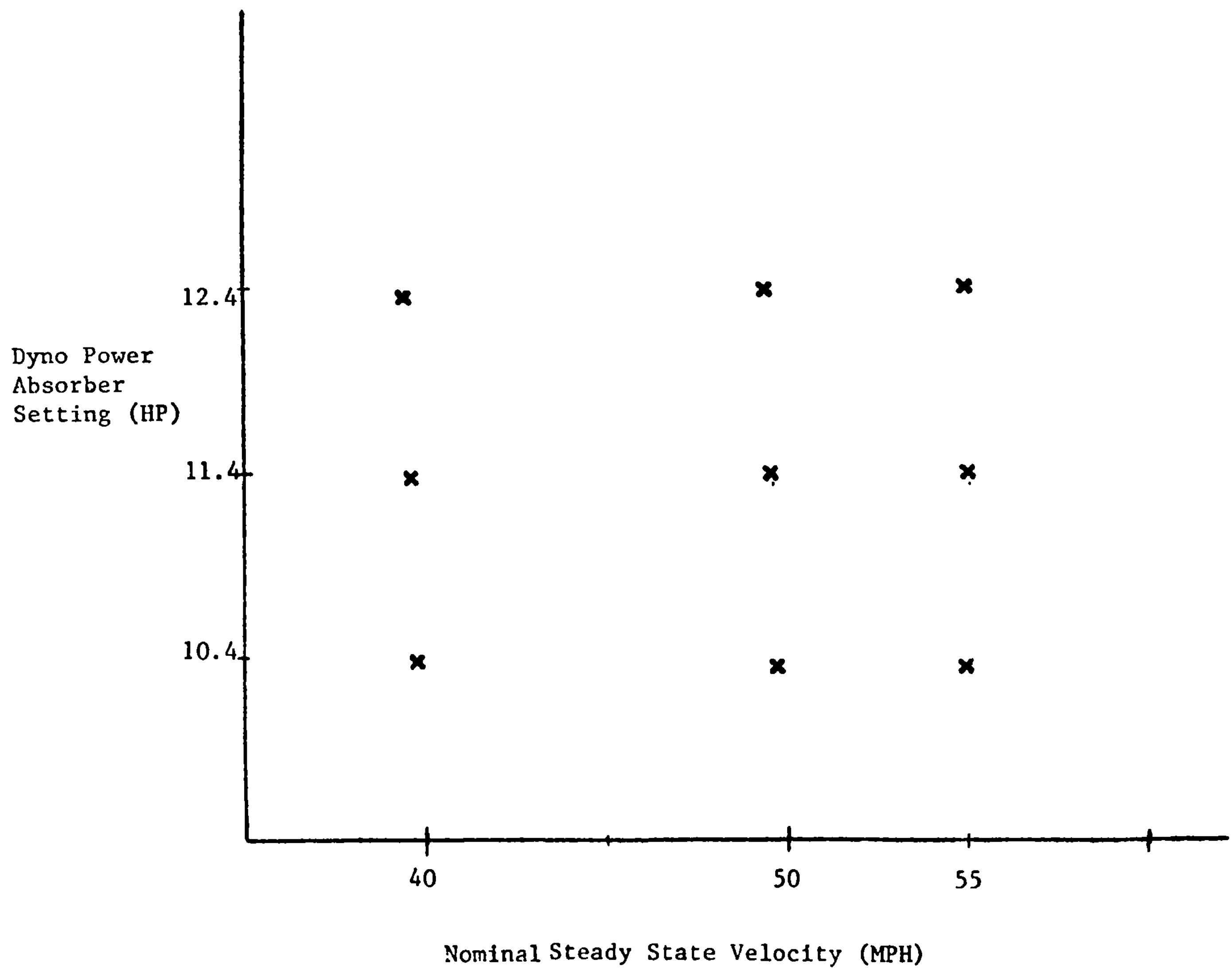
For the dynamometer tests it was decided to warm-up the tires so that they would be at approximately the same conditions as are vehicle tires during typical EPA tests. This was chosen since the results would be more representative of conditions during EPA tests than would result from a 20 minute 50 mph steady-state warm-up and there would be reduced probability of tire failures. This approach also resulted in tire temperatures which were closer to the road tire temperatures than would have occurred with the 20 minute steady-state warm-up.

The test cycle chosen consisted of a tire warm-up of one complete FTP cycle followed by three consecutive 5 minute steady-state measurements at a single horsepower. At this time a 15 minute cool down period was provided before the dynamometer adjustment was changed, then the first 505 seconds (bag one) of the LA4 cycle was driven to precondition the tires and three more steady-state measurements were obtained. This cycle of a cool down followed by a preconditioning was repeated until all data necessary for the 9 point matrix were obtained. The 15 minute cool down followed by the 505 seconds of preconditioning was chosen on the basis of tire temperature measurements, to be appropriate to yield approximately the same tire temperatures as were obtained after one complete LA-4 cycle starting with a cold tire. No tire failures were observed in this program, either as a result of the warm-up cycle or the measurement conditions.

The vehicle was tested with each set of tires at three steady-state speeds, nominally: 1) 50 mph, 2) 40 mph, and 3) 55 mph. For greater precision the actual measured velocities were used in the data analysis. The 55 mph point was chosen instead of 60 mph since, at 60 mph the tire temperature increased rapidly, indicating possible tire failure problems. Data were collected during each steady state for 5 minutes at a once/second rate. As in the track portion, both rear-wheel torques and rear-wheel speeds were recorded. Instead of a fifth wheel speed, the front and rear dynamometer roll speeds were recorded. Fuel flow and rear roll distance traveled were also measured. Each steady state was followed by a vehicle/dynamometer coastdown from 55 mph to 45 mph and the coastdown time was recorded. The dynamometer coastdown times were only used for a fuel economy comparison as described in Section III. The steady-states and the coastdowns were repeated at each speed for three different indicated dynamometer power absorber settings: 1) 11.4 HP, 2) 12.4 HP, 3) 10.4 HP, in that order. This test sequence is summarized in the 9-point test matrix shown in Figure 1. The 11.4 HP value

Figure 1

Dynamometer Test Matrix



approximately represented the road load of the vehicle with the midrange set of tires, as determined by matching the road and dynamometer coast-down times. The 10.4 and 12.4 test values were chosen to cover the range of road loads, observed with different tires. For greater precision, actual wheel torques, and wheel speeds were used to match the dynamometer test to the road. This was done by a linear regression which is described in Section IIC.

The entire configuration was then repeated, for each tire set, with the front and rear rolls coupled by a motorcycle chain and sprockets connected to each roll. A detailed description of the test sequence including warm-up cycles for the dyno portion is given in Appendix B. All the equipment used in the dyno portion was the same as the equipment used in the track portion with the exception of replacement of some minor damaged components and the additional equipment associated with the dynamometer. These are included in the equipment list of Appendix A-2.

C. Data Analysis for Road to Dynamometer Comparisons

For each set of tires, one 50 mph steady-state test was conducted on the road. For each test, mean rear wheel angular speeds, mean rear wheel torques and a mean fifth wheel speed were calculated.

Conceptually, the intent was to reproduce the rear wheel torque and speed conditions of the vehicle which were observed on the road, for each set of tires on the dynamometer. Under these conditions, the different possible speed measurements would be sampled, and that method of measurement which best agreed with the road fifth wheel velocity would be selected as the most appropriate method of measuring the dynamometer simulated speed.

The conceptual approach could not be used directly because of the experimental precision considered necessary to resolve the small velocity variations among the different methods of dynamometer speed simulation. Therefore, we chose to use the 9-point steady-state speed/torque test matrix described in Figure 1.

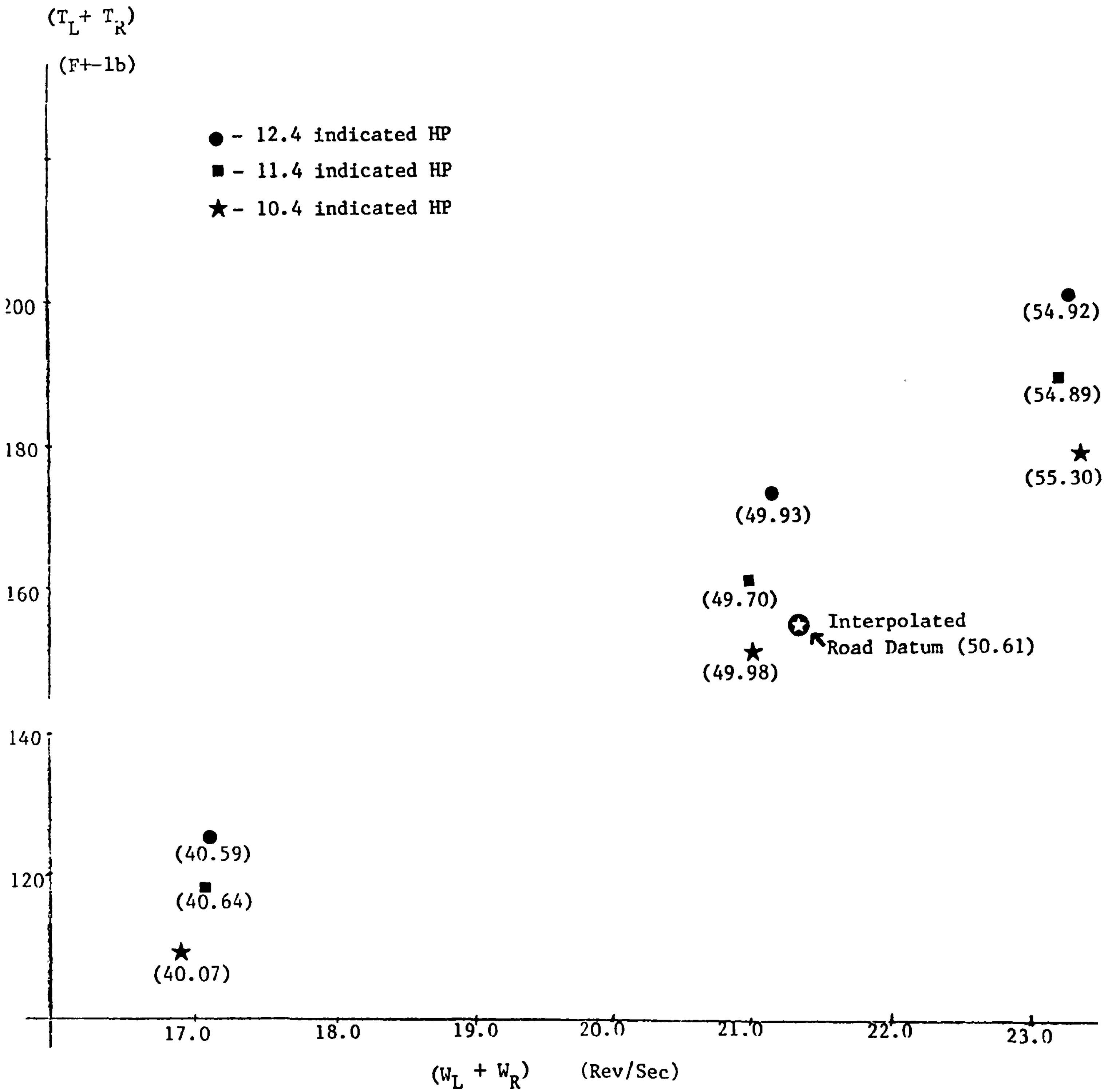
The data obtained at these points uses the interpolated velocity to obtain a roll velocity corresponding to the conditions observed during the road tests. The interpolation was conducted by means of a multiple linear regression using the mean of the data at each point of the test matrix.

First, as discussed, the mean values of each rear wheel angular speed, each rear wheel torque, and each dynamometer roll velocity, with rolls coupled and uncoupled was calculated for every steady-state test. An example of these data for one of the nine point matrices is graphically shown in Figure 2. The interpolation of these data to the observed road point was accomplished by regressing each roll velocity versus the sum of the mean rear wheel angular speeds and the sum of the mean rear wheel torques, over each 9-point test matrix, yielding the coefficients fo the following equations:

Figure 2

Tire 4 With Rolls Coupled

Sum of the Rear Wheel Torques vs. Sum of the Rear Wheel Angular Speeds



*The values in parenthesis under each point are the rear roll velocities in (MPH)

$$\bar{V}_{RR} = a_R(\bar{W}_L + \bar{W}_R) + b_R(\bar{T}_L + \bar{T}_R) + C_R \tag{1}$$

$$\bar{V}_{FR} = a_F(\bar{W}_L + \bar{W}_R) + b_F(\bar{T}_L + \bar{T}_R) + C_F \tag{2}$$

$$\bar{V}_{coup} = a_C(\bar{W}_L + \bar{W}_R) + b_C(\bar{T}_L + \bar{T}_R) + C_C \tag{3}$$

Where:

$\bar{V}_{RR}$ = mean rear roll velocity

$\bar{V}_{FR}$ = mean front roll velocity

$\bar{V}_{coup}$ = mean rear roll velocity with rolls coupled

$\bar{W}_L$ = mean left wheel angular speed

$\bar{W}_R$ = mean right wheel angular speed

$\bar{T}_L$ = mean left wheel torque

$\bar{T}_R$ = mean right wheel torque

$a^{'s}$, $b^{'s}$, $c^{'s}$ = unique sets of regression coefficients for each roll condition and each 9-point test matrix

The road values of mean wheel torques and speeds were inserted into equations (1), (2), and (3) for each set of tires to obtain the simulated road velocity for each method of speed measurement interpolated to the road conditions. The predicted road velocities as given by the above equations, were then compared to the actual mean road velocity for the same set of tires:

$$V_{RR/Road} = a_R(\bar{W}_L + \bar{W}_R)_{Road} + b_R(\bar{T}_L + \bar{T}_R)_{Road} + C_R$$

$$V_{FR/Road} = a_F(\bar{W}_L + \bar{W}_R)_{Road} + b_F(\bar{T}_L + \bar{T}_R)_{Road} + C_F$$

$$V_{coup/Road} = a_C(\bar{W}_L + \bar{W}_R)_{Road} + b_C(\bar{T}_C + \bar{T}_R)_{Road} + C_C$$

Where:

$V_{RR/Road}$ = Road velocity as simulated by the rear roll at the road conditions

$V_{FR/Road}$ = Road velocity as simulated by the front roll at the road conditions

$V_{coup/Road}$ = Road velocity as simulated with the rolls coupled

a,b,c = the set of coefficients obtained from the regressions of the dynamometer data for each tire (different for the rear roll, front roll, and coupled roll predictions)

Sample calculations and the original data, including the regression coefficients are given in appendix C.

III. RESULTS

The results of all tests on the radial and bias belted tires are given in Table 1.

The mean deviation from the actual road velocity for the radial tires was +1.10% using the rear roll velocity simulation, -1.07% using the front roll, and -0.22% with the rolls coupled. Where, a positive deviation corresponds to an observed dynamometer velocity greater than the road velocity under the same wheel condition.

For the bias-belted tires, the rear roll deviated by +1.23% from the road, the front roll deviated by -0.04%, and the coupled rolls deviated by +0.40%.

Overall, the rear roll was in error by +1.15%, the front roll by -0.71%, while the error with the rolls coupled was only -0.02%. Therefore, on the average and particularly for radial tires the coupled mode most closely simulated the road.

Since coupling the rolls improved the vehicle velocity simulation, the vehicle fuel economy effect of this change was investigated. In the majority of EPA fuel economy tests, alternate dynamometer adjustments, obtained by the coastdown technique, are used. Also the coastdown method is used in dynamometer calibration, and therefore, would account for the increased friction of the coupling mechanism. Consequently, a comparison of vehicle fuel economy, obtained with dynamometer adjustments which produced equal coastdown times was considered the most appropriate approach to evaluate the fuel economy effect of coupling the dynamometer rolls. This comparison could easily be made since during the dynamometer portion of this test program vehicle dynamometer coastdown times were recorded immediately following the fuel consumption tests.

Figure 3 shows the 50 mi/hr fuel consumption of the vehicle equipped with radial tires plotted versus the coastdown time obtained for both the uncoupled and coupled tests. This plot indicates that coupling the dynamometer rolls results in a 2 to 6 percent increase in measured fuel consumption for the same vehicle-dynamometer coastdown time. For example, at a coastdown time of 14.0 sec, the fuel consumption was approximately 7150 cc/km with the rolls uncoupled and about 7450 cc/km with the rolls coupled, a difference of approximately 4%.

The fuel economy results obtained in this test program are all from steady-state measurements. However, the results are consistent are preliminary investigations of the effect on transient cycles. For example, computer modeling has estimated the transient cycle fuel economy effect to be 4%.(2) Limited empirical data from transient cycle tests also indicate the effect to be about 4%.(3)

IV. CONCLUSIONS

Operating with the rolls coupled most closely simulates the road experience of a vehicle using radial tires, and therefore, provides the most accurate method of testing for fuel economy. The current EPA method for simulating the vehicle velocity, using the rear roll speed, causes an over prediction of steady-state 50 mph fuel economy by approximately 4%. This occurs because the velocity error results in both an underloading of the energy demand from the vehicle and an overcredit of the distance travelled.

The same mechanism occurs during transient cycles and in this instance, inertial forces applied to the vehicle are also inappropriately low because of the velocity error. Computer modeling and limited empirical data indicate the transient cycle fuel economy errors resulting from this velocity error are also about 4%. It should be noted that these conclusions are based on data from vehicles equipped with radial tires, however this is the most important case. It is estimated that over 70% of the vehicles tested at EPA are equipped with radial tires.

Table 1

Radial Tires

Tire No.	Road Velocity Predicted by the Front Roll (mph)	Road Velocity Predicted by the Rear Roll (mph)	Road Velocity Predicted with Rolls Coupled (mph)	Observed Road Velocity (mph)
1	50.05	51.19	50.45	50.81
2	49.72	50.58	50.00	50.10
3	50.18	51.43	50.85	50.83
4	50.26	51.38	50.61	50.62
Mean	50.05	51.15	50.48	50.59
% Deviation	-1.07	+1.10	-0.22	-

$\left(\frac{\text{Predicted - Observed}}{\text{Observed}}\right) \times 100$

Bias Belted Tires

Tire No.	Road Velocity Predicted by the Front Roll (mph)	Road Velocity Predicted by the Rear Roll (mph)	Road Velocity Predicted with Rolls Coupled (mph)	Observed Road Velocity (mph)
6	50.53	51.16	50.82	50.51
7	50.55	51.20	50.70	50.60
Mean	50.54	51.18	50.76	50.56
% Deviation	-0.04	+1.23	+0.40	-

TOTALS

	Road Velocity Predicted by the Front Roll (mph)	Road Velocity Predicted by the Rear Roll (mph)	Road Velocity Predicted with Rolls Coupled (mph)	Observed Road Velocity (mph)
Mean	50.22	51.16	50.57	50.58
% Deviation	-0.71	+1.15	-0.02	-

Error analysis indicated that on the average, we were 95% confident that the predicted values were accurate to within $\pm$0.23 mph.

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1. Richard Burgeson, Myriam Torres, "Tire Slip on the Clayton Dynamometer", EPA Technical Support Report, LDTP 78-02, March 1978.

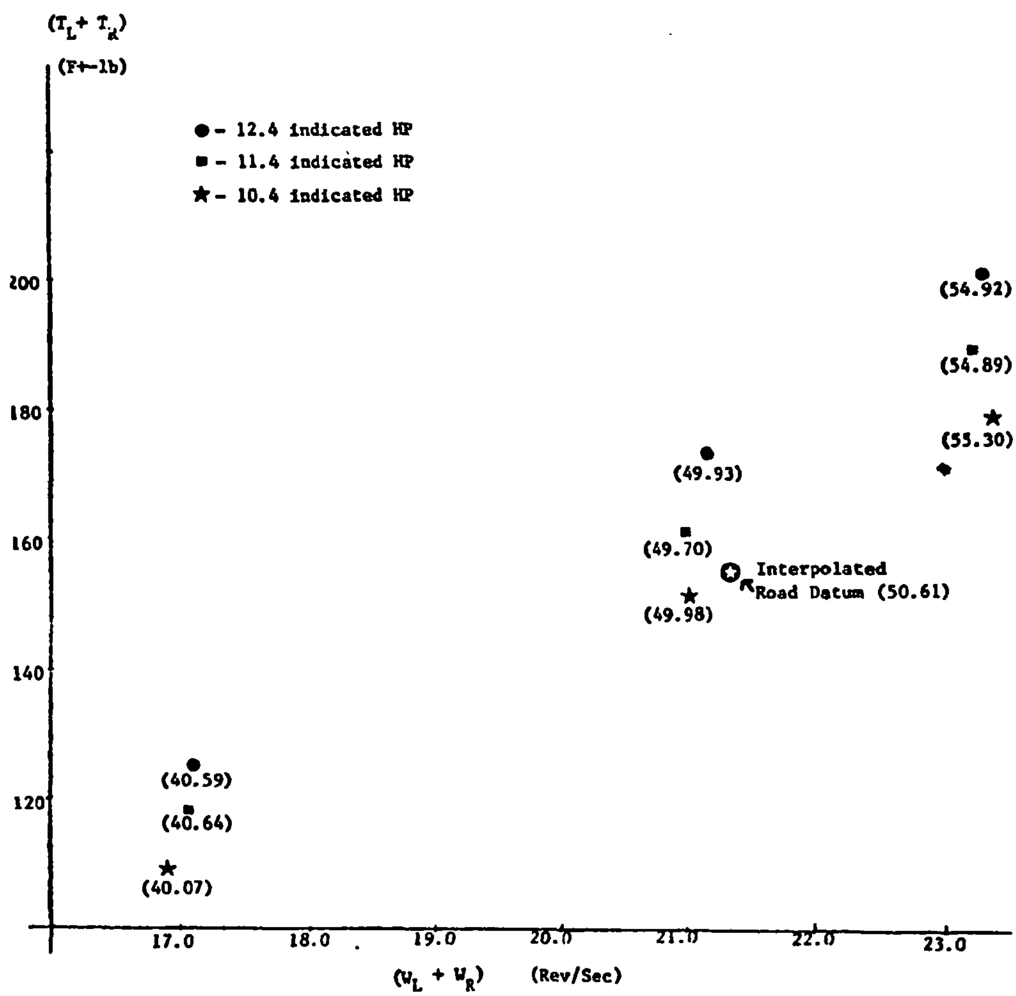
2. John Yurko, "Computer Simulation of Tire Slip on a Clayton Twin Roll Dynamometer", EPA Technical Support Report, SDSB 79-10, February 1979.

3. Conversation with Don Paulsell, of the EPA Ann Arbor Laboratory, March 1979.

Figure 2

Tire 4 With Rolls Coupled

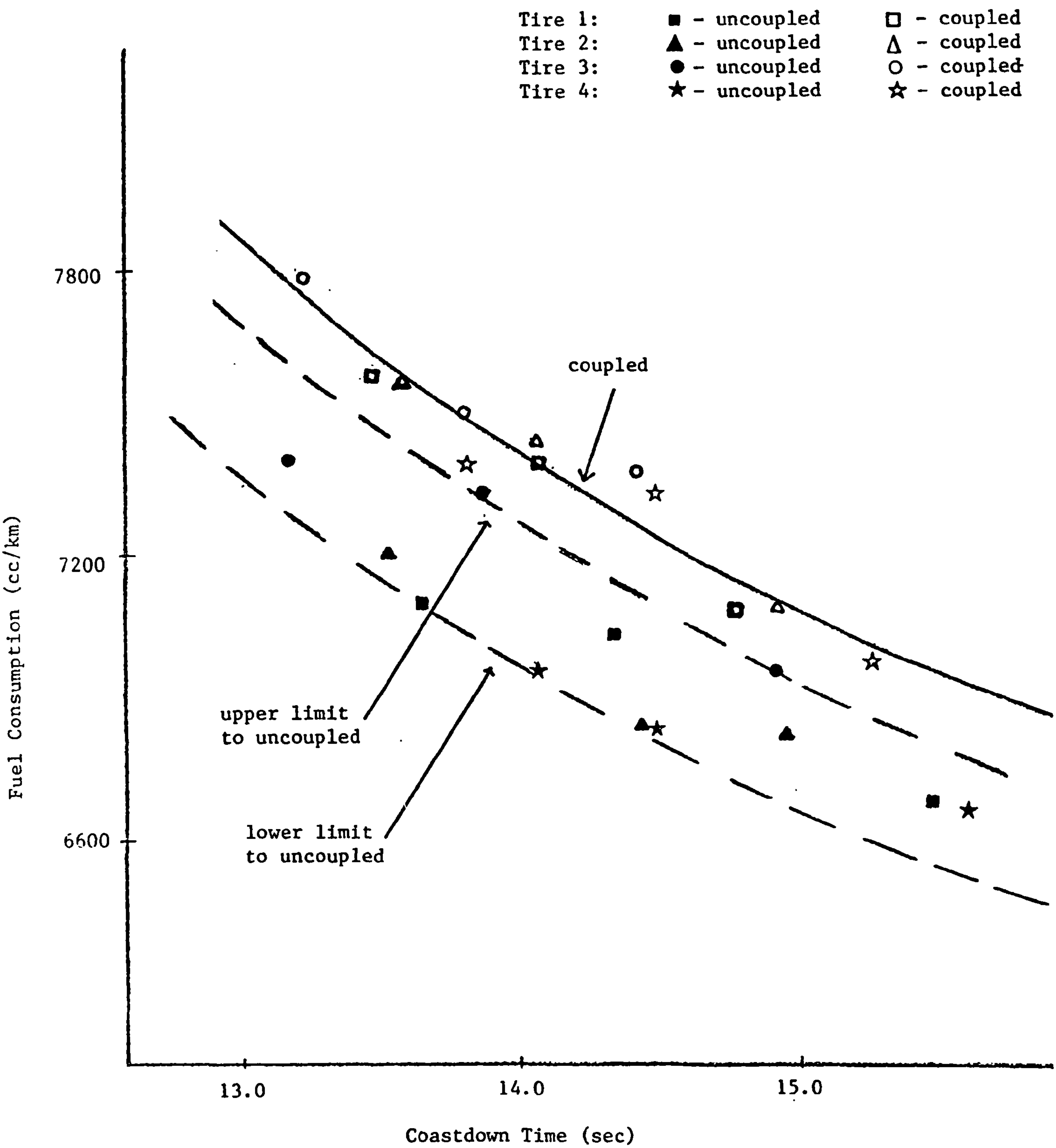
Sum of the Rear Wheel Torques vs. Sum of the Rear Wheel Angular Speeds



*The values in parenthesis under each point are the rear roll velocities in (MPH)

Figure 3

Coupled vs Uncoupled Fuel Consumption
Plotted Against Coastdown Time
(with radial tires)



Appendix A-1

Tire No.	Tire	Tire Type	Tire Size
1	Michelin-X	Radial	GR78x15
2	Firestone 721	Radial	GR78x15
3	Firestone 721	Radial	GR78x14
4	Multimile Supreme	Radial	GR78x15
6	Uniroyal Fastrak	Bias Belted	G78x15
7	Uniroyal Fastrak	Bias Belted	G78x15

Vehicle

1976 Mercury Montego w/ 29,000 accum. miles

Appendix B

OVERVIEW OF TEST SEQUENCE
(eg. using tire no. 3)

1. Tire no. 3 mounted, pressure set to 45 PSI.

2. Tires broken in with 1 FTP.

3. Allowed to cool at least 4 hours.

4. Reset pressure to 45 PSI.

5. Vehicle rear axle weight approximately 2290 lb with driver and full gas tank.

6. Set dynamometer inertia to 5000 lbs.

7. Set dynamometer horsepower to 11.4 horsepower.

8. Set fixed data to 350114.

9. Conduct 1 FTP, then obtain tire temperatures.

10. Insert tape in techtran, ready for scan at 1 second intervals (Tape labeled: uncoupled 350114, 3501124).

11. Conduct a 5-minute Steady State at 50 mph, collect data.

12. Conduct a coastdown, collect 55 to 45 mph time only.

13. Record tire temperature during or right after coastdown, reset fixed data to 340114.

14. Conduct a 5-minute Steady State at 40 mph, collect data.

15. Conduct a coastdown. (NOTE: be sure to collect data only during the 5-minute Steady State. All data collection devices should be reset before new Steady State speed is set.) Collect coastdown time and tire temperatures.

16 Reset fixed data to 355114, conduct a Steady State at 55 mph for 5 minutes collecting data. Stop data, conduct a coastdown, record time and tire temperatures. Increase speed above 60 mph.

17. Life vehicle, conduct a dynamometer only coastdown, check zero, adjust on torque meter. Record 55 to 45 mph time. Reset horsepower to 12.4, fixed data to 350124. Tires should be allowed to cool 15 minutes starting from when the vehicle was lifted.

Appendix A-2

Type of Data Being Collected	Equipment
Drive wheel torques (analog voltage output)	Lebow torque sensor Model No. - 7510
Wheel angular velocities (frequency output)	Disc/Rotaswitch pulse Encoders
Conversion of frequency to analog voltage	Anadex frequency to voltage converter
Collect and digitize analog signals for output to a recording device	Fluke datalogger Model 2240B
Record data	Techtran Data Cassette Model 8400
Record fuel flow	Fluidyne Flowmeter Model 1250T
Tire temperatures	Wahl Heat Spy Infared thermometer

18. Conduct a 505 second warm up, record tire temperature.

19. Repeat 11 and 12.

20. Reset fixed data to 340124.

21. Repeat 14 and 15.

22. Reset fixed data to 355124.

23. Repeat 16.

24. Repeat 17. Reset horsepower to 10.4 after dynamometer coast-down, fixed data to 350104, allow 15 minutes cooling, rewind tape and insert new one. (Tape labeled: Uncoupled, 350104).

25. Conduct a 505 second warm up, repeat 11 and 12.

26. Reset fixed data to 340104.

27. Repeat 14 and 15.

28. Reset fixed data to 355104.

29. Repeat 16.

30. Conduct dynamometer only coastdown, recheck zero drift. Rewind tape.

31. Steps 1 through 30 complete a tire for the uncoupled configurations. Approximately 3 to 4 hours of testing and 2 cassette tapes are required. If nothing is done to the vehicle but to let it set for an hour (say for lunch), you should be able to start at step 6 with rolls coupled and fuel tank filled, and conduct steps 6 through 30 to complete a tire type.

    Steps 1 through 31 will be repeated for each tire set.

Appendix C-1

Tire 3
9-Point Test Matrix Data on Dynamometer
with Rolls Uncoupled

Test	$\overline{RRV}$ (mph)	$\overline{FRV}$ (mph)	$(\bar{W}_L+\bar{W}_R)$ (rev/sec)	$(\bar{T}_L+\bar{T}_R)$ (ft.-lbs.)
350114	50.116	49.170	21.598	155.892
340114	39.979	39.423	17.239	120.743
355114	55.125	53.932	23.672	181.191
350124	49.955	48.556	21.402	167.662
340124	40.091	39.183	17.198	125.286
355124	55.074	53.421	23.508	187.087
350104	50.068	48.857	21.415	149.114
340104	40.166	39.375	17.181	112.087
355104	54.881	53.459	23.426	164.744

With Rolls Coupled

351114	50.031	50.075	21.710	178.241
341114	40.109	40.130	17.340	138.261
356114	55.030	55.084	23.941	206.123
351124	49.870	49.902	21.618	177.924
341124	39.936	39.948	17.306	131.293
356124	55.011	55.045	23.894	202.540
351104	49.990	50.041	21.678	159.231
341104	40.029	40.040	17.312	118.504
356104	55.031	55.079	23.821	183.033

Regression Coefficients
$(Vi = a\ (W_L + W_R) + b\ (T_L + T_R) + C)$

Vi	a	$b \times 10^2$	C	R-SQR
Rear Roll	2.3808	-0.26600	-0.58549	.99934
Front Roll	2.3639	-1.2120	-0.078181	.99991
Coupled Roll	2.3320	-0.46453	+0.23591	.99991

$(W_L + W_R)$ road = 22.03 rev/sec and $(T_R + T_L)$ road = 162.98 ft.-lb

(R-SQR signifies the confidence in the fit of the regression. For example, R-SQR = .99991 means a 99.991% confidence in the fit.)

Appendix C-2

Example Calculation Using Tire 3

Rear Roll Velocity with the Rolls Coupled

$$V_{coup} = a\ (\bar{W}_L + \bar{W}_R) + b\ (\bar{T}_L + \bar{T}_R) + C$$

from linear regression with $\overline{RRV}$ from appendix C-1 as the dependent variable: :

$$a = 2.3320,\ b = -0.46453 \times 10^{-2},\ c=0.23591$$

therefore, applying the coefficients to the road data results:

$$V_{coup/road} = 2.3320\ (\bar{W}_L + \bar{W}_R)\ \text{road} + -0.46453 \times 10^{-2}\ (\bar{T}_L + \bar{T}_R)\ \text{road} + 0.23591$$

where:

$(\bar{W}_L + \bar{W}_R)$ road = 22.03, and $(\bar{T}_R + \bar{T}_L)$ road = 162.98

therefore:

$$V_{coup/road} = 50.85$$

this compares to the actual road velocity:

$$V_{road} = 50.83$$

(These correspond to the results given in Table 1. Section III of this report.)

Appendix C-3

Tire 1
9-Point Test Matrix on Dynamometer
with Rolls Uncoupled

Test	$\bar{RRV}$ (mph)	$\bar{FRV}$ (mph)	$(\bar{W}_L + \bar{W}_R)$ (rev / sec)	$(\bar{T}_L + \bar{T}_R)$ (ft. - lbs.)
150114	50.073	49.995	20.902	165.847
140114	40.002	39.340	16.690	127.049
155114	54.950	53.651	22.893	188.137
150124	50.001	48.694	20.800	167.585
140124	40.019	39.163	16.643	122.969
155124	54.982	53.488	22.848	187.133
150104	49.971	48.865	20.782	146.920
140104	39.946	39.218	16.618	111.007
155104	55.021	53.681	22.849	168.222

with Rolls Coupled

Test	$\bar{RRV}$ (mph)	$\bar{FRV}$ (mph)	$(\bar{W}_L + \bar{W}_R)$ (rev / sec)	$(\bar{T}_L + \bar{T}_R)$ (ft. - lbs.)
151114	50.044	50.100	21.164	181.263
141114	39.983	40.006	16.828	150.630
156114	54.937	55.042	23.239	205.630
151124	50.032	50.071	21.139	175.900
141124	40.058	40.066	16.861	128.142
156124	55.088	55.125	23.278	210.694
151104	49.927	49.963	21.068	158.839
141104	40.035	40.042	16.829	114.663
156104	54.945	54.998	23.175	180.070

Regression Coefficients

Vi	a	$b \times 10^2$	c	R-SQR
Rear Roll	2.4586	-0.48606	-0.37559	.99988
Front Roll	2.3336	-0.10248	-0.56480	.99750
Coupled Roll	2.3689	-0.29440	0.50768	.99996

$(\bar{W}_L + \bar{W}_R)$ road = 21.27, and $(\bar{T}_L + \bar{T}_R)$ road = (150.616)

(R-SQR signifies the confidence in fit of the data by the regression)

Appendix C-4

Tire 4
9- Point Test Matrix on Dynamometer
with Rolls Uncoupled

Test	$\overline{RRV}$ (mph)	$\overline{FRV}$ (mph)	$(\bar{W}_L + \bar{W}_R)$ (rev / sec)	$(\bar{T}_L + \bar{T}_R)$ (ft. - lbs)
450114	49.960	48.951	20.864	152.446
440114	39.968	39.374	16.623	115.488
455114	55.013	53.765	22.849	176.968
450124	49.990	48.780	20.786	156.114
440124	40.007	39.238	16.591	114.620
455124	55.120	53.680	22.849	178.410
450104	49.956	48.907	20.754	142.134
440104	39.952	39.287	16.575	107.053
455104	55.054	53.873	22.856	165.278

with Rolls Coupled

451114	49.697	49.847	20.999	160.798
441114	40.636	40.733	17.021	119.410
456114	54.887	55.113	23.223	189.825
451124	49.929	50.084	21.154	173.993
441124	40.589	40.688	17.048	125.797
456124	54.918	55.084	23.268	200.720
451104	49.983	50.126	21.086	151.774
441104	40.074	40.175	16.810	109.630
456104	55.295	55.484	23.359	175.490

Regression Coefficients

Vi	a	$b \times 10^2$	c	R-SQR
Rear roll	2.3939	0.12847	0.071297	.99973
Front roll	2.3838	-0.78016	0.57721	.99977
Coupled roll	2.3781	-0.64562	0.84438	.99992

$(\bar{W}_L + \bar{W}_R)$ road = 21.35 and $(\bar{T}_L + \bar{T}_R)$ road = 155.603

Appendix C-5

Tire 2
9- Point Test Matrix Data on Dynamometer
with Rolls Uncoupled

Test	$\overline{RRV}$ (mph)	$\overline{FRV}$ (mph)	$(\bar{W}_L + \bar{W}_R)$ (rev / sec)	$(\bar{T}_L + \bar{T}_R)$ (ft. - lbs)
250114	49.880	48.803	20.916	159.274
240114	39.949	39.251	16.773	123.905
255114	54.976	53.764	22.994	182.863
250124	49.921	49.108	20.980	165.863
240124	40.014	39.561	16.863	123.553
255124	55.061	54.018	23.037	189.958
250104	49.945	49.237	21.036	147.109
240104	39.887	39.492	16.820	108.334
255104	55.121	54.222	23.209	165.357

with Rolls Coupled

251114	50.197	50.340	21.404	174.465
241114	40.167	40.251	17.071	130.972
256114	55.121	55.290	23.511	198.727
251124	50.009	50.132	21.347	172.235
241124	39.980	40.056	16.993	124.866
256124	54.958	55.139	23.455	197.955
251104	49.925	50.061	21.243	161.941
241104	40.111	40.194	17.025	123.351
256104	55.061	55.218	23.437	186.192

Regression Coefficients

Vi	a	$b \times 10^2$	c	R-SQR
Rear roll	2.3000	1.1332	-0.085243	.99987
Front roll	2.3081	0.17002	0.40039	.99984
Coupled roll	2.4186	-0.91726	0.045152	.99996

Appendix C-6

Tire 7
9-Point Test Matrix Data on Dynamometer
with Rolls Uncoupled

Test	$\bar{\bar{RRV}}$ (mph)	$\bar{\bar{FRV}}$ (mph)	$(\bar{W}_L + \bar{W}_R)$ (rev / sec)	$(\bar{T}_L + \bar{T}_R)$ (ft. - lbs)
750114	49.996	49.435	21.102	154.821
740114	39.878	39.562	16.891	114.676
755114	55.081	54.341	23.189	178.088
750124	50.054	49.396	21.114	156.746
740124	39.947	39.576	16.915	112.423
755124	54.917	54.065	23.090	177.521
750104	50.108	49.549	21.136	142.726
740104	40.084	39.773	16.994	106.785
755104	55.158	54.473	23.312	165.109

with Rolls Coupled

Test	$\bar{\bar{RRV}}$ (mph)	$\bar{\bar{FRV}}$ (mph)	$(\bar{W}_L + \bar{W}_R)$ (rev / sec)	$(\bar{T}_L + \bar{T}_R)$ (ft. - lbs)
751114	50.123	50.281	21.368	166.150
741114	39.894	39.994	17.040	121.278
756114	55.142	55.391	23.513	191.962
751124	50.028	50.168	21.333	167.352
741124	40.034	40.117	17.101	118.877
756124	54.957	55.104	23.415	191.912
751104	49.947	50.100	21.289	157.123
741104	40.041	40.131	17.102	113.344
756104	54.966	55.129	23.372	177.934

Regression Coefficients

Vi	a	$b \times 10^2$	c	R-SQR
Rear roll	2.3246	0.84300	-0.32574	.99992
Front roll	2.3209	0.20201	0.12043	.99994
Coupled roll	2.4311	-0.59811	-0.83437	.99998

$(\bar{W}_L + \bar{W}_R)$ road = 21.59 and $(\bar{T}_R + \bar{T}_L)$ road = 159.165

Appendix C-7

Tire 6
9-Point Test Matrix Data on Dynamometer
with Rolls Uncoupled

Test	$\overline{\overline{RRV}}$ (mph)	$\overline{\overline{FRV}}$ (mph)	$(\bar{W}_L + \bar{W}_R)$ (rev / sec)	$(\bar{T}_L + \bar{T}_R)$ (ft. - lbs)
650114	50.078	49.485	20.599	160.995
640114	40.077	39.786	16.552	119.909
655114	54.923	54.293	22.551	184.427
650124	50.029	49.440	20.537	161.376
640124	40.036	39.712	16.527	117.758
655124	55.033	54.272	22.574	184.391
650104	50.133	49.597	20.624	153.316
640104	39.972	39.665	16.483	113.553
655104	54.979	54.303	22.519	172.812

with Rolls Coupled

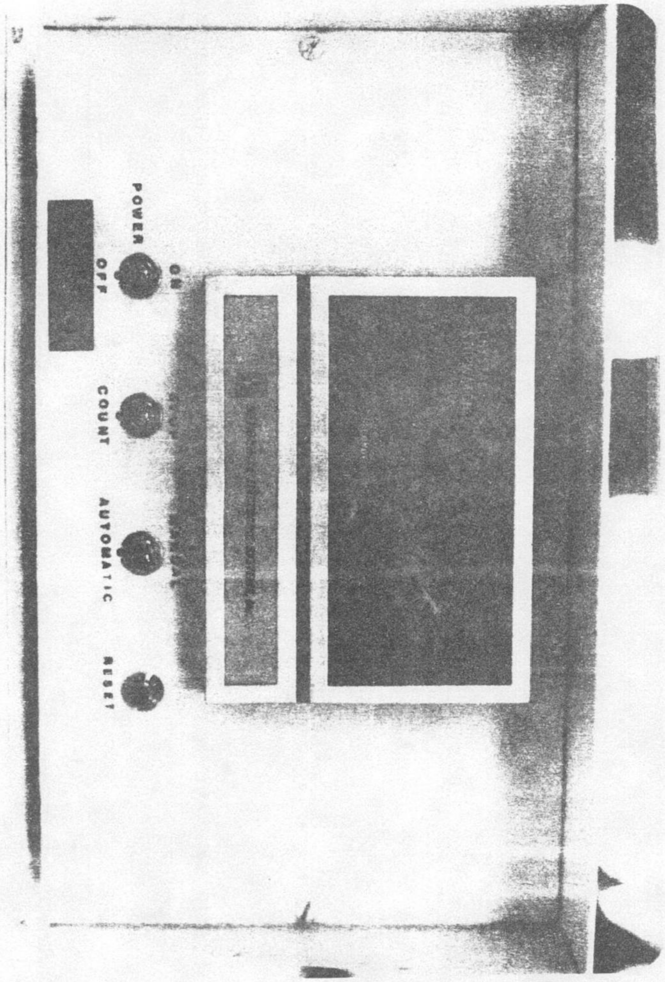
651114	49.979	50.137	20.736	168.248
641114	40.078	40.179	16.623	122.804
656114	55.094	55.271	22.716	194.150
651124	50.004	50.167	20.738	182.684
641124	39.860	39.960	16.526	135.170
656124	55.095	55.268	22.793	210.181
651104	50.082	50.223	20.593	157.842
640104	40.094	40.187	16.606	115.606
656104	54.939	55.215	22.745	180.082

Regression Coefficients

Vi	a	$b \times 10^2$	c	R-SQR
Rear roll	2.5062	-0.24795	-1.0851	.99995
Front roll	2.4694	-0.50356	-0.48771	.99996
Coupled roll	2.4904	-0.46889	-0.70698	.99960

$(W_L + W_R)$ road = 21.0166 and $(T_L + T_R)$ road = 173.9324

COASTDOWN TIMER
USED BY EPA
(ATTACHMENT X)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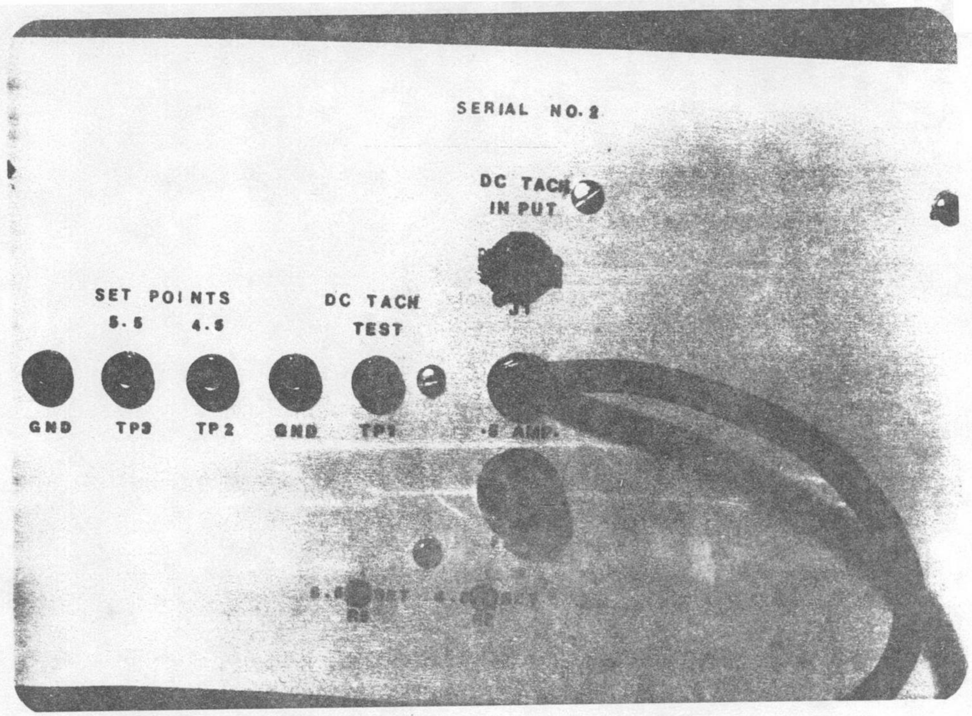
OFF
COUNT
AUTOMATIC
RESET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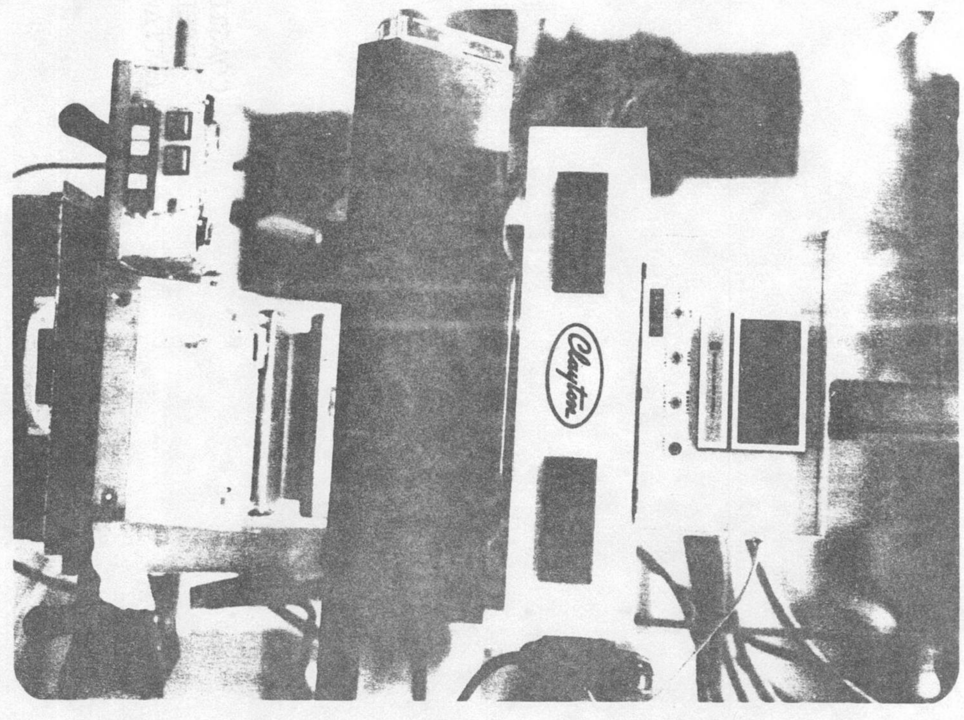
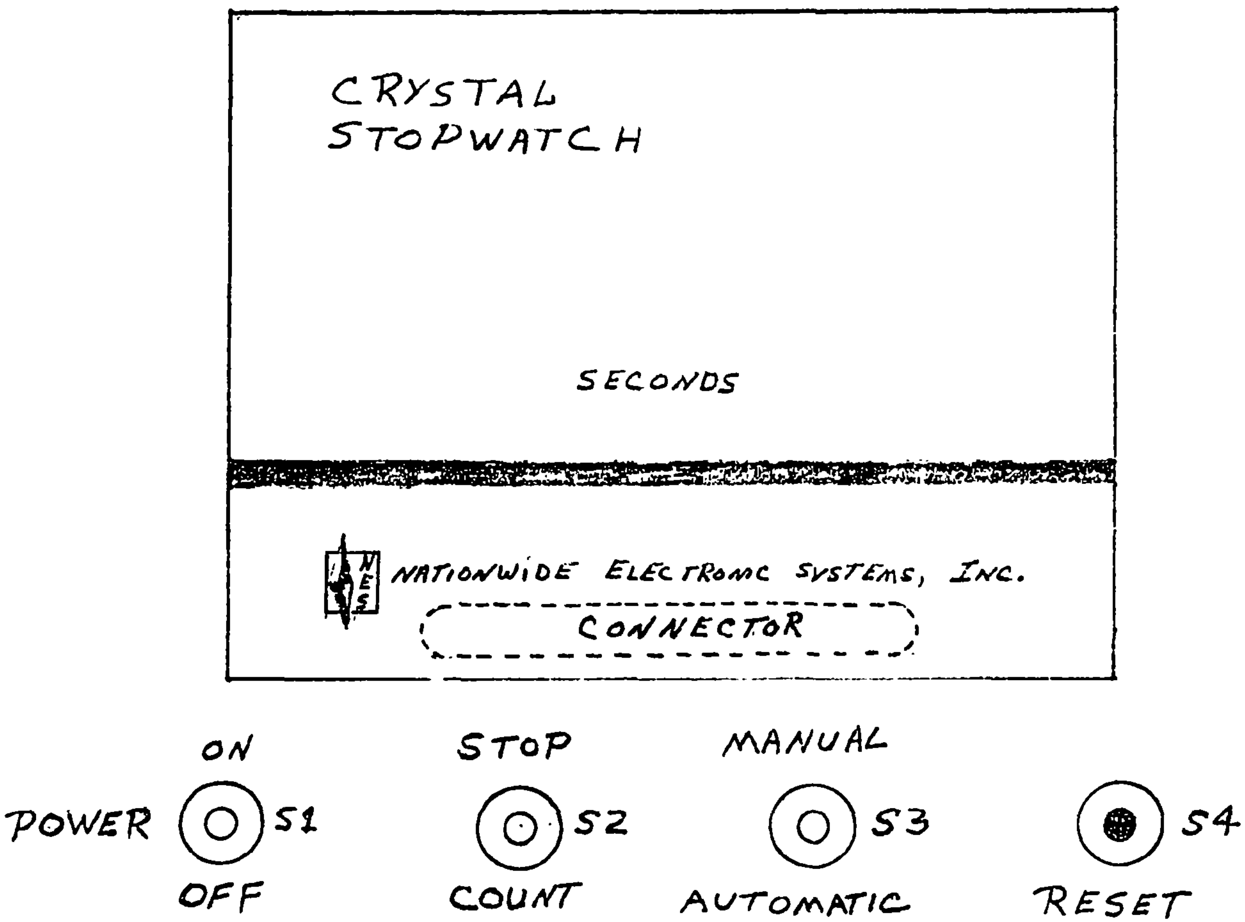

Clayton

FRONT PANEL LAYOUT



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4-14-78

REAR PANEL LAYOUT

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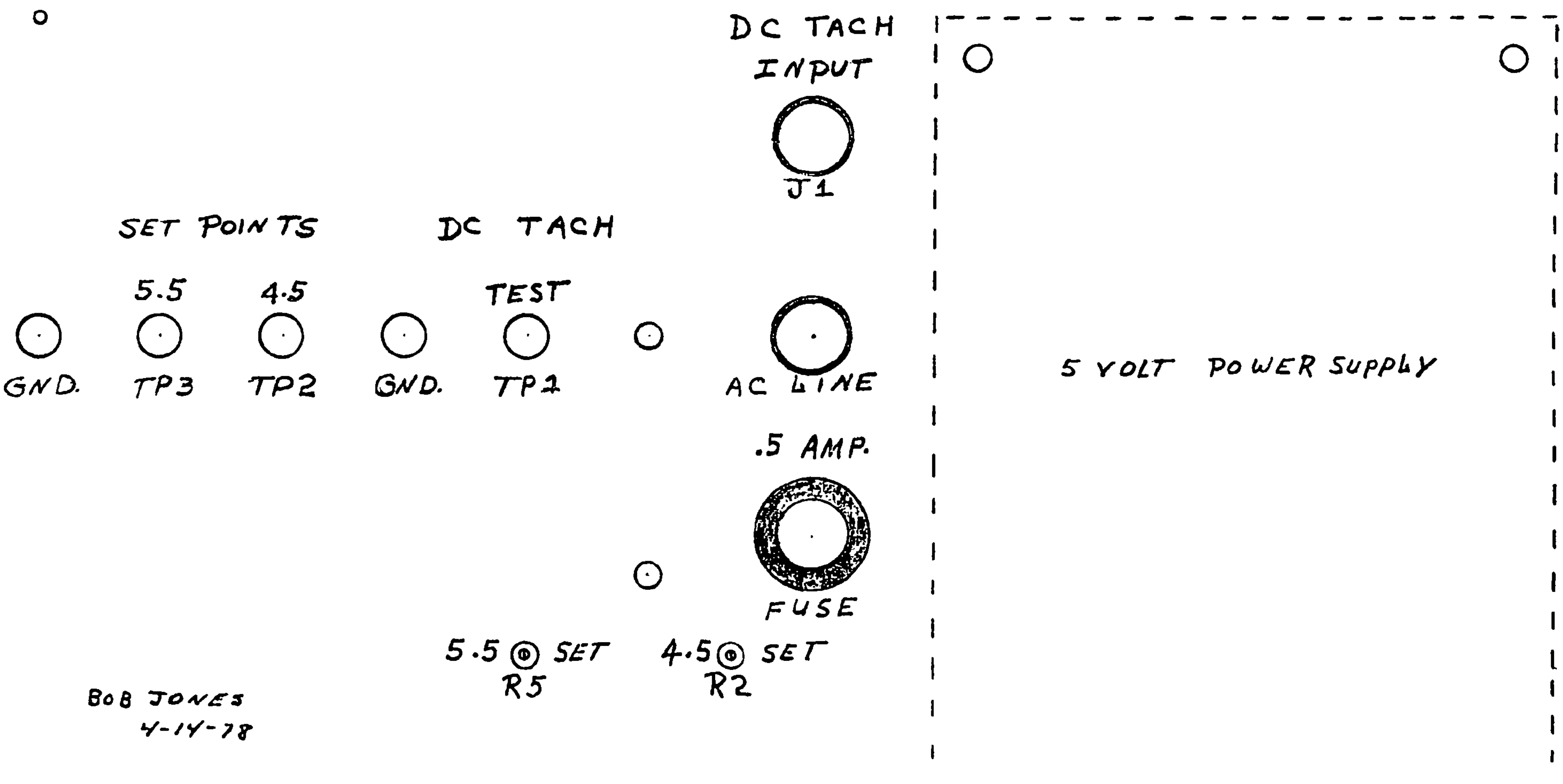
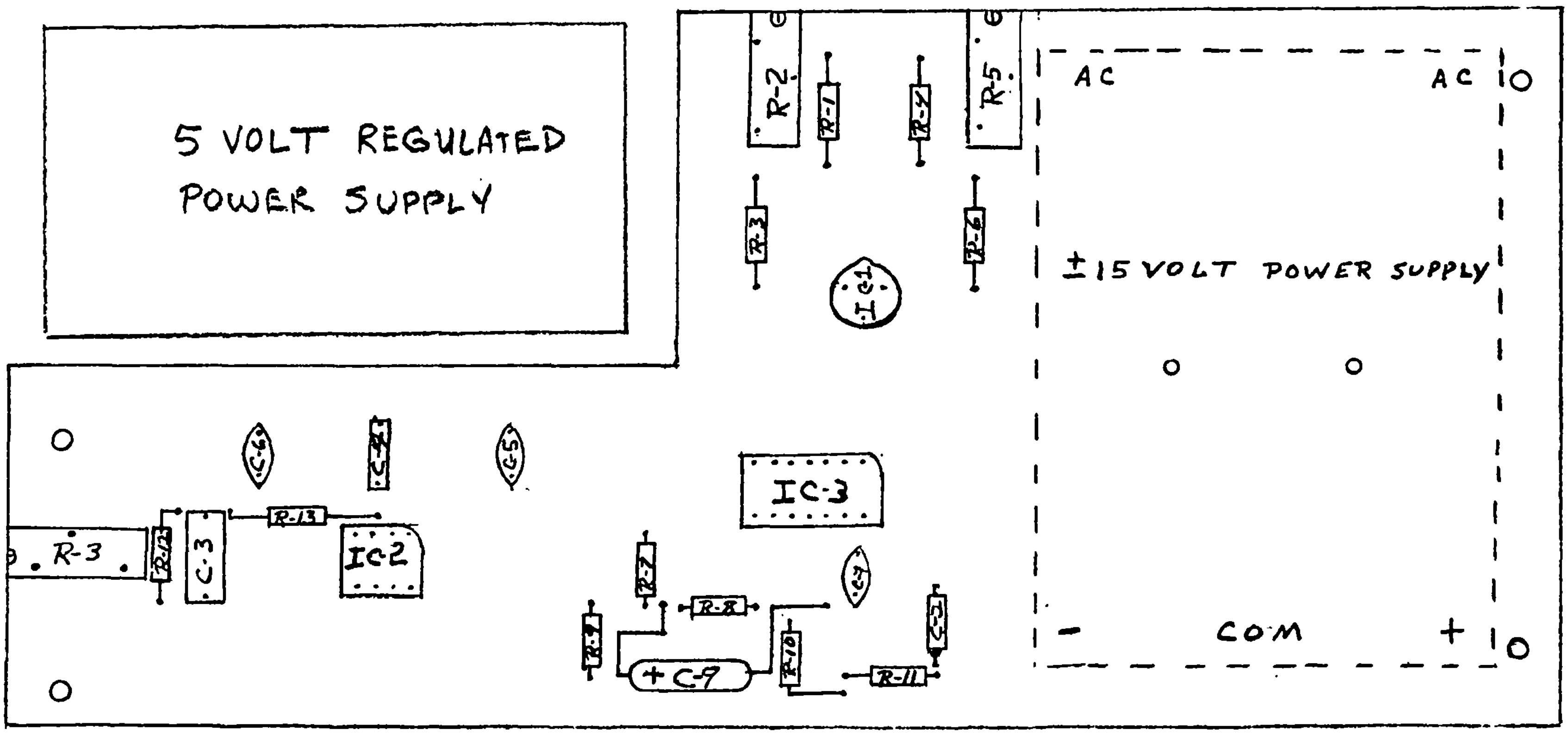


PC BOARD LAYOUT



BY BOB JONES
4-17-78

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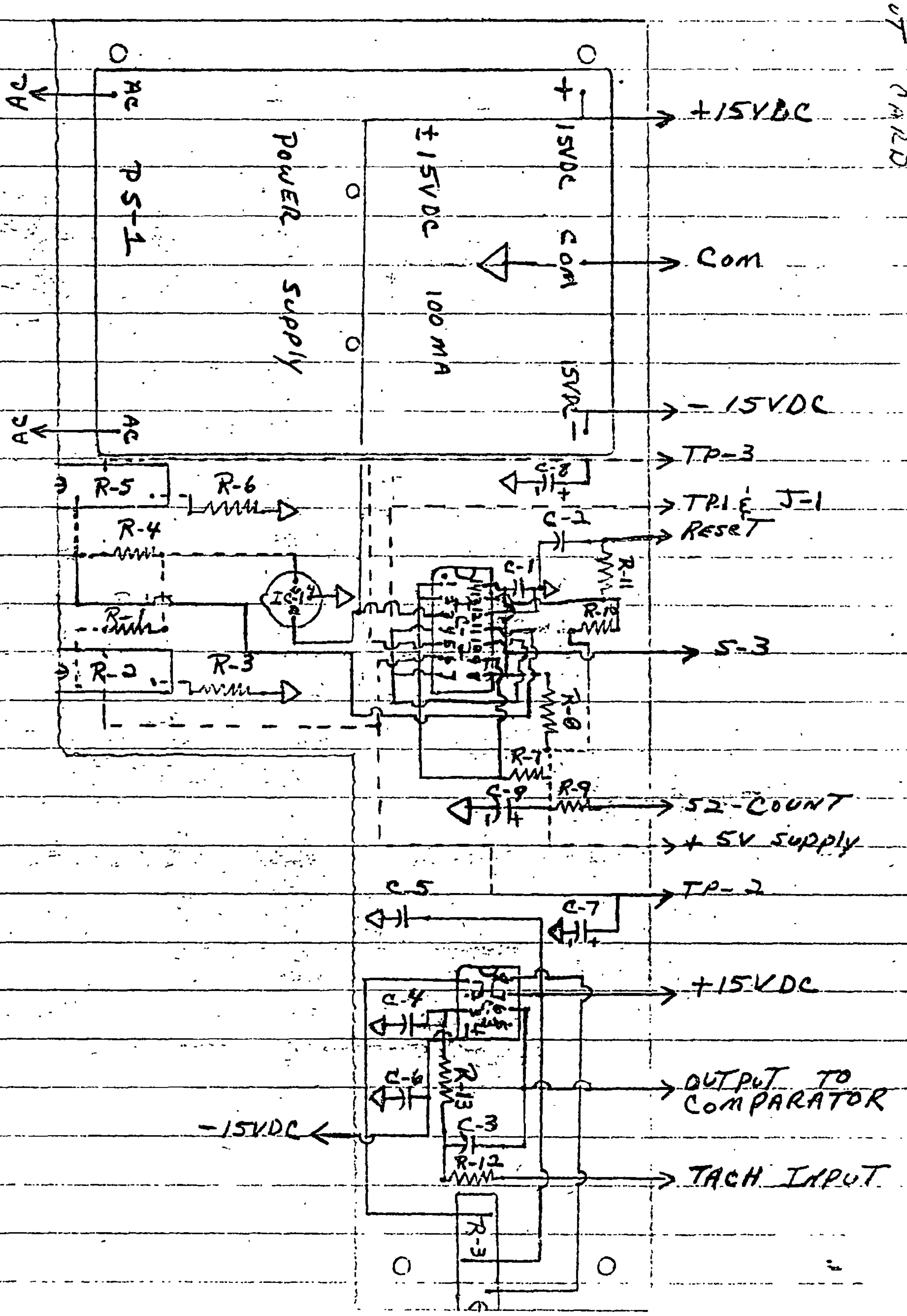
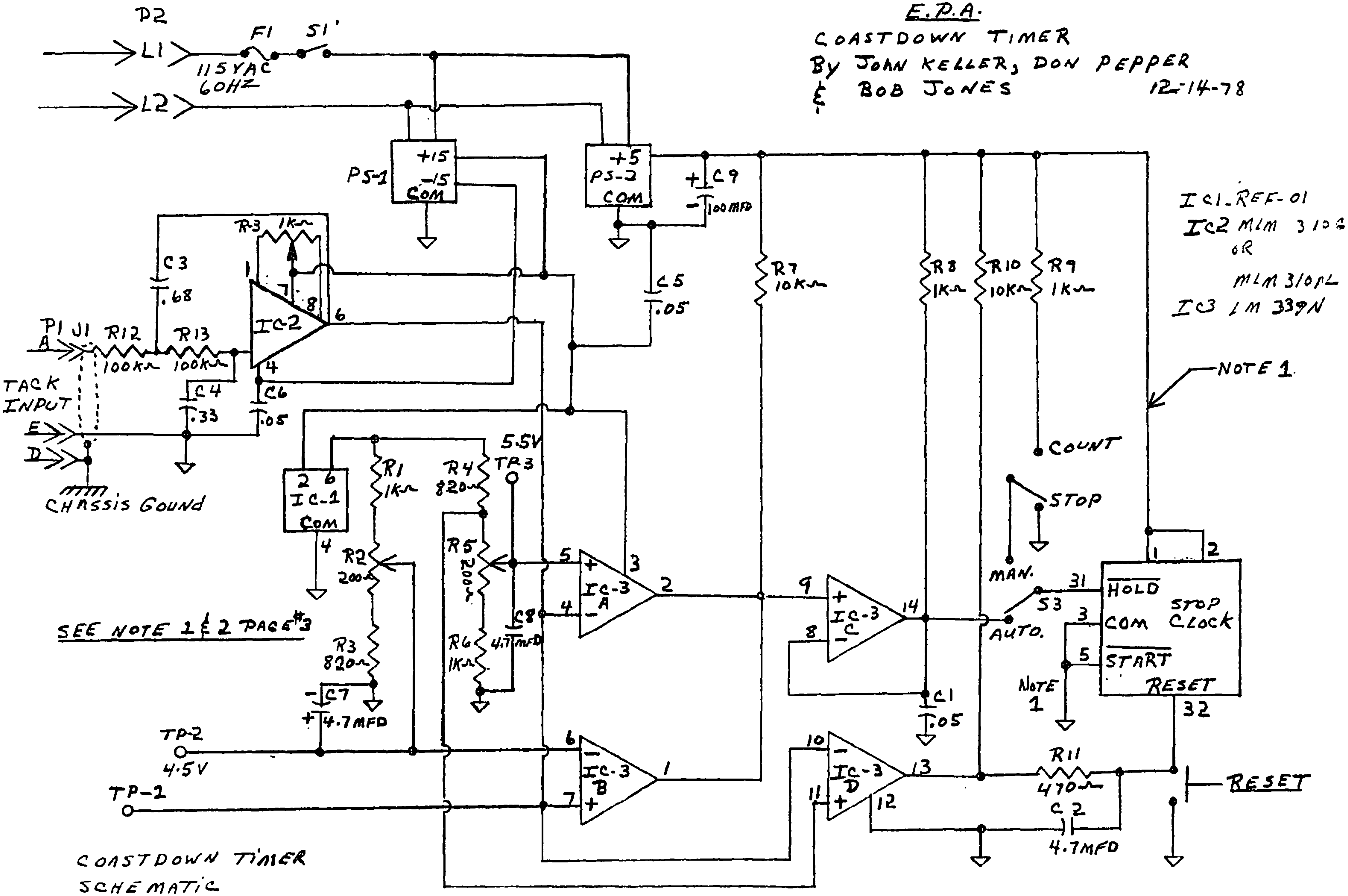


R. Jones
2/27/78



SEE NOTE 1 & 2 PAGE #3

COASTDOWN TIMER
SCHEMATIC

COASTDOWN TIMER PARTS LIST

POWER SUPPLY #1 — ±15VDC, 100 MA SOLA ELECTRIC, 84-15-02110E

POWER SUPPLY #2 — +5VDC 3A POWER-ONE, MODEL BS-3

IC 1 — 10V REFERENCE DIODE, PMI REF-01CJ

IC 2 — UNITY GAIN OPERATIONAL AMPLIFIER, MOTOROLA MLM-310G OR 310PL

IC 3 — COMPARATOR, LM-339AN

STOPWATCH - NATIONWIDE ELECTRONIC SYSTEM, INC. MODEL DS 1461. 99.99 SEC.

RESISTORS	R1, R6, R8 & R9	1KΩ 5%, 1/4 WATT CARBON
"	R3, R4	820Ω 5%, 1/4 WATT CARBON
"	R7, R10	10KΩ 5%, 1/4 WATT CARBON
"	R11	470Ω 5%, 1/4 WATT CARBON
"	R12, R13	100KΩ 5%, 1/4 WATT CARBON
POTENTIOMETER	R2, R5	200Ω, 20 TURN BOURNS 3069P-1-201
"	R3	1KΩ, 20 TURN BOURNS 3069P-1-102
CAPACITORS	C1, C5, C6	.05 MFD. 25WVDC DIS CERAMIC
"	C2, C7, C8	4.7 MFD 10WVDC ELECTROLYTIC M39003101-2014
	C9	100MFD 20WVDC ELECTROLYTIC M39003101-261

PARTS LIST CONTINUED

CAPACITORS	C3	.68 MFD 100WVDC CERAMIC
	C4	.33 MFD 100WVDC CERAMIC

TOGGLE SWICHES S1, S2, S3 SPDT ALCO MST 105D

MOMENTARY N.O. PUSHBUTTON ALCO MPA 103C

LINE FUSE .5 AMP.

PLUG, TACH IN P1 APHENOL 126-016

RECEPTICAL TACH IN J1 AMPHENOL 126-223

LINE PLUG & CORD 18-3 TYPE SVT E-3462 LL 7874

FUSE HOLDER BUSS HEM

BANANA JACKS 108-0903-001, JOHNSON

CABINET BUD COWL TYPE MINI-BOX SC-3030

PC VECTOR BOARD 169P84-062

NOTE: #1 USE #18GA. WIRE FOR COMMON & +5VDC CONNECTIONS BETWEEN CLOCK & POWER SUPPLY

#2 ADJUST IC-2 FILTER BALANCE FOR 5.000VDC OUTPUT WITH 5.000VDC INPUT

☆ U.S. GOVERNMENT PRINTING OFFICE: 1979- 650-029/0059