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Technical Support Report for Regulatory Action

Performance and Cost Analysis of Chassis Dynamometers

by

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Office of Air and Waste Management
U.S. Environmental Protection Agency

Performance and Cost Analysis of Chassis Dynamometers

Abstract

The purpose of this report is to identify the types, makes and models of light-duty chassis dynamometers which are presently available for purchase. Particular attention is given to those types which can be used for measuring exhaust emissions by the current Federal Testing Procedure; i.e., units with adjustable inertia simulation capability and adjustable steady-speed power absorption capability. The required inertia range is from 1,750 to 5,500 lbs. and the corresponding 50 mph power range is from 7.7 to 15.8 horsepower. The dynamometers applicable for emissions testing are then compared on the basis of performance and cost.

Introduction

The first task in this project was to contact American marketers of light-duty vehicle chassis dynamometers. A list of companies was made using the "Thomas Register of American Manufacturers, 1973" and a listing of chassis dynamometer manufacturers from the report entitled, "Development of Specifications for a Motorcycle Dynamometer and Motorcycle Cooling System" by Olson Laboratories. The marketers which were contacted in this study are listed in Table I. As shown, of the 19 companies contacted, 10 manufactured or marketed chassis dynamometers; and of these 10 there were four whose dynamometers were capable of simulating both speed-load relationships and vehicle inertia weight. Consequently, four companies produce dynamometers which meet the basic requirements of the Federal Test Procedure. These are Burke E. Porter Machinery Company, Sun Electric Corporation, Clayton Manufacturing Company, and Laboratory Equipment Company (Labeco).

As listed in Table I, only Clayton uses a hydrokinetic power absorption unit (PAU). The other three marketers use electric PAU's. Porter uses regenerative direct-current (DC) absorption systems and Sun uses an eddy current absorber. Labeco builds many different types of large roll units. Their system which looks most attractive for emissions testing uses an eddy current absorber. The mechanical components of this unit are the same as the Model 100-75P mileage accumulation dynamometer. It would require some modifications to the control system for emission testing applications. Labeco could assembly other type systpms specifically for emission testings but these would be custom machines, requiring some basic design work. Additional information regarding the dynamometers sold by Clayton, Porter, Sun and Labeco is listed in Table II and is discussed below.

Discussion

A. Roll Configuration

One obvious difference among dynamometers is roll configuration and size. These parameters have an effect on tire rolling resistance which is related to the amount of power loss in the tire. Ford Motor Co. has done some recent experimentation to determine the effects of roll size and tire type on rolling resistance (1). They conducted tests with bias belted and radial ply tires. At 25 psi both type tires showed approximately twice the rolling resistance on the 9.5" diameter, double-roll configuration of the Porter Model 1059 as on a flat road. When these tires were inflated to 45 psi on the dynamometer there was a noticeable drop in rolling resistance; but it was still substantially greater than on a flat road at 25 psi.

Experiments have also been conducted to compare rolling resistance on large diameter single rolls to a flat road. Ford's data of this type indicate very little difference in rolling resistances between a 48 inch roll and a flat road. The SAE Tire Power Consumption Task Group is presently conducting tire power consumption tests on various roll sizes. Although testing is not complete, Calspan Corporation data showed that tire rolling resistance on a 67-inch diameter roll was also approximately the same as on a flat surface. Both surfaces had the same texture. Some tests have shown that a 67" steel roll with a smooth surface gives less rolling resistance than a flat road surface; however this could be a surface texture effect.

Sun Electric Corporation was contacted in order to obtain data on tire rolling resistance on their 21.6" diameter, double-roll configuration. During the week of October 27-31, 1975, GM ran a series of tests on one Sun dynamometer in order to compare it to a Clayton unit. One of the parameters which was investigated was tire rolling resistance. Rolling resistance measurements were made using a wheel torque meter with a 13 inch tire. Results of this test are shown in Figure A-1 of the Appendix. The Sun dynamometer gave approximately 15 percent less rolling resistance than the Clayton dynamometer in the 40 to 60 mph range. At a speed of 50 mph, a rolling resistance of 10 lb was equivalent to approximately 1.2 hp, so the observed difference in rolling resistance at 50 mph represents a 0.4 hp difference in tire power consumption.

From the rather limited amount of testing that has been done concerning dynamometer roll configurations effect on rolling resistance, investigators have found that there can be a substantial difference due to tire brands as well as basic tire construction. For example, a certain brand bias ply tire may have very nearly the same on-road rolling resistance as a certain brand radial ply tire; however, two other brands of bias and radial tires may have much different on-road rolling resistances. Dynamometer tire correction factors could be developed for different types of tires; however, because of the variation between different brands of tires, these correction factors should represent the mean value for each tire type. A test program involving all brands of tires concerned would be required to accurately predict such correction factors.

Some data presented in a previous Ford Motor Co. SAE paper (2) showed that the power consumed by a radial tire was greater than that of a bias tire when measured on a dynamometer. The paper indicated (although no data were given) that the reverse of this situation was observed on the road. Personnel connected with this test work, now have some reservations about the validity of these particular data. Some of Ford's more recent data, in reference (1), show that although a certain radial tire did have more power consumption than a certain bias tire on dynamometer tests, the same situation was true on a flat road. This relationship was observed for a certain brand and size of bias and radial tire, and it can not be generalized to the type of tire used.

B. Power Absorption Units (PAUs)

As listed in Table II there are three general types of power absorption units used in chassis dynamometers suitable for light duty vehicle exhaust emission testing. These are the hydrokinetic (water brake) unit used by Clayton, eddy-current units used by Sun and Labeco and the direct-current absorbers used by Porter. The least expensive (and least versatile) of these units is the hydrokinetic absorber. In this unit the mechanical energy from the rolls is used to do work on a specific amount of water contained within the absorber. The water rises in temperature and this heat energy is carried away by cooling water supplied to the unit. The amount of power absorbed by a particular hydrokinetic unit is determined by the physical configuration inside the absorber unit. This relationship is of the form

$$P = AV^a$$

where A is determined by the quantity of water in the unit and V is the shaft velocity. "a" is dependent on construction in the unit. For Clayton dynamometers, "a" has an average value of 2.83 and a range of 2.81 to 2.87 (3).

In addition to the power absorbed by the PAU itself there is also an amount of power absorbed due to bearing friction and air resistance of moving parts. This amount of power absorption is commonly called the frictional power loss of the dynamometer, and it is the difference between the power applied to the dynamometer rolls and the power absorbed by the power absorption unit. In a Clayton dynamometer, friction commonly accounts for from 2 to 4 horsepower at 50 mph. The amount of friction is strongly dependent on the inertia setting and shows much less dependency on the power absorber setting. This is shown in Figure 1 which presents the frictional power loss of one Clayton dynamometer as a function of dynamometer speed. From these data it appears that the frictional horsepower is not a linear relationship, particularly at the higher inertia values.

Considering both the dynamometer friction and the PAU, the road-load power absorbed by the entire Clayton chassis dynamometer is of the form

$$P = KV + AV^{2.83}$$

where K will vary with dynamometer speed and vehicle inertia weight. For the Clayton dynamometers which are now in the EPA lab, the values of frictional horsepower at 50 mph have changed very little or none since their installation, so the value of K at 50 mph appears to be quite stable for a particular dynamometer. Since frictional power determination is not routinely done at other speeds, any changes which may occur in the shape of the frictional power curve are not known.

Since the actual amount of water in the hydrokinetic units is not measured, the unit to unit variability of A and any long-term time change in A is not known (and is really of no practical concern since the steady-state 50 mph hp requirement is set before each test). However, the short-term change in A is worth some comment. A vehicle's road-load power at 50 mph is set at a steady-state condition. However, some time is required for the water in the hydrokinetic unit to reach a steady-state condition at any constant shaft speed. Due to this hysteresis effect, the load at any certain speed during a driving cycle may be substantially different from the steady-state load, and the amount of hysteresis is not constant from unit to unit. Data has been obtained from an automotive manufacturer who has conducted hysteresis tests on some of their Clayton dynamometers. These showed that some units had almost no time lag, whereas other units had a hysteresis of up to ± 1 ft-lb for accel. and decel. rates of 3.0 mph/sec (typical acceleration in the emissions driving cycle) in the 10 to 40 mph speed range. Unit to unit differences are believed to be mainly due to variations in the heat exchanger section. For a 4500 lb vehicle at 20 mph, the torque supplied by the dynamometer is about 17 ft-lbs. So this is a maximum hysteresis error of ± 6% in the load vs speed curve. For acceleration rate of 1.0 mph/sec the maximum hysteresis effect dropped to about ± 0.7 ft-lbs torque. This is a ± 4% variation in the load vs speed curve at 20 mph.

Although the hysteresis effect does account for a sizable error in the transient load vs speed curve, it must be put in the correct perspective. The previous type of analysis causes the hysteresis effect to appear more serious than it really is during actual vehicle operation. The total dynamometer torques required (including inertia simulation) for accel rates of 3.0 and 1.0 mph/sec are approximately 260 and 100 ft-lbs respectively, for a 4500 lb vehicle. Therefore, for accelerations of 3.0 and 1.0 mph/sec, the Clayton dynamometer hysteresis effect results in a total dynamometer torque (or horsepower) error of ± 0.4% and ± 0.7% respectively, at 20 mph, and at higher speeds, this percentage error decreases.

Probably the most important criteria for any dynamometer is the accuracy to which it duplicates vehicle load as measured on the road. The hysteresis effect for the hydrokinetic unit has already been discussed. The following discussion omits consideration of the hysteresis affect and only steady-state values of dynamometer speed-load curve are considered.

Clayton states that vehicle power required to overcome wind resistance is given by the relationship

$$P = V^b$$

where b varies between 2.8 and 2.9. Because of this, Clayton has designed their PAUs so they have this same relationship between power and speed. However, due to dynamometer and vehicle frictional losses, this does not mean that the chassis dynamometer unit will give the correct relationship between vehicle speed and road load power. In fact, data supplied from GM on nine of their vehicles indicate that the Clayton dynamometer has a noticeably different speed-power curve than the average vehicle speed-power curve on the road. This information is shown in Table III. When the passenger cars are set up on a dynamometer at their true road-load hp at 50 mph, all of them are loaded lighter than their true road-load power at speeds less than 50 mph and heavier at speeds above 50 mph. The difference between dynamometer and true road-load power (dyno minus road) at 20 mph ranged from -0.3 hp (-12%) to -1.0 hp (-33%) with an average value of -0.5 hp (-18%). At 35 mph the range was from -0.1 hp (-2%) to -1.0 hp (-14%) with an average difference of -0.5 hp (-7%). GM also supplied data on a Chevrolet C-10 pickup and this road-load comparison is also listed in Table III. As shown, the dynamometer road-load power for this vehicle was 28% and 10% higher than measured on the road at speeds of 20 mph and 35 mph, respectively. These tests showed that even though a vehicle's 50 mph load may be matched on a Clayton dynamometer, the vehicle's true speed-power relationship may not be (and in many cases is not) accurately simulated on the dynamometer.

Typical day to day variability of a Clayton dynamometer in regard to steady-state road-load power at any speed is about ± 3%. This includes variability in either the manually loaded or automatically loaded 50 mph power setting, both of which are approximately ± 2%. Temperature variations under typical stabilized conditions may account for another ± 1% in day to day operation.

Unit to unit variability in Clayton dynamometer steady-state road-load power is due to both differences in friction and differences in PAU's. Under constant temperature conditions, PAUs which are set up to give the same load at 50 mph will vary by about ± 1% in their loads at any speed between 20 and 60 mph (3). Unit to unit variations in friction makes a greater contribution than PAU variability in regards to the variability in road-load power between dynamometers. Of the eight light-duty dynamometers at the EPA MVEL, the 50 mph frictional power at the 4000 lb inertia setting ranges from 2.0 to 4.0 hp. Assuming frictional power curves which are of similar shape to those in Figure 1, this 2 hp difference will give variations in the total road load power of up to ± 3% in the 20 to 60 mph range. Table IV summarizes the sources and magnitudes of dynamometer variability.

The most significant difference between hydrokinetic PAUs and electric PAUs is the ability of the electric units to change the shape of the load vs speed relationship. As Table II showed, the load vs speed torque (T) equation for the DC absorbers is

$$T = B + DV + CV^2$$

where B, D and C are constants which can be adjusted with potentiometers The B, DV, and CV^2 terms are commonly referred to as grade, friction and windage simulation terms respectively. By adjusting the values of B, D, and C it is possible to duplicate individual speed vs power curves within $\pm$ 3% from 15 to 60 mph. Variability among DC PAUs with similar control systems is quite small and unit to unit variability is about $\pm$ 1%. Variations in frictional characteristics between dynamometers does not cause a variability problem because of the ability to change shape of the torque curve. Day to day variation in a particular DC unit involves errors in resetting the potentiometers and the variability in torque values for a given current input due to temperature differences. Under typical warmed-up conditions this variability is about $\pm$ 0.5% throughout the speed-load curve.

The two largest suppliers of DC power absorption units are General Electric and Reliance Electric. Burke E. Porter will supply either brand with their Model 1059 unit and the two systems are competitive in price and performance. The type of absorber used in this dynamometer is a DC motor which may vary in size from 30 to 60 hp. These are less expensive than the larger torque capacity DC motor-generator sets which, because of the slower roll speed, are necessary on the large roll dynamometers. The large roll Porter (Model 1098) uses a General Electric MG system. Regardless of which type of DC power absorber is used, the accuracy of the system depends on the control system which is used. The electric dynamometer accuracy information listed in Table IV is a compilation of specifications and estimates from General Electric and data from organizations which have purchased and/or tested such dynamometers.

Power absorbers used by the Sun and Labeco Systems are eddy current dynamometers. The torque (T) of the Sun PAU is controlled by an equation of the form

$$T = B + CV^2$$

where B and C are adjustable values. Sun Corporation has conducted tests to compare the road-load power curve of their unit to that of a Clayton dynamometer. The test consisted of running a vehicle equipped with a wheel torque meter on both a Clayton and a Sun dynamometer. Under warmed-up conditions, wheel torque was recorded as the vehicle was run at steady speeds of 10, 20, 30, 40, 50 and 60 mph. The data show that the Sun unit is capable of duplicating the wheel torque measurements obtained on the Clayton dynamometer to within $\pm$ 3% throughout the speed range. It is estimated that specific vehicle road-load curves could be duplicated with the same degree of accuracy.

Power absorption in the Labeco System (modified 100-75P mileage accumulator) is done quite differently than in the other dynamometers we have considered. The dynamometer rolls are mechanically connected to a large centrifugal fan which directs air flow to the front of the vehicle. Air is blown on the vehicle at the same velocity as the vehicles rear wheels, so air flow during road driving is simulated very closely. Because of this arrangement, most of the dynamometer load is supplied by the fan. If some additional loading is required, the eddy current absorber supplies it. In the cases where the fan absorbes more power than the vehicle requires, a constant speed motor and clutch supply a motoring torque to the rolls. The torque equation for the absorber and motor is of the form

$$T = B + CV^{d}$$

where B, C and D can be changed independently. This provides an added degree of flexibility over Sun's control equation. Labeco has not attempted to determine how closely this system can match individual vehicles' road load vs speed curves. It is estimated that it would be at least as accurate as the Sun eddy-current dynamometer.

In addition to the fact that electric power absorption units can be adjusted to simulate specific load vs speed relationships, the hysteresis effect in the road-load curve of the electric dynamometers is also less than that in the Clayton dynamometers. For the electric units, calculations based on the system response times indicate that the maximum hysteresis error in the emissions driving cycle is $\pm$ 0.3% of the total dynamometer power requirement.

The degree to which differences in road-load power will affect exhaust emission and fuel economy measurements is dependent on the particular driving cycle. Of the power transmitted to the dynamometer by a 4000 lb inertia class vehicle operating according to the Urban Dynamometer Driving Schedule (UDDS), approximately 35% goes into the road-load requirement and 65% goes into the inertia requirement. On the Highway Fuel Economy (HFE) Driving Cycle, approximately 70% goes into the road-load requirement and 30% into the inertia requirement. These values were calculated from the speed vs time specifications of the driving cycles, and additional information concerning these calculations is contained in Table A-I of the Appendix.

Some test work has been conducted to determine any effect of changes in road-load requirement on exhaust emissions and fuel economy (4). Of the work performed in reference (4), one vehicle was given multiple tests at each of three different road-load power values on both the UDDS and the HFE cycle. On both of the driving cycles there was a statistically significant effect (at greater than the 95% confidence level) of road-load on fuel economy. On the UDDS, the magnitude of the effect was approximately 1 percent for a 10 percent change in the 50 mph horsepower. On the HFE cycle the effect was approximately 2.6 percent for a 10 percent change in the 50 mph horsepower. The only exhaust emission value which was significantly affected (at greater than a 95% confidence level) by road-load was NOx on the UDDS. The magnitude of this effect was about 2.4 percent for a 10 percent change in the 50 mph horsepower requirement.

C. Inertia Simulation

As shown in Table II, there are different methods of dynamometer inertia loading. Clayton dynamometers use a set of five flywheels which are driven directly from the front roll shaft. Although early model Claytons had inertia wheels which were belt driven from the drive-roll shaft, this system is no longer available. Belt durability was a major problem with this arrangement. Also, correlation tests have shown that the belt-driven arrangement caused greater vehicle loading than the direct-drive system.

The PAU in the Sun unit is a 250 hp eddy current absorber. It can be used to simulate inertia on acceleration, but it can not motor the vehicle as is required on deceleration. Therefore, for emission testing, Sun has available a set of inertia flywheels. These flywheels are driven at 2 1/2 times drive-roll speed by means of two cog belts from the drive roll shaft. Since the rolls of the Sun dynamometer are about 2 1/2 times larger in diameter than those of the Clayton, the flywheel speed of the two units is very nearly equal. Unlike other double roll dynamometers, the front and rear rolls of the Sun unit are connected by a cog-belt (similar to the one used to drive the flywheels). Ford has had one of these dynamometers in operation for about two years and is pleased with its operation. They have had no instances of belt breakage or slippage.

The Labeco unit uses a set of seven flywheels for inertia simulation. These flywheels are located in the drive line that connects the dynamometer rolls to the centrifugal fan, and they can simulate vehicle weight from 2,000 to 9,700 lbs.

The standard method of inertia loading on the small roll Porter (Model 1059) is by the DC power absorption unit. Unlike an eddy current absorber this unit can simulate inertia both on acceleration and deceleration. However, if mechanical inertia is desired the unit can be supplied with a set of inertia flywheels.

The large roll Porter (Model 1098) dynamometer uses exclusively the PAU for inertia simulation. Mechanical inertia would be rather impractical and expensive for this unit because of the relatively low roll speed. If inertia flywheels rotated at roll speed, the mass required would be extremely large and/or located a large distance from the axis of rotation. If the inertial mass were to be kept approximately the same size as Clayton's flywheels, then the rotational speed would have to be geared up to several times that of the rolls. This could create durability problems.

As the above discussion indicates, there are basically two methods of inertia simulation--mechanical and electrical. For dynamometers which use a DC PAU, electrical inertia has certain advantages over mechanical inertia. One advantage is lower cost. The added absorber size and control system needed for the inertia simulation costs less than the price of an inertia flywheel system. A DC dynamometer system is also smaller in size when not equipped with flywheels; and since electrical inertia requires no additional mechanical parts, there may be an advantage from the standpoint of system durability.

For dynamometers which use hydrokinetic or eddy-current PAU's, mechanical inertia has a definite cost advantage over electrical simulation. A hydrokinetic equipped unit would require a DC absorber for electrical inertia. An eddy current absorber could supply acceleration inertia to the rolls, but a motor and accompanying control system would be required to electrically simulate deceleration inertia. Both of these systems are more expensive than a set of flywheels.

A major consideration in inertia simulation is accuracy and repeatability. In these areas, mechanical inertia has a distinct advantage over electrical simulation. The control system for electrical simulation monitors roll speed and from this, calculates acceleration. Rate of application of the inertia load is dependent upon the response time of the system. Consequently, there is always some time lag between change in roll speed and application of the correct inertia load. A typical value of the inertia response time (to 90% of value) for a step change in vehicle torque is 0.5 seconds. The system also has a "settling time" which refers to the time that it takes the inertia torque to settle down to the new steady state value. This time may vary from 1 to 5 seconds. The greater the difference between vehicle weight and mechanical inertia of the dynamometer, and the greater the change in vehicle torque, the greater will be the transient inertia error. Consequently, accuracy of transient inertia simulated by electrical means varies with vehicle size and specific test cycle. In contrast to this, mechanical inertia simulation involves essentially no time lag. So improvements in response of electrical inertia simulation can only result in an accuracy which approaches the accuracy already contained in the inertia flywheel system.

During a driving cycle there is never a true step change in vehicle torque and most throttle position changes and acceleration rates during the emission test cycle are moderately slow. Consequently, the transient inertia error may have only a small effect on emission levels. Perhaps just as important (or maybe more so) as the transient error in electrical inertia simulation is the error in the steady-state inertia (ie., stabilized inertia at a constant acceleration) values. As listed in Table IV, the accuracy of steady-state electrically simulated inertia value is $\pm$ 5%. This is substantially more variability than the Clayton flywheel system which is accurate to $\pm$ 1%. Although the electrical simulation of steady-state inertia may be in error by as much as ± 5%, the repeatability error is only about ± 1%. Therefore, only a very small increase in emission variability would result from a change from mechanical to electrical simulation. A somewhat larger change could occur in the absolute level of emissions due to the ± 5% range in inertia accuracy. As in the case of an inaccurate speed-power relationship, it is difficult to predict the effect of such inertia inaccuracies on absolute emission levels. To do an evaluation would involve determining how the simulated inertia differed from the true inertia throughout the driving cycle. Perhaps a better method for determining the difference in emissions between a flywheel system and electrical simulation would be to equip a DC dynamometer with a set of flywheels and run a series of tests using each type of inertia. I am not aware of any such data. Obviously, such a test would not isolate the transient inertia error effect from the steady-state error effect.

Another complication involved in use of electrical simulation is system calibration and routine varification of accuracy. In the case of direct-drive mechanical simulation, the various inertias can be calculated from the size of the flywheels. The inertia loadings can then be verified by simple visual inspection of which flywheels are turning. To determine the actual inertia loading by electrical simulation would require measuring the resistance of the dynamometer drive-roll to speed changes. To determine the correct operation of the complete system would require these measurements at each inertia setting (both accel and decel) and at different speeds within each inertia setting.

In regard to system reliability, there is relatively little experience with electrical inertia systems used for emission testing. It is not known if this system would be more or less durable than the flywheel system.

D. Cost

Table IV contains price information on the dynamometers. The least expensive units are the Clayton dynamometers, which cost about $20,000. The $8,000 site preparation and installation charge assumes that there is already power in the room and that dewatering of the pit is not necessary (the typical situation). It does include a service trench for water and air, pit digging, concrete pouring and machine installation.

The complete Sun dynamometer price is about $40,000 and the site preparation costs are somewhat higher than for a Clayton Dynamometer because of the larger roll size.

A Burke E. Porter Model 1059 dynamometer electrical inertia simulation would cost approximately $53,000. The DC absorber and control system accounts for about $35,000 of the total price. This cost varies somewhat depending on the particular power absorption unit and control system which is used. If mechanical inertia is desired, the total cost would be about $60,000.

The Burke E. Porter Model 1098 and the systems built by Labeco are generally used for mileage accumulation. However, the control systems on these units are the same (or could be the same) as on electric dynamometers intended for emission testing work. These large roll units are normally equipped with large centrifugal fans. These fans are desirable in that they simulate road air flow much better than the fans which are presently used for emission testing. The complete Burke E. Porter unit costs about $105,000.

The Labeco system is about $70,000 and this includes a $7,000 push button inertia selection option. It is less expensive than the Porter unit mainly because of the cost difference between eddy-current and DC MG systems. The fan on the Porter unit is driven by its own DC motor. Therefore, it would be quite easy to remove this from the system if desired, and cost would drop to about $75,000. Since the fan in the Labeco unit provides a part of the road load, it would be more difficult to omit from this sytem, and the cost savings would not be as great. The site preparation and installation cost for the two large roll units includes $15,000 for a six foot deep pit with a sump pump and $8,000 for machine installation and wiring.

Summary and Conclusions

Tire rolling resistance is dependent on tire type, brand and size as well as dynamometer roll configuration. Clayton dynamometer roll configuration gives approximately twice the tire rolling resistance as a flat road, a 67 inch diameter single roll gives about the same rolling resistance as a flat road, and intermediate configurations have intermediate effects.

The Clayton dynamometer gives a steeper load vs speed curve than that exhibited by an average passenger car. This results in dynamometer loads that are less than actual road loads at speeds under 50 mph. At 20 mph this difference ranges from about 10% (0.3 hp) to 30% (1.0 hp). An electric power absorption dynamometer has the ability to duplicate load vs speed curves to within $\pm$ 3%. However, to achieve this type of accuracy would likely require on-road torque measurements on every vehicle before each emission test. If this were done, then differences in tire rolling resistance between the road and the dynamometer would be automatically taken into account.

If on-road torque measurements were not made before each emissions test, then an electrical dynamometer could be set to give a fixed load vs speed curve shape which would best represent a majority of the vehicles. This would give some increase in accuracy over the present hydrokinetic unit which underloads the average passenger car at speeds under 50 mph. This alone would not solve the problem presented by certain types of vehicles (pickups, vans, etc.), which have considerably more wind resistance than passenger vehicles. However, the proposed light duty truck regulations have made a correction for this discrepancy. The remaining differences between true road load at 50 mph and the Federal Register values could be minimized by taking into account individual vehicle aerodynamics. This could be done by considering vehicle frontal area and/or drag coefficient when determining the 50 mph horsepower requirement.

When a predetermined power setting is used for each class of vehicle, tire rolling resistance (power consumption) becomes of concern. The important question is whether or not tires have the same relative rolling resistance on the dynamometer as they do on the road. Sufficient test data are not available to answer this question. If relative tire behavior on the dynamometer is similar to that on the road, then there is no need to consider differences in tire rolling resistance. The vehicle with the "poorer" tire simply has to work harder. However, if one tire has greater power consumption on the dyno than another tire, and if the reverse of this is true on the road, then there are two methods of correcting this inaccuracy. One method is to change to a dynamometer roll configuration which simulates the road. The other is to establish correction factors for individual brands and sizes of tires. Experimental work is currently being done by an SAE committee and by the EPA which should help identify the effect of roll configuration on relative tire power consumption.

If the relative tire rolling resistance is different between the road and the Clayton dynamometer, serious consideration should be given to a roll configuration which accurately simulates the road. Calspan Corporation has recently submitted to EPA an unsolicited proposal to build and test a dynamometer with a flat tire contact surface. The proposal includes a testing program which would compare the new unit with a Clayton dynamometer.

In the area of inertia simulation, mechanical inertia is superior to electrical inertia in the areas of accuracy, repeatability and ease of calibration. Direct-drive or positive-drive flywheel system are the recommended type.

Recommendations

1. Direct-drive (or positive drive) mechanical inertia should continue to be used on emission (and fuel consumption) test dynamometers.

2. Present inaccuracies in the 50 mph road load horsepower settings, as listed in the Federal Testing Procedure, could be minimized if both vehicle weight and frontal area were taken into consideration.

3. Data should be obtained to compare relative power consumption between the road and the Clayton dynamometer. If the current dynamometer roll configuration behaves differently than the road in regards to relative tire power consumption, then it could be more accurate (and possibly less expensive on the long term basis) to change roll configuration rather than develop tire correction factors.

4. More data should be obtained to better define the difference between true steady-state road-load curves and the Clayton dynamometer steady-state load curve.

5. Additional data should be obtained to define the difference between the Clayton dynamometer steady-state load curve and the Clayton dynamometer transient load curve (i.e., define the hysteresis effect).

References

(1) W.B. Crum, "Road and Dynamometer Tire Power Dissipation," SAE Paper 750955, October, 1975.

(2) B. Simpson, "Improving the Measurement of Chassis Dynamometer Fuel Economy," SAE Paper 750002, February, 1975.

(3) Clayton Manufacturing Co., "Requirements and Consideration of Test Methods and Equipment for Passenger Vehicle Fuel Economy Measurement," August, 1975.

(4) "Variables Affecting 1975 Light Duty Truck Exhaust Emissions and Fuel Economy", EPA contract No. 68-03-2196, Task No.1, 1976.

Table I. Prospective Dynamometer Marketers which were Contacted

Company	Manufacture or Market Light-duty vehicle Chassis Dynamometers?	Make or Model	PAU (1) Type	Variable Speed-load Simulation	Variable Inertia Simulation
Ostradyne, Inc.	No				
Automotive Environmental Systems	No				
Go-Power Systems	No				
AW Dynamometer, Inc.	No				
Greening Associates, Inc.	No				
Hartzell Corp.	No				
Maxwell Dynamometer	Yes		DC Motor	Yes	No
Taylor Dynamometer and Machine Co.,Inc.	Yes	TAY 400	Hydrokinetic	Yes	No
Mid-West Dynamometer & Engineering Co.	No				
Inductor, Inc.	Yes		Disc Brake	No	No
Cox Instrument	No				
Burke E. Porter Machinery Co.	Yes	1059	DC Motor	Yes	Yes
		1098	DC MG (2)	Yes	Yes
KAHN Industries	No				
Bear Manufacturing Corp.	Yes	46-150	Prony Brake	No	No
Marquette	Yes	Bear 46-150	Prony Brake	No	No
Autoscan	Yes	8100	Disc Brake	No	No
Sun Electric Corp.	Yes	RAM 937	Eddy Current	Yes	Yes
Clayton Manufacturing	Yes	ECE-50	Hydrokinetic	Yes	Yes
		CTE-50	Hydrokinetic	Yes	Yes
Labeco	Yes		Eddy Current	Yes	Yes

(1) Power Absorption Unit
(2) Motor- Generator set

Table II. Dynamometer Specifications on Units Capable of Complying with Exhaust Emission Test Requirements

Marketer	Clayton		Sun Electric	Burke E. Porter		Labeco
Model No.	ECE-50	CTE-50	RAM-937	1059	1098	Modified 100-75P
Rolls						
Configuration	Double	Double	Double[1]	Double	Single	Single
Diameter, In.	8.65	8.65	21.63	9.50	48	40.19
Centerline Distance, In.	17.25	20.00	21.81	20.5		
Vehicle Treadwidth, In.	0-78	20-107	40-75	50-84	50-84	30-80
PAU						
Type	Hydrokinetic	Hydrokinetic	Eddy Current	DC Motor	DC MG Set	Eddy Current
Torque Control	$T = AV^{1.83}$	$T = AV^{1.83}$	$T = B + CV^2$	$T = B+DV+CV^2+E\frac{dV}{dt}$	$T = B+DV+CV^2+E\frac{dV}{dt}$	$T = B+CV^d$
Max Power, hp	50	50	250	30 Road load	200	100
Inertia						
Type	Direct-Drive Flywheels	Direct-Drive Flywheels	Belt-Drive Flywheels	Electrical or Flywheels	Electrical	Direct-Drive Flywheels
Range, Lb.	1,750-5,500	1,750-5,500	1,500-5,500	1,500-6,000	1,500-6,000	2,000-9,750
Increments, Lb.	Per FTP[2]	Per FTP[2]	Per FTP[2]	250[3] or per FTP[2,4]	Continuous	250

(1) Front and rear roll connected by a cog belt.
(2) 1750, 2000, 2250, 2750, 3000, 3500, 4000, 4500, 5000, 5500.
(3) For electric simulation.
(4) For flywheels.

Table III. Comparison of Clayton Dynamometer Road-Load Curve and True Road-Load Curve when 50 mph Power is Identical

	Dyno Power Minus Road Power, HP		% Difference in Dyno and Road Power	
Vehicle	20 mph	35 mph	20 mph	35 mph
Corvette	-0.3	-0.1	-12	- 2
Firebird	-0.6	-0.7	-22	-11
Grand Prix	-0.3	-0.4	-12	- 5
Impala	-0.4	-0.3	-13	- 4
Nova	-1.0	-1.0	-33	-14
Vega	-0.3	-0.4	-14	- 7
Eldorádo	-0.9	-0.7	-26	- 9
Riviera	-0.5	-0.5	-14	- 7
8-Car Average	-0.5	-0.5	-18	- 7
C-10 Pickup	+0.9	+0.8	+28	+10

Table IV. Comparison of Dynamometer Prices and Accuracies

MARKETER	CLAYTON		SUN ELECTRIC	BURKE E. PORTER		LABECO
MODEL	ECE-50	CTE-50	RAM-937	1059	1098	Modified 100-75P
<u>PRICES</u>						
Roller Ass. & PAU	9,725	11,325	25,000	53,000	105,000[1]	70,000[2]
Inertia Flywheels	10,600	10,600	14,300	9,757	NA[3]	Standard
Automatic Loading	2,275	2,275				
Site Prep. & Installation	8,000	8,000	9,000	9,000	23,000	23,000
Total	30,600	32,200	48,300	71,800	128,000[1]	93,000
<u>ACCURACY</u>						
<u>Steady-State Road Load</u>						
Duplication of speed-torque curves	± 30%	± 30%	± 3%	±3%	±3%	(± 3%)[4]
Unit to Unit Variability	± 4%	± 4%	(± 1%)[4]	± 1%	± 1%	(± 1%)[4]
Day to Day Variability	± 3%	± 3%	± 0.5%	± 0.5%	± 0.5%	(± 0.5%)[4]
<u>Inertia</u>						
90% Response Time	Neg.[5]	Neg.[5]	Neg.[5]	0.5 sec	0.5 sec	Neg.[5]
Duplication of Vehicle Inertia	± 1%	± 1%	± 1%	± 5%	± 5%	± 1%
Day to Day Variability	Neg.[5]	Neg.[5]	Neg.[5]	± 1%	± 1%	Neg.[5]

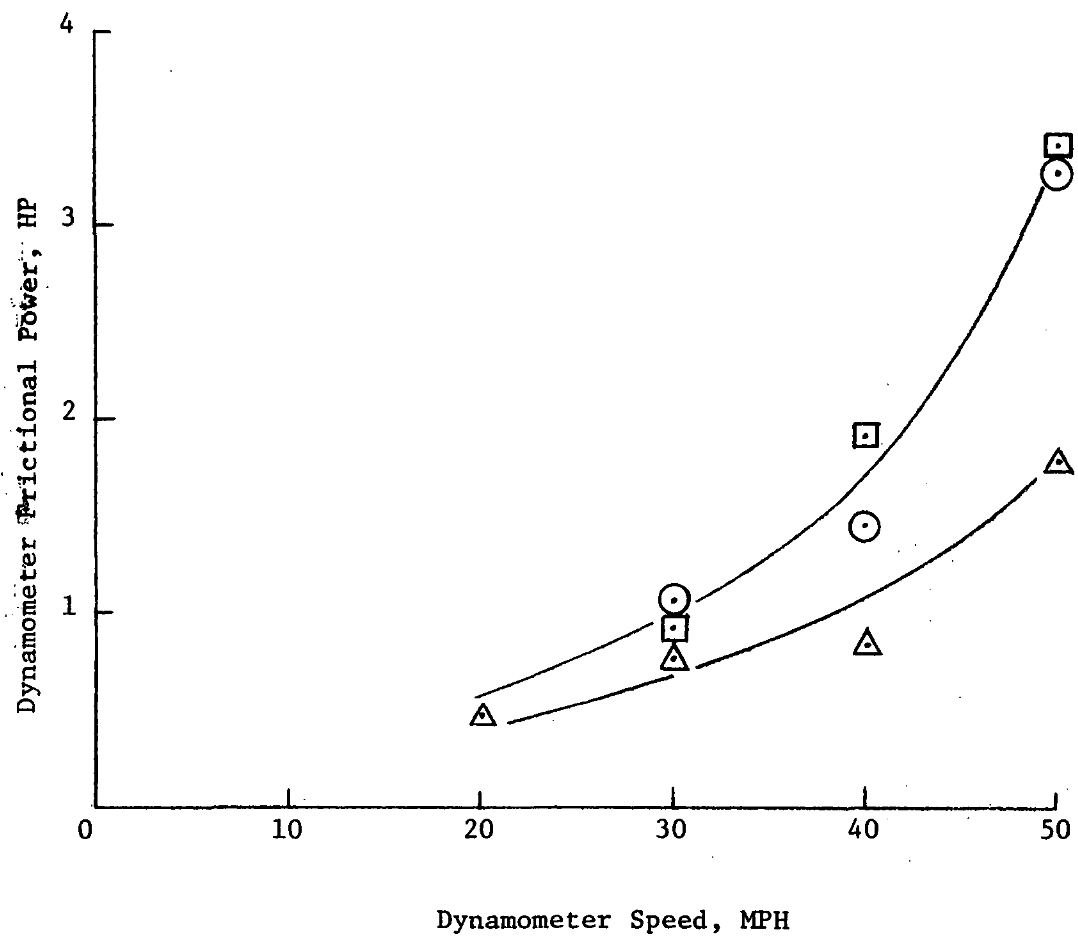
(1) Includes large centrifugal fan which simulates on-road cooling. Cost without fan would be reduced approximately $30,000.
(2) Includes large centrifugal fan which simulates on-road cooling.
(3) Not Available
(4) Estimated
(5) Negligible

Figure 1. Effect of Dynamometer Speed on Frictional Power Absorption For one Clayton Dynamometer.

⊙ Inertia = 4500 lbs., 50 MPH Indicated HP = 9.0.

⊡ Inertia = 4500 lbs., 50 MPH Indicated HP = 11.0.

△ Inertia = 2000 lbs., 50 MPH Indicated HP = 9.0.



APPENDIX

Table A-I. Vehicle Power Requirements for Road Load and Inertia Simulation

Driving Cycle	UDDS		HFE	
Inertia Weight	4000		4000	
Actual HP @ 50 mph	12.0		12.0	
Indicated HP @ 50 mph	10.0	8.0	10.0	8.0
Frictional HP @ 50 mph	2.0	4.0	2.0	4.0
Average vehicle HP into road load	1.72	2.00	9.64	9.55
Average vehicle HP into inertia	3.66	3.66	3.66	3.66
Average vehicle HP	5.38	5.66	13.30	13.21

Assumption: Road Load Power = $KV + AV^{2.87}$, and K = constant for a particular frictional hp.

Figure A-1. Comparison of Tire Rolling Resistance Between a Clayton and a Sun Dynamometer.