**Theory chassis dyno operations**

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# Determination of road load and dyno coefficients

## Determination of road load coefficients

All investigations in this guideline are based on newton’s fundamental laws of motion.

It is all about the conservation of translational and angular momentum.

Furthermore, it is the attempt to utilize the chassis dyno as testing environment by achieving the same load conditions for the propulsion system as when operating the vehicle under real world conditions.

To achieve that, we need to address the differences occurring between both testing environments.

When operating a vehicle in real world environment the energy provided from fossile fuel or electric energy is converted into

* finally heat by all kinds of friction (air drag, tires, bearings)
* translational and angular momentum (accelerating the vehicle mass and all of their rotating components)
* potential energy when driving uphill.

The last two can be recuperated highly efficient in electric vehicles, but friction is irreversible.

So, when operating a vehicle on a flat road (for simplification reasons, let’s keep slopes from now on out of considerations) the vehicle

* accelerates if the driving force provided from the propulsion system is **higher** than the sum of all frictional forces,
* remains in constant speed if the driving force **equals** all frictional forces and
* decelerates if the driving force provided from the propulsion system is **lower** than the sum of all frictional forces, recuperation or active braking.

Hence, when comparing the operation of the vehicle in real world and on the chassis dyno a practical method has to be found to simulate its differences on the chassis dyno to provide the proper forces on the wheels.

Unfortunately, all the different legislative regulations like ECE, EPA, JP, China etc. come up with different approaches, furthermore distinguishing between light duty, heavy duty, two or four wheel drive operations as well as single or dual axle chassis dynos with vehicle mass tables and so on. One gets lost very easily when starting with all those variants. So, let’s stick at the beginning to the physical basics. If they are understood well, applying legislative specifics becomes a nasty but important handwork which at the end has to be handled user friendly by the chassis dyno application software.

One common approach to transfer from road to chassis dyno is the so called coast down of a vehicle. You need to accelerate your vehicle on the road that is also specified by the regulations (max. slope, wind conditions, temperature etc.) to a certain constant speed e.g. 145 km/h and then changing to gear disengaged or neutral state.

From now on as described above the vehicle has no acting driving force from the propulsion system or brakes anymore and decelerates only by all the frictional forces. Hence, the idea is to model all those frictional forces from this experiment, where the vehicle velocity decreases over time as shown in Figure 1.



Figure 1 Vehicle speed of time during a coast down

The principle idea is to model the coast down curve by approximating all frictional forces by a vehicle speed depending polynomial of second order. The three resulting coefficients of the polynomial are the so called **road load coefficients A,B and C**.

To obtain those road load coefficients we need to derive the newtons law of motion, which is given in Equation 1

Equation 1

where

* *mE* is the effective vehicle mass of the vehicle (refer 1.2)
* ***a*** is the acceleration
* *FFriction* is the sum of all frictional forces (tires, air drag etc.).

Shall be approximated by a polynomial of second order, refer to Equation 2.

Equation 2

Merging Equation 2 in Equation 1 and considering ***a*** as deceleration has a negative sign leads to Equation 3 in scalar form

Equation 3

As the velocity is given in descrete time steps from a data logger the acceleration “a” has to be calculated by its first derivative in time (refer Equation 4) with numerical methods (refer Equation 5).

Equation 4

Equation 5

In Equation 4 and Equation 5

* *t* is the time,
* the values indexed by *t* are the values of the current time step whereas
* the values indexed by *t-1* are the values of the previous time step.

For better understanding, the previously shown mathematical approach is shown graphically in Figure 2. For each point on the curve of the vehicle speed the acceleration can be calculated and can be plotted depending on the vehicle speed on that point.



Slope of vehicle speed is the vehicles acceleration, which leads to the frictional force when multiplied with the vehicle mass

Figure 2 Determination of road load as a function of vehicle speed.

Finally, we just need to approximate the gained function of *FFriction* over *vehicle speed* by a polynomial of second order with provided functions e.g. in Excel, MatLab or Concerto to obtain the polynomial coefficients A,B and C.

## Determination of the effective test track mass

One has to distinguish between the common mistake naming as a so called inertia force, and as the frictional force. is the time rate of change of the momentum caused by the sum of all acting forces of that body.

Hence, is a force, which is by means of a practical method approximated by a polynomial of second order only depending on the vehicle speed as shown in the previous subchapter and must not be depending on any mass calculations or modified accelerations.

But the derivation of the effective vehicle mass needs to be done now in more detail.

For simplification a 4WD vehicle which will finally be tested on a 4WD chassis dyno will be taken into account.

As stated in the previous chapter, the vehicle

* accelerates if the driving force provided from the propulsion system is **higher** than the sum of all frictional forces,
* remains in constant speed if the driving force **equals** all frictional forces and
* decelerates if the driving force provided from the propulsion system is **lower** than the sum of all frictional forces, recuperation or active braking.

As the overall vehicle mass contains also all rotating parts, they have a rotational inertia at the beginning of the coast down procedure on the road as well that contributes to the deceleration.

The impact of all rotational parts like tires and rotating parts of the transmission driveline has to be taken into account when deriving the dyno coefficients.

Let’s analyze two cases, where the overall vehicle mass as well as all frictional losses are constant in both cases, but in one case big sized tires with huge mass moment of inertia are used. Both coast downs are shown in Figure 3.



Figure 3 Impact of rotational inertia on road load

Although the vehicle mass is the same in both cases the deceleration decreases with the huge tires as the translational momentum of the whole vehicle *and* the angular momentum of its rotating parts is reduced by frictional forces.

Hence, if the rotational masses are not taken into account, either the coast down curve implies a higher vehicle mass or it implies lower frictional forces, which would be in both cases the wrong conclusions.

By utilizing the laws of energy conservation in combination of kinematic relationships rotating masses can be transformed into a translational mass. This simplifies all calculations, which is done in all regulations. Therefore, the basic equations will be derived. To find the equivalent translational mass of the rotating masses we need to set up the equation for energy conservation. A not driven vehicle has two “sources” of energy, the so called kinetic energies that is conserved when no external forces are acting, otherwise those energies are transferred to potential energy like vertical displacement, recuperated if possible or dissipated by frictional losses, expressed by Equation 6.

Equation 6

Where is the mass moment of inertia, is the translational vehicle mass, is the angular speed of the wheels (and all connecting rotating parts), m is the overall vehicle mass, v is the translational vehicle speed and is the potential energy of the vehicle.

By means of simplification we assume no slip between the wheels and the ground. Therefore, we can apply the kinematic relation in Equation 7.

Equation 7

Where is the so called effective rolling radius. Inserting Equation 7 in Equation 6 leads to Equation 11 Equation 8

Equation 8

As can expressed by a point mass at any rotating point with the unit [kgm2] we preferably chose the radius being the effective rolling radius to get an equivalent translational mass having the same impact on the coast down as a translational mass , resulting in Equation 9

Equation 9

Substitution of in Equation 8 leads to Equation 10.

Equation 10

And finally by simplifying to

Where is the effective vehicle mass .

The legislative regulations either allow to add the calculated equivalent rotational masses or as a percentage of the overall vehicle mass. Taking the percentage approach in Equation 3 into account we get Equation 11

or

or

Equation 11

## Determination of dyno coefficients

The aim by utilizing the chassis dyno as testing environment is, that the load for propulsion system is equal as when operated under real world conditions.

For that we shall emphasize the major differences between both testing environments, where again the chassis dyno is operated in 4WD mode.

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
|  |  |  | ***Air Drag*** | **Friction Tires** | **Friction Transmission** |
| **Real World** | Yes | Yes | Complete | Complete | Yes |
| **Chassis Dyno** | No | Yes | No | Slightly different | Yes |

Table 1

To derive the dyno coefficients the momentum equation has to be formulated according to Equation 11, resulting in Equation 12 with new indices in the acceleration and the frictional forces

Equation 12

is now defined by the actual sum of all frictional forces acting on the vehicle when mounted on the dyno added by simulated force of the dyno, like shown in

Equation 13

Equation 12 becomes with Equation 13 to Equation 14

Equation 14

Which shall be finally with the so called road load adaption be equalized with Equation 1 to ensure the same load for the propulsion system.

shall be equal with

**Important:** in Equation 14 needs to be emphasized with referring to Table 1 that only and need to be provided by the dyno controller, for is provided already by the actual rotating masses of the vehicle.

## Determination of dyno coefficients based on measured forces

According to chapter 1.1 the frictional losses during the coast down procedure on a flat road are approximated by a polynomial of second order where its coefficients are described as the road load coefficients A,B and C.

By operating the vehicle now on a chassis dyno leads now to the following differences compared to the road:

* No translatoric masses
* No air drag
* Tire friction changes due to different contact situation between roller and tire (radius, material etc.)

Generally, a chassis dyno is a very accurate speed and force controller and can therefore be used as a force measurement device.

One of the most important outputs is the so called F\_KFZ (=in previous Equations), which is the net force acting on the contact points between the rollers and the tires, refer to Figure 4, where

* F-LC is the measured force with a load cell
* r is the distance of roller center to center of load cell and
* R is the roller diameter.

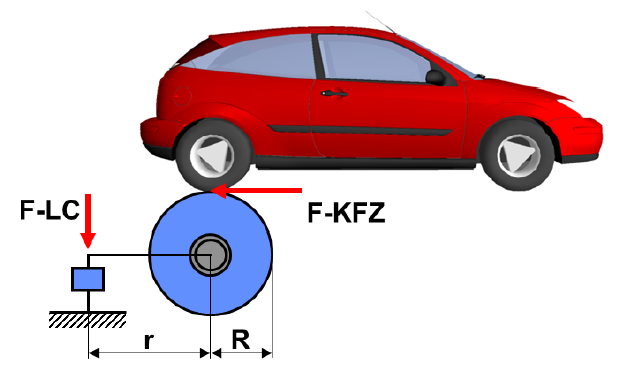


Figure 4 Definition of FKFZ

As F\_KFZ is the net force on the contact points between the rollers and the tires the reading of the force of the load cell has to be corrected in addition by

* all frictional losses (parasitic losses) and
* the angular inertia of the rollers itself.

For this the provided correction function of the chassis dyno automation system have to be performed regularly.

A very easy verification of F\_KFZ is by simply driving a test cycle with the dyno without a mounted vehicle, where F\_KFZ has to be close to zero along the whole test.

As there are much lower frictional losses of the vehicle when operated on the chassis dyno we need to find out the residual missing frictional forces that shall be simulated by the dynos.

Hence, if the coast down procedure is repeated on the chassis dyno without driving forces from the propulsion system the measured force F\_KFZ at the contact points between the rollers and the tires are the actual frictional forces of the vehicle, refer to Figure 5, second subplot.



Figure 5 Measured FKFZ during road load adaption on the chassis dyno

If F\_KFZ measured during the coast down on the chassis dyno is substracted from FFriction, which is expressed by the polynomial with the road load coefficients the residual force FVehFriction can be obtained, that again have to be approximated by a polynomial of second order, as shown in Figure 6.



**Polynomial second order**

**F\_KFZ = measured force during coast down on cassis dyno**

**FFriction = A + B\*v + C\*v2**

FDyno

**missing frictional forces that have to be applied by the dynos.**

Figure 6 Obtaining dyno coefficients

Hence, the force to be simulated by the dyno results in Equation 15.

Equation 15

As *FDYNO* is a function of vehicle speed its approximation by a polynomial of second order leads to the so called **dyno coefficients F1, F2** and **F3** as shown in