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COMPUTER AIDED VEHICLE DYNAMICAL CALCULUS

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Abstract

This estimating method and the computer program, based on this method, aim at making simpler and faster of the lengthy dinamic estimate by the vehicles. The accordance of the estimate with the real facts is depend considerably on the exactness of the input data. The publication demonstrates the method of the insertion of data, the estimating process and the diagrammatic results. The demonstrated method can be used for both mechanical and hydraulic gear.

Keywords: vehicle dynamics, computer simulation.

1. Vehicle Dynamics as a Subject of Study

During the motion the vehicle is subjected to several forces which influence its behaviour. In the vehicle dynamical calculations one determines the characteristic parameters and produces a model describing the behaviour of the vehicle. Simplifying the examination of the model we distinguish between the lateral motion, the longitudinal motion and the effects generating the vibration of the vehicle. The side forces have an influence on the stability of a vehicle. These characteristic features are the following: wind sensitivity (side slip), turning behaviour and stability, roll axis and roll centres. Examining the vertical vibration of a vehicle we are interested in the ride comfort which has direct connection to the pitching during breaking and driving, suspension, state of the dampers. In this lecture we are going to deal with the longitudinal dynamics of a vehicle. In this case the most relevant parameters are the engine power and engine torque, acceleration, 'slope overcome' abilities. These parameters are competent only if the function of the vehicle is known.

2. 'Domino' Principle

The vehicles available in the market are not completely different, several parts of them can be the same, though not all car manufacturers have their own products of some parts. Other important viewpoint is that the customer may have different claims to be fulfilled, thus the manufacturer should be able to produce the required car as well. First in case of duty-vehicles is important to have several parts and to build the required vehicle on a basic model. But this vehicle has to operate satisfactorily and if it is unique, the manufacturer should do some vehicle dynamical calculations.

3. Vehicle Dynamical Parameters

3.1 Engine

First we need a chassis on which the body and other parts will be built. After that we must choose an engine for the vehicle. The torque driving vehicle is generated by the engine. That is why at starting of a vehicle the torque produced by the engine at low RPM is very important. Other important parameters are the maximum power at high speed riding and the available RPM. Summarizing, the two most important characteristics of the engine from the longitudinal dynamics of a vehicle is: $P_{\max}$, $n_{P_{\max}}$ and $M_{\max}$, $n_{M_{\max}}$. Other operational points than mentioned can be obtained from the exactly known characteristic of the engine, or missing it one can approximate on the basis of the maximum torque and moment fitting a given shaped curve.

$$P = P_{\max}\left[A_1\,\frac{n}{n_p} + A_2\left(\frac{n}{n_p}\right)^2 - \left(\frac{n}{n_p}\right)^3\right].$$

Engine type	A_1	A_2
Otto:	1	1
Direct injection:	1.5	1.5
Pre-combustion chamber:	0.6	1.4
Whirl-chamber:	0.7	1.3

Using the well known formulae $P = M\,w$ from the power and RPM data the torque can be calculated. We note that during the vehicle dynamical calculations we always use maximum throttle position. Not the complete power of the engine can be used for the driving of the vehicle, a part of it dissipated and another part is necessary for the driving of several units (e.g. air conditioning system, generator, water pump, etc.). From the viewpoint of vehicle dynamics only the net power is important, thus one has to decrease the gross engine power according to the standards.

3.2 Gear-box

As it is known, the characteristics of IC engine are not nearly ideal, which optimum is ensured by electric motors. That's why we have to use gear-box between the engine and driven axle. The covering curve of the characteristics at several gears approaches to the ideal one. The hydrodynamical gear gives better approximation than the mechanical one especially at lower revolutions. Disadvantage of the hydrodynamical gear is the larger sizes, more complicated devices and the poor efficiency, but it can be automatized quite easily. Combining the hydrodynamical and mechanical transmissions we can eliminate the low efficiency.

3.3 Driven Axle — Final Transmission

The last part of the transmission is the transversal gear with the differential. The final transmission is calculated as a product of their transmission ratios. The final transmission plays a role when calculating the 5th gear, whose modification is smaller than 1.

3.4 Diameter of the Wheel

Torque appearing on the axle drives the vehicle via the wheels. The accelerating force at a given moment depends only on the diameter of the wheel. On a larger tyre the driving force can be smaller than which is necessary. The smaller diameter of wheel decreases the available speed of the car, but it ensures larger acceleration even on higher revolution. Further important viewpoints are the road conditions, terrain or highway, etc.

3.5 Riding Resistance

We understand all of the effects hindering the motion of the vehicle. Two most important of them are the rolling resistance and the aerial resistance.

3.5.1 Rolling Resistance

The rolling resistance arising from the elastic contact between the tyre and roadway (deformation of the tyre and frictional forces). On one hand the friction makes the driving possible but on the other hand hinders that at the same time. The rolling resistance is considered with a factor denoted by f which can be constant or it can be function of the speed of the vehicle. There is about 10% difference in the final speed.

$$\psi = f\,[1 + (0.006\,v)]^2 .$$

3.5.2 Aerial Resistance

Aerial resistance can be described by a well known shape factor c_w. Its value can be determined in the wind channel. The most important vehicle parameter is the area of the forehead part. Its role is contradictory: if it is large the c_ω becomes larger but there is enough room for the pessangers, and vice versa.

3.6 Losses

Up-to-now all the elements of the driving chain were considered as ideal one without any losses. Of course, it is not so. The inertial friction, the hydraulic efficiency of the hydromechanical gears can be seen in the following table:

$$k_\Sigma = 0.98^a \times 0.97^b \times 0.99^c ,$$

a — number of cog-wheel connections
b — number of conical cog-wheel connections
c — number of cardanlinks

3.7 Inertial Forces

The inertial forces are arising from the mass of the vehicle and from the rotating masses. This last part of inertial forces are calculated on the circumference of the wheels and can be considered.

$$F_T = \frac{G}{g}\frac{\mathrm{d}v}{\mathrm{d}t}\left[1 + \sigma_1 + \sigma_2 \frac{\mathrm{d}\omega_e}{\mathrm{d}\omega_w} k_\Sigma\right] ,$$

$$\sigma_1 = \frac{g}{G}\sum \frac{I_w}{r^2} , \qquad \sigma_2 = \frac{g}{G}\sum \frac{I_\epsilon}{r^2}\eta_\Sigma ,$$

ω_c — RPM of engine,
ω_w — RPM of wheel,
I_ϵ — Inertial force of engine,
I_w — Inertial force of wheel,
k_Σ — total transmission ratio,

h_Σ — total transmission efficiency ($k_\Sigma = k_0 k_i$),
k_0 — final transmission ratio,
k_i — transmission ratio of mechanical gear,
k_h — transmission ratio of hydromechanical gear,
ω_T — RPM of turbine.

4. Viewpoints of Determination of Vehicle Parameters

4.1 Equations of Motion to be Solved (Fig. 1)

The parameters of the vehicle can be determined by solving the equations of motion, which has a following form at a given scope:

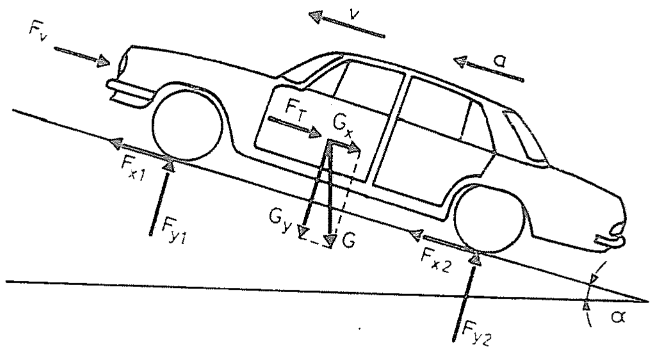

Fig. 1.

$$F_{x1} + F_{x2} - G_x - F_T - F_w = 0\,,$$
$$F_I = F_{x1} + F_{x2} - G_x\,,$$
$$F_I = F_r - F_\psi\,, \qquad F_T = F_{TG} + F_{T\epsilon} + F_{Tw}\,,$$
$$F_r - F_\psi - F_{TG} - F_{T\epsilon} - F_{Tw} - F_w = 0\,. \tag{1}$$

$$F_r = \frac{M_\epsilon k_s k_0 \eta_m}{r} \quad - \quad \text{traction force,} \tag{2}$$

$$F_\psi = G(f\,\cos\alpha + \sin\alpha) \quad - \quad \text{rolling resistance,} \tag{3}$$

$$F_{TG} = m\frac{\mathrm{d}v}{\mathrm{d}t} \quad - \quad \text{inertial force of mass,} \tag{4}$$

$$F_{T\epsilon} = \frac{I_\epsilon k_0 k_i \eta_{\text{mech}}}{r}\,\frac{\mathrm{d}\omega_\epsilon}{\mathrm{d}t} \quad - \quad \text{inertial force of engine,} \tag{5}$$

$$\frac{\mathrm{d}\omega_\epsilon}{\mathrm{d}t} = \frac{\mathrm{d}\omega_\epsilon}{\mathrm{d}\omega_w}\,\frac{\mathrm{d}v}{\mathrm{d}t}\,,$$

$$\frac{d\omega_\epsilon}{d\omega_w} = k_\Sigma \qquad - \quad \text{mechanical transmission,}$$

$$\frac{d\omega_\epsilon}{d\omega_w} = k_0\, k_h\, \frac{d\omega_\epsilon}{d\omega_T} \qquad - \quad \text{hydromechanical transmission,}$$

$$F_{Tw} = \frac{I_w}{r}\frac{d\omega_w}{dt} = \frac{I_w}{r^2}\frac{dv}{dt} \qquad - \quad \text{inertial force of wheels,} \tag{6}$$

$$F_w = c_w \frac{\rho}{2} A v^2 \qquad - \quad \text{aerial resistance.} \tag{7}$$

All inertial forces are calculated on the radius of wheels.

4.2 Engine Parameters

The main features of the engine characteristics are the maximum torque, the RPM of maximum torque, the maximum power and the related revolution. The power of the engine should be chosen according to the aim of the usage of the vehicle. The first step is to determine the power calculated on the mass of the vehicle (that is how many kilograms can be moved with 1 kW). For example the mass/power ratio under 14 kg/kW gives sporty characteristics to the car. In case of lorries the standard determines this ratio.

4.3 Transmission of the Driven Axle

Assuming that the maximal speed of the car is reached at the maximum engine power, the transmission ratio can be calculated using backwards algorithm:

$$k_0 = \frac{P_{\max}}{v_{\max}\, k_n},$$

r — rolling radius
$n_{P_{\max}}$ — RPM at maximum power
$v_{\max}$ — maximum speed of the vehicle
k_n — ratio of the last gear

4.4 Transmission Ratios of the Gearbox

The maximum ratio can be calculated according to the slope assuming zero acceleration and no air force due to the small speed or to the slope overcome ability of the vehicle. This should be checked by the maximum available tractive force on a given road.

$$k_I = \frac{G\,\psi_{\max}\,r}{M_e k_0 \eta_{\text{mech}}}, \qquad F_\psi \le F_\varphi = \sum G_1\,\varphi\,.$$

Assuming equivalent ratio between the gears:

$$k = \sqrt[n-1]{k_I^{n-i}}$$

4.5 Rolling Radius (Fig. 2)

The rolling radius is calculated according to the dimensions of the selected rim and tyres: $r = 0.5\,d + k_{HB}\,B$

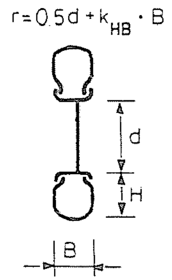

Fig. 2.

d — diameter of the rim,
k_{HB} — height/width,
B — width of the tyre.

4.6 Hydromechanical Gear

The hydromechanical gear mainly contains hydraulic cluth which has mechanical part improving the hydraulic efficiency in some ranges. The torque modification is achieved between the pump and turbine wheels. In order to use the design procedure applied in case of mechanical gear we have to know the torque modification in the hydraulic part of the gear. The total torque modification is given as the product of the ratios of the mechanical and hydraulical part of the gear.

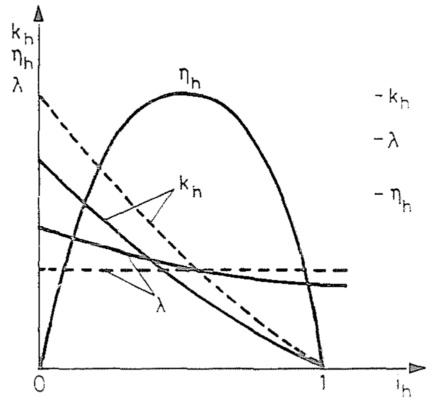

Fig. 3.

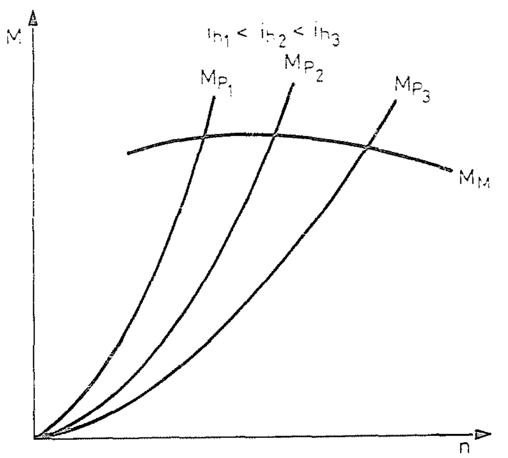

Fig. 4.

In case of hydromechanical gear we have to know the input characteristics in dimensionless form (*Fig. 3*). This characteristic is given by the manufacturer determined via measurement. The main features of the hydromechanical gear:

i_h — hydraulic ratio – revolution,
k_h — hydraulic transmission – torque,
η_h — hydraulic efficiency,
λ — dimension coefficient.

The torque of the pump: $M_p = \rho g \lambda D^5 n_p^2$

ρ — density of the liquid,
D — diameter of the gear,
n_p — revolution of the pump.

In a function of the RPM of the engine the dimension coefficient can be obtained from the diagram at a given hydraulic ratio and some new curves are obtained. The operational point of the engine and the pump is given by the intersection of two curves. At several hydraulic ratios the operational points can be obtained. This diagram is called a 'broom diagram' (*Fig. 4*). Having the torque of the turbine can be determined from the dimensionless characteristics.

4.6.1 Types of Hydromechanical Gears (Fig. 5)

It can be seen that after $k_H = 1$ $M_T < M_{sz}$ and the efficiency will be much worse. To eliminate this effect one can use the following solutions:

- directly switchable, the hydraulics is disconnected,
- complex switchable, the gear becomes free-running and the gear operates in clutch mode.

The diameter of the gear can be calculated backward from the maximum necessary torque at the starting.

$$D = \sqrt[5]{\frac{M_{p\ ih=0}}{\rho g \lambda_{ih=0} n_{p\ ih=0}}} .$$

Approximately the revolution of the pump is satisfactory in the range: stiff: $(0.3 \div 0.45)\, n_{P_{\max}}$, rigid: $(0.5 \div 0.75)\, n_{P_{\max}}$.

Afterward the ratios of the additional mechanical gear are calculated:

$$k_m = \frac{k_\Sigma}{k_h}, \qquad k_{mI} = \frac{k_I}{k_{h\,\eta=80\%}} .$$

The remaining ratios are calculated according to the mechanical gears.

The efficiency of the hydraulic gear at the starting is zero and it becomes again zero at the slip equal to 1. To bypass this unfavourable case we can use the power-ramified gear. The input power is divided into two parts by some planetary gear: The part of it goes through purely mechanical gear to the output shaft, and the remaining part is passed through hydraulic gear (*Fig. 6*).The advantage of this solution is the following:

- growing efficiency at the starting
- there are no wearing elements

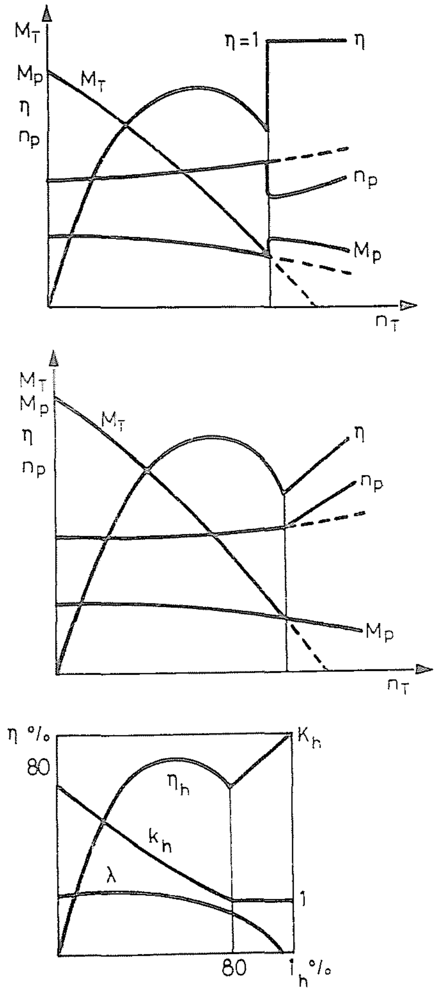

Fig. 5.

– the fuel consumption is improved
– it can be automatized.

The ratio of the planetary gear can be determined as follows: the 2nd gear is purely mechanical and only the 1st modifies the RPM. On the basis of the known i_0 the ratio of the additive planetary gear can be calculated:

$$\text{2nd gear:} \quad i_0 = 1 - k_{II} \,,$$

$$\text{1st gear:} \quad i_{mII} = \frac{i_0 \quad k_{\max}}{k_I - 1 + i_0} \,.$$

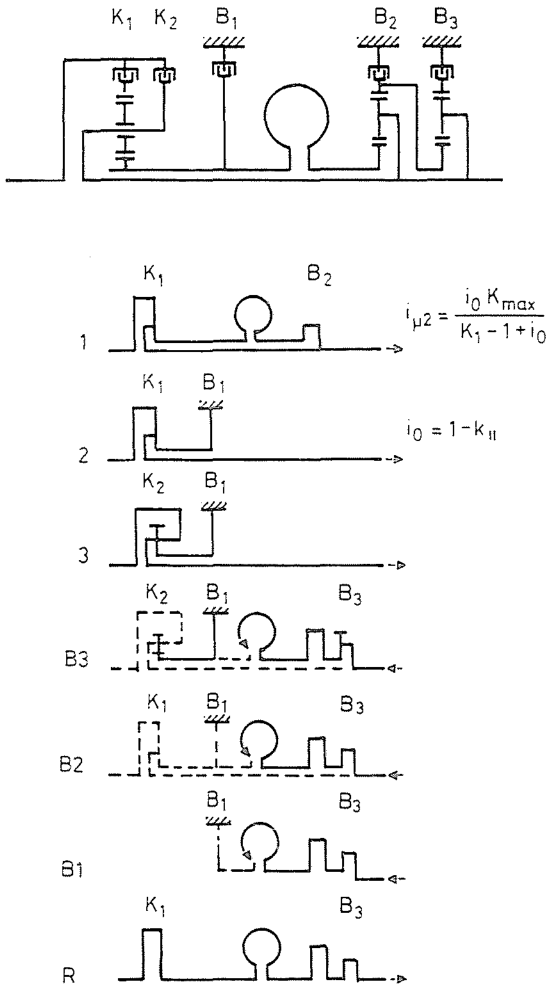

Fig. 6.

In order to use the design procedure of the classical hydromechanical gear, we can handle that as a virtual hydromechanical gear in the 1st lever. The input characteristics of the virtual gear can be determined from the ratios as follows:

$$i_v = \frac{i_h i_m}{i_0 + (1 - i_0) i_m} \,,$$

$$k_v = \frac{k_h \;\, i_0}{i_m} - i_0 + 1 \,,$$

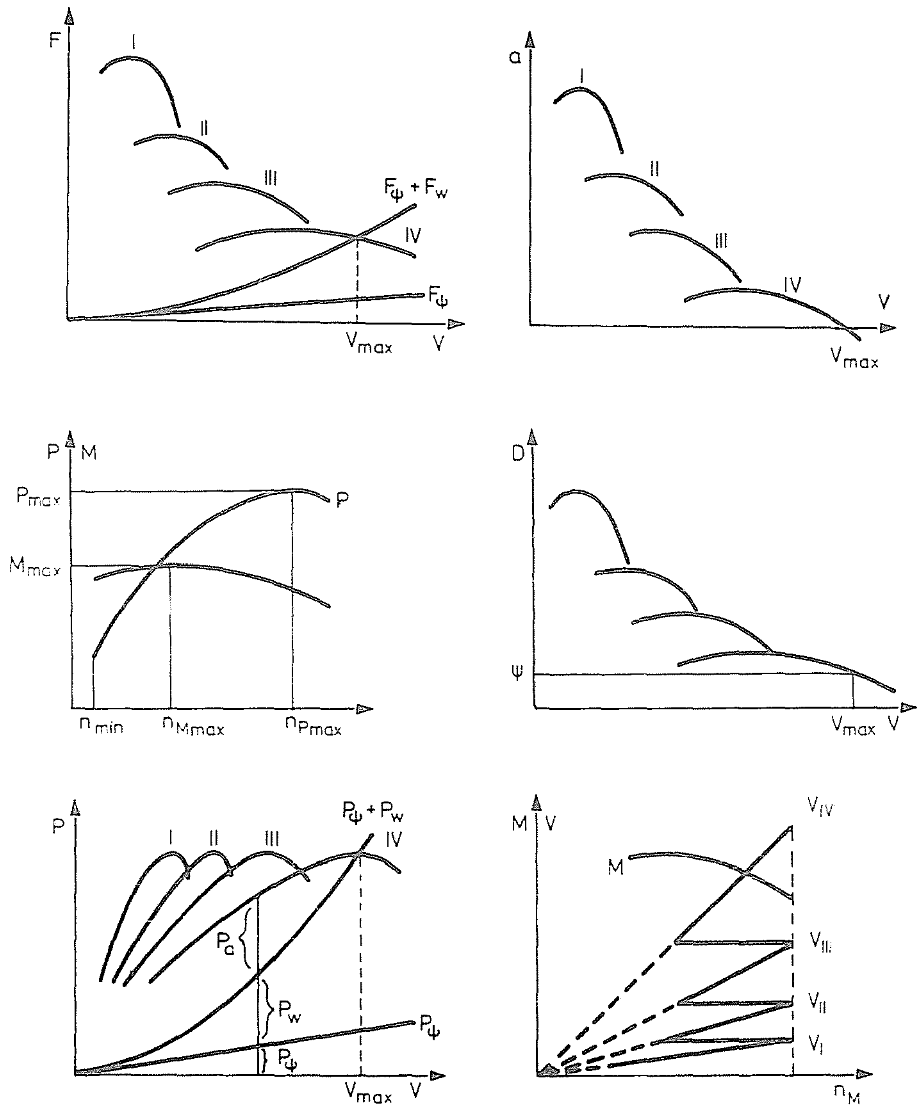

Fig. 7.

$$\lambda_v = -\frac{1}{i_0^3}\lambda_{sz}\left[1-(i_0-1)i_v\right]^2 ,$$

$$\eta_v = i_v \quad k_v .$$

The remaining calculations are calculated according to the hydromechanical gear (*Fig. 7*).