UNIVERSAL MECHANISM 10

universal mechanism

User`s manual

Driveline Modeling

2024

Contents

1. UM Module for driveline modeling

1.1. General information

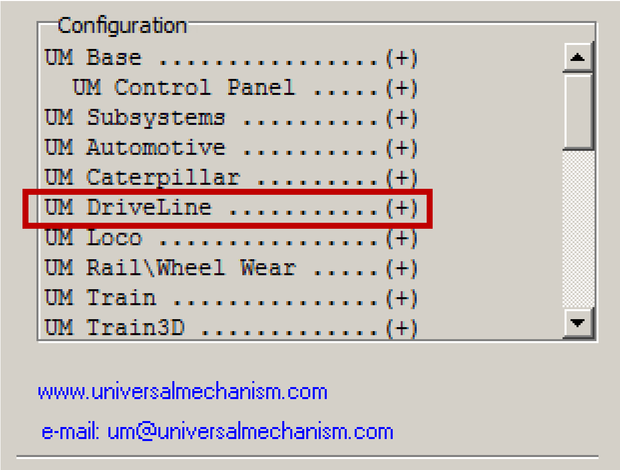


Figure 1.1. **'About'** window. List of available modules

Program package Universal Mechanism includes a specialized module **UM Driveline**, which contains force elements for simulation of drivelines e.g. in road, tracked, rail vehicles and other mechanical systems, Figure 1.1. The following elements are available in the module:

- Mechanical rotation converter with a constant ratio;
- Simplified model of a planetary drive;
- Fluid coupling;
- Hydraulic torque converter;
- Hydrostatic drive;
- Differential;
- Internal combustion engine.

1.2. Driveline elements

Here we consider special force elements, which are included in the UM Driveline module. Force elements described in this section connect two or three shafts. Usually, each of the shafts must have one rotational degree of freedom relative to the third body.

1.2.1. Mathematical models of force elements

1.2.1.1. Mechanical rotation converter

The element is used for simplified modeling a mechanical rotation converter with given transmission ratio i_{12}. In particular, the force element models gear trains, e.g. a gearbox. The element couples two rotating shafts: input 1 and output 2.

Equations of motion of the shafts:

$$J_1\ddot{\varphi}_1 = -c(\varphi_1 - \varphi_2 i_{12}) - d(\omega_1 - w_2 i_{12}) + M_1,$$
$$J_2\ddot{\varphi}_2 = -c\eta i_{12}^2(\varphi_2 - \varphi_1 i_{21}) - d\eta i_{12}^2(\omega_2 - \omega_1 i_{21}) + M_2 =$$
$$-i_{12}\eta\big(-c(\varphi_1 - \varphi_2 i_{12}) - d(\omega_1 - \omega_2 i_{12})\big) + M_2,$$
$$\omega_1 = \dot{\varphi}_1, \omega_2 = \dot{\varphi}_2, i_{12} = i_{21}^{-1} = \omega_1/\omega_2$$

Here c (Nm/rad), d (Nms/rad) are the stiffness and damping constants reduced to the input shaft; M_1, M_2 are the external load torques; J_1, J_2 are the shaft moments of inertia; $0 < \eta \leq 1$ is the converter efficiency specifying the energy loss in the element.

Both shafts are connected with some third body by rotational joints with one degree of freedom. For instance, in the model of a car gearbox the first body is the input shaft and the second one is the output shaft; the shafts are connected to the car body by rotational joints. Rotation of the input shaft can be a prescribed time function.

Models:

{UM Data}\SAMPLES\LIBRARY\Driveline\MechConverter;

{UM Data}\SAMPLES\LIBRARY\Driveline\MechConverter 1t.

In the second example the input shaft rotation is an explicit time function.

Adding element to a model: Sect. 1.2.2.1. *"Data input for mechanical rotation converter"*, p. 1-16.

1.2.1.2. Fluid coupling

Let n_1, n_2 be the speeds of the input and output shafts (impeller and turbine), the transmission ratio is $i = n_2/n_1$. The torque transmitted by the coupling is

$$T = \lambda_M \rho n_1^2 D^5,$$

where

D is the impeller diameter,

ρ is the fluid density,

$\lambda_M(i)$ is the torque coefficient which value depends of the ratio i, Figure 1.2.

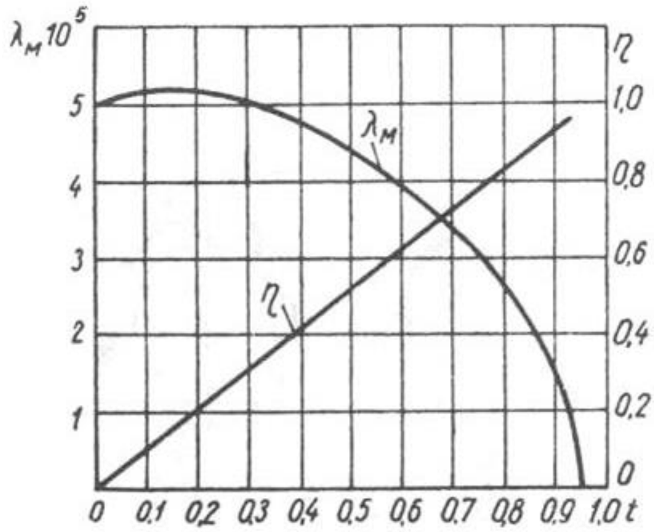


Figure 1.2. Example of fluid coupling characteristics (torque coefficient and efficiency η)

In UM the following model of the torque is used:

$$M = k_M \lambda_M \omega_1^2,$$

where ω_1 is the turbine (output) angular velocity in rad/s. The user specifies the plot $\lambda_M(i)$ and the factor k_M.

For instance, let the data units are T (Nм), ρ(kg/m^3), D(м), n (rpm). Taking into account that the speed in rpm and angular velocity are connected by the expression $n = \frac{30\omega}{\pi}$ we obtain

$$M = (30\,\omega_1/\pi)^2 \rho D^5 \lambda_M,$$

i.e.

$$k_M = (30/\pi)^2 \rho D^5 = 91.2\rho D^5$$

Model: {UM Data}\SAMPLES\LIBRARY\Driveline\Fluid Coupling.

Adding element to a model: 1.2.2.2. *"Data input for fluid coupling"*, p. 1-17.

Internet source: http://en.wikipedia.org/wiki/Fluid_coupling.

1.2.1.3. Hydraulic torque converter. Combined torque converter

Let n_1, n_2 be the speeds of the input and output shafts (impeller and turbine), the transmission ratio is $i = n_2/n_1$. The torque on the impeller (M_1) and turbine (M_2) are

$$M_1 = \lambda_{M1} \rho n_1^2 D^5,$$
$$M_2 = \lambda_{M2} \rho n_2^2 D^5$$

where

D is the converter diameter,

ρ is the fluid density,

$\lambda_{M1}, \lambda_{M2}$ are the torque coefficients which value depends of the ratio *i*.

In addition, the following characteristics are in use:

- Torque ratio

$$K = \frac{M_2}{M_1}$$

- Efficiency

$$\eta = \frac{M_2 n_2}{M_1 n_2} = Ki$$

Thus, the torque converter model can be described in several ways. Here are the equivalent converter description variants (the dependences of characteristics on *i*):

a) $\lambda_{M1}, \lambda_{M2}$
b) λ_{M1}, K
c) λ_{M2}, K
b) λ_{M1}, η
c) λ_{M2}, η

The element can be used for modeling a combined torque converter, i.e. a combination of the torque converter and the fluid coupling, Figure 1.3.

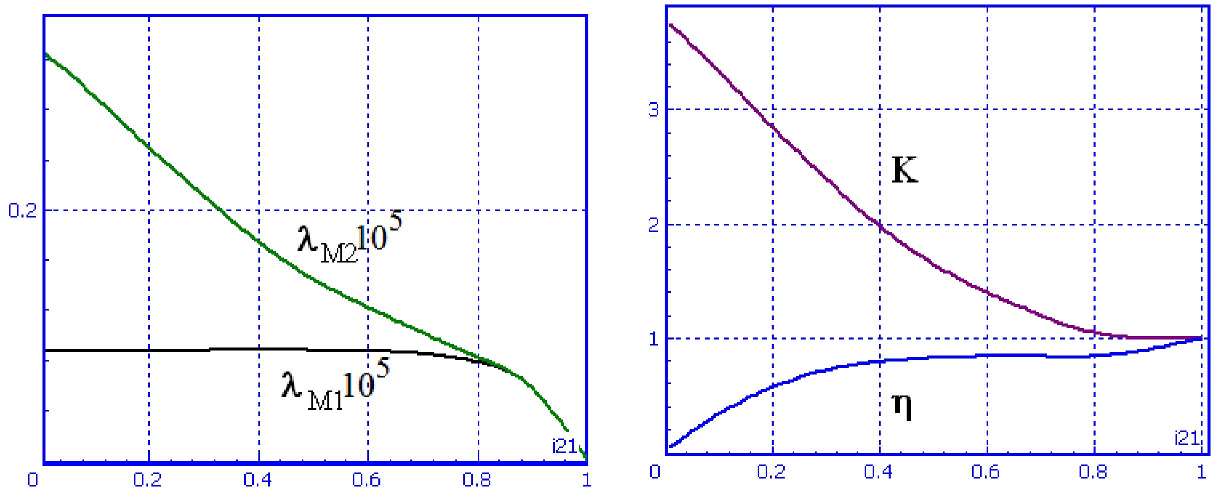


Figure 1.3. Example of combined torque converter characteristics

In UM the following model of the torques is used:

$$M_1 = k_M \lambda_{M1} \omega_1^2$$
$$M_2 = k_M \lambda_{M2} \omega_2^2,$$

where ω_1, ω_2 are the angular velocities of the impellor and turbine in rad/s. The user defines one of the pair of curves a)-c) as well as the factor k_M so that the torques are measured in Nm.

Model: {UM Data}\SAMPLES\LIBRARY\Driveline\Hydraulic Torque Converter.

Adding element to a model: 1.2.2.3. *"Data input for hydraulic torque converter"*, p. 1-18.

Internet source: http://en.wikipedia.org/wiki/Torque_converter.

1.2.1.4. Hydrostatic drive

One of possible design of the hydrostatic drive is shown in Figure 1.4:

1 – variable displacement reversible pump
2 – reversible motor
3 – pressure relief valves
4-7 – charge system
    4: charge pump
    5: relief valve
    6: check valves
    7: reservoir

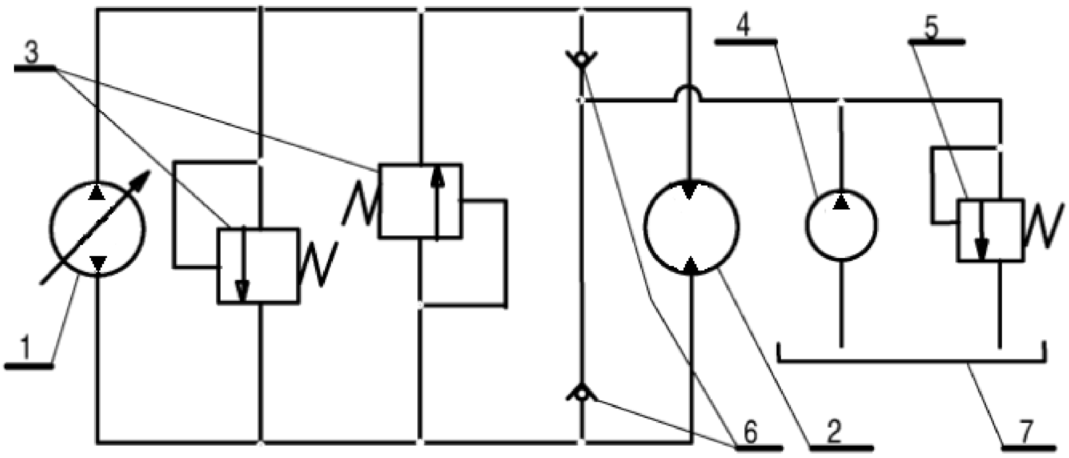


Figure 1.4. Hydrostatic drive

The following mathematical model of the hydrostatic drive is implemented:

$$\begin{cases} \dfrac{dp}{dt} = \dfrac{E}{V}\left(-rp + q_p e_p \omega_p - q_m e_m \omega_m\right), p < p_{u,\max} - p_l \\ \dfrac{dp}{dt} = 0, p > p_{\max} - p_l \end{cases}$$

$$M_p = -v_p \omega_p - q_m e_p p$$

$$M_m = -v_m \omega_m + q_m e_m p$$

$$p = p_u - p_l$$

Indices *p*, *m* correspond to the pump and motor respectively. The following designations are introduced:

p is the difference of pressures in the high p_u and low p_l sides; $p_{u,max}$ is the maximal pressure (relief pressure): when the pressure in the high side exceeds the maximal one, the relief valve opens; it is assumed that p_l is equal to the charge pressure;

ω_p, ω_m are the pump and motor angular velocities, rad/s;

M_p, M_m are the hydraulic torques acting on the pump and motor shafts, Nm;

E is the fluid bulk modulus, Pa;

V is the volume of the high or low sides, which assumes to be equal, m^3;

r is the total leakage coefficient, m^5/N/s;

q_p, q_m are the maximal pump and motor displacements, m^3;

$e_p, e_m \in [-1,1]$ are the displacement factors (control parameters) depending on the swash plate angle; $e_m = 1$ if the motor has a constant displacement;

υ_p, υ_m are the damping coefficients, Nms.

To estimate influence of the model parameters on dynamic properties of the hydraulic drive, consider a simple dynamic model consisting of two shafts connected by the drive.

$$J_p\dot{\omega}_p = M_1 - \upsilon_p\omega_p - q_p e_p p$$
$$J_m\dot{\omega}_m = -M_2 - \upsilon_m\omega_m + q_m e_m p \qquad (*)$$
$$\dot{p} = \frac{E}{V}\left(-rp + q_p e_p \omega_p - q_m e_m \omega_m\right)$$

Here J_p, J_m are the moments of inertia of the input and output shafts, M_1, M_2 are the external torques, M_2 is the output shaft load.

Removing external torques and setting zero values to parameters r, υ_p, υ_m we obtain the following system of linear differential equations:

$$J_p\dot{\omega}_p = -q_p e_p p$$
$$J_m\dot{\omega}_m = q_m e_m p$$
$$\dot{p} = \frac{E}{V}\left(q_p e_p \omega_p - q_m e_m \omega_m\right)$$

Differentiation of the third equation and substitution of the first two equations result in the equation of pressure oscillation

$$\ddot{p} = -\frac{E}{V}\left(\frac{q_p^2 e_p^2}{J_p} + \frac{q_m^2 e_m^2}{J_m}\right)p.$$

Thus, compressibility of the fluid introduce into the model the frequency

$$f_p = \frac{1}{2\pi}\sqrt{\frac{E}{V}\left(\frac{q_p^2 e_p^2}{J_p} + \frac{q_m^2 e_m^2}{J_m}\right)}\ \text{Hz}.$$

Consider a numeric example.

Let V=200 cm^3, $q_p = q_m = 16$ cm^3, $J_p = J_m = 1$ kgm^2, $e_p = e_m = 1$, $E = 1.5 * 10^9$ Pa. In this example the frequency is $f = 9.86$ Hz.

Let us simplify equations (*) assuming that the pump shaft rotates with a constant speed.

$$J_m\dot{\omega}_m = -M_2 - \upsilon_m\omega_m + q_m e_m p \qquad (**)$$
$$\dot{p} = \frac{E}{V}\left(-rp + q_p e_p \omega_p - q_m e_m \omega_m\right)$$

Consider a stationary solution of equations (**) by neglecting the leakages

$$0 = -M_2 + q_m e_m p$$
$$0 = -rp + q_p e_p \omega_p - q_m e_m \omega_m$$

Here are the stationary values of the pressure and output angular velocity:

$$p = \frac{M_2}{q_m e_m}$$

$$\omega_m = \frac{q_p e_p}{q_m e_m}\omega_p - \frac{rM_2}{(q_m e_m)^2}$$

Decrease of the output speed in comparison with the ideal value

$$\omega_m = \frac{q_p e_p}{q_m e_m}\omega_p$$

allows estimating the leakage r.

Finally, from the linear part of equations (**) we obtain the equation for computation of eigenvalues

$$\lambda^2 + \left(\frac{rE}{V} + \frac{v_m}{J_m}\right)\lambda + \frac{(q_m^2 e_m^2 + v_m r)E}{VJ_m} = 0.$$

The eigenvalues can be used for adjusting the damping parameter v_m.

Remark. Rotation of the input shaft can be set by an explicit time function.

Adding element to the model: Sect. 1.2.2.4. *"Data input for hydrostatic drive"*, p. 1-21.
Internet source: http://en.wikipedia.org/wiki/Hydraulic_drive_system

1.2.1.5. Epicyclic or planetary gearing

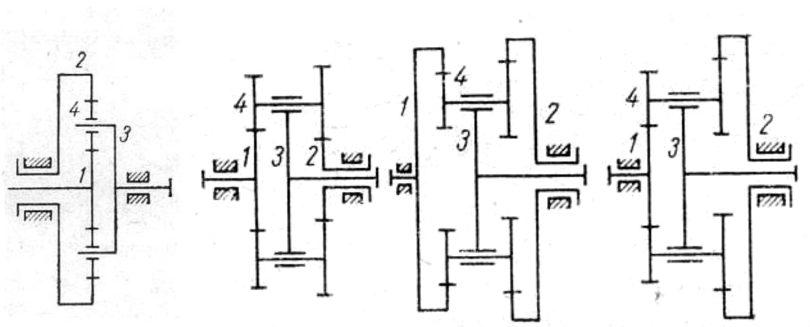


Figure 1.5. Planetary gearing schemes: sun gear 1, ring gear or *annulus* 2, carrier 3, planet gears 4

The force element is used for simplified modeling planetary gearing of four types shown in Figure 1.5. Simplification consists in neglecting the inertia properties of planet gears, i.e. it is assumed that kinetic energy of planets is small compared to energy of other parts and shafts. The model takes into account the compliance of teeth and bearings by introduction of an equivalent stiffness constant c [1]

$$c = \frac{1}{\dfrac{1}{c_1} + \dfrac{i_4^2}{c_2} + \dfrac{(1 - i_4)^2}{c_3}},$$

$$i_4 = \frac{r_1 - r_3}{r_2 - r_3}.$$

Here c_1, c_2, c_3 (Н/м) are the summarized spring constants in the planet/sun and planet/annulus toothing as well in planet gear bearings, r_1, r_2, r_3 are the radii of the sun, annulus gears and carrier. The summarized stiffness is equal to the stiffness of one toothing or bearing multiplied by the number of planet gears.

Consider a mathematical model of the element. Let us introduce the deflection

$$\Delta = r_1\Delta\varphi_1 - i_4 r_2 \Delta\varphi_2 - (1 - i_4) r_3 \Delta\varphi_3,$$
$$\Delta\varphi_i = \varphi_i(t) - \varphi_i(0), i = 1,2,3.$$

It can be shown that the moments acting from the planet gears to other gears are

$$M_1 = -c r_1 \Delta,$$
$$M_2 = c r_2 i_4 \Delta,$$
$$M_3 = c r_3 (1 - i_4)\Delta,$$

and the natural frequency of free vibration of the mechanism is equal to

$$\omega = \sqrt{c\left(\frac{r_1^2}{J_1} + \frac{r_2^2 i_4^2}{J_2} + \frac{r_3^2 (1 - i_4)^2}{J_3}\right)}.$$

Here J_1, J_2, J_3 are the moments of inertia of the sun gear, annulus and carrier. Experimental evaluation of this frequency allows estimating the value of stiffness c

$$c = \frac{\omega^2}{\left(\frac{r_1^2}{J_1} + \frac{r_2^2 i_4^2}{J_2} + \frac{r_3^2 (1 - i_4)^2}{J_3}\right)}.$$

By locking one or two elements of the planet gearing, the natural frequency could be obtained from the above one by removing the corresponding summand. For example, if the sum gear is locked, the frequency results in

$$\omega_1 = \sqrt{c\left(\frac{r_2^2 i_4^2}{J_2} + \frac{r_3^2 (1 - i_4)^2}{J_3}\right)}.$$

A damping constant μ is introduced to specify the desired damping ratio β.

$$M_1 = -c r_1 \Delta - \mu r_1 \dot{\Delta},$$
$$M_2 = c r_2 i_4 \Delta + \mu r_2 i_4 \dot{\Delta},$$
$$M_3 = c r_3 (1 - i_4)\Delta + \mu r_3 (1 - i_4)\dot{\Delta},$$
$$\mu = \frac{2\beta c}{\omega}.$$

Realistic values of the β parameter lie in the interval 0.01-0.1. The latter equations present the final mathematical model of the planet gearing in UM.

Remark 1. Stiffness constants c_1, c_2 must be reduced to the tangent to the base circle. If α is the angle between the contact action line and the tangent to the base circle the toothing stiffness constant c_{ig} must be multiplied by $cos^2 a$, i.e.

$c_i = nc_{ig}cos^2 a,$

where *n* is the number of planet gears.

Example: {UM Data}\SAMPLES\LIBRARY\Driveline\PlanetaryGearing.

Adding element to a model: 1.2.2.5. *"Data input for planetary gearing"*, p. 1-22.

1.2.1.6. Differential

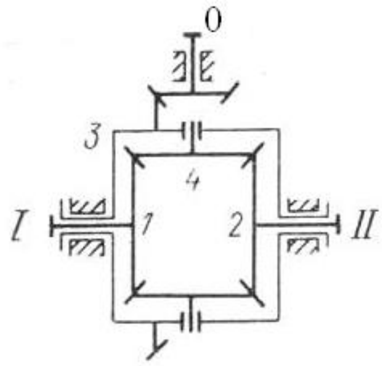


Figure 1.6. Differential: pinion (input) shaft 0, axle shaft (side) gears, 1,2, crown (ring) wheel 3, pinion gears 4

The differential is modeled as a force element connecting the input and axle shafts, Figure 1.6. Two models of differential are recommended to be used. The first model named Differential V1, includes all elements shown in Figure 1.6. The second model named Differential V2 excludes the input shaft 0 and the corresponding toothing input pinion – crown wheel, which must be included in a separate element e.g. the mechanical rotation converter.

1.2.1.6.1. Differential V1

Similar to the planetary gearing, Sect. 1.2.1.5. *"Epicyclic or planetary gearing"*, p. 1-10, the mathematical model of the differential is simplified by neglecting the inertia properties of the crown wheel, crown wheel housing and pinion gears, i.e. it is assumed that their contribution in kinetic energy is small. The model takes into account the compliance of side gears (1,2) and pinion teeth (4) as well as pinion (4) bearings by introduction of an equivalent stiffness constant c. As opposed to the planetary gearing, the expression for the equivalent stiffness is more complicated and it depends on the differential design. It is recommended an experimental evaluation if the stiffness, or computation based on the mechanism design.

Consider a mathematical model of the element. Let us introduce the deflection

$$\Delta = r_0\Delta\varphi_0 - \frac{r_3}{2}\Delta\varphi_1 - \frac{r_3}{2}\Delta\varphi_2,$$
$$\Delta\varphi_i = \varphi_i(t) - \varphi_i(0), i = 0,1,2.$$

It can be shown that the moments acting on the input and output shafts are

$$M_0 = -cr_0\Delta,$$
$$M_1 = c\frac{r_3}{2}\Delta,$$
$$M_2 = c\frac{r_3}{2}\Delta,$$

and the natural frequency of free vibration of the mechanism is equal to

$$\omega = \sqrt{c\left(\frac{r_0^2}{J_0} + \frac{r_3^2}{4J_1} + \frac{r_3^2}{4J_2}\right)}.$$

A damping constant value is computer by the given damping ratio like in the case of the planetary gearing.

By locking one or two elements of the planet gearing, the natural frequency could be obtained from the above one by removing the corresponding summand. The value of the equivalent stiffness can be obtained from the experimental measures frequency or from a static experiment. Let two of three shafts are locked and a definite torque M is applied to the third shaft. Then the stiffness can be evaluated by the shaft rotation angle. For instance, if both of the output shafts are fixed, and a torque M is applied to the input shaft, the equivalent stiffness is computed according to the formula

$$c = \frac{M}{r_0^2\Delta\varphi_0},$$

where $\Delta\varphi_0$ is the rotation angle of the input shaft.

If a field experiment is impossible, it is recommended to create a detailed model of the mechanism in UM in which all of the gears are rigid bodies connected by compliant force elements. After that the equivalent stiffness can be evaluated from the computer experiment.

1.2.1.6.2. Differential V2

This model is simpler than V1 because the input shaft 0 is excluded in the model. The input body is the crown wheel housing. Inertia properties of pinion gears are neglected. The model takes into account the compliance of side gear (1,2) and pinion (4) teeth and pinion bearings by introduction of an equivalent stiffness constant c

$$c = \frac{1}{\frac{1}{2c_b} + \frac{1}{4c_t}},$$

Here c_b is the pinion bearing stiffness reduced to the pinion plane, and c_t is the pinion/gear stiffness constant.

Consider a mathematical model of the element. Let index 0 now indicates the crown housing. Introduce the deflection

$$\Delta = r_s\left(\Delta\varphi_3 - \frac{1}{2}\Delta\varphi_1 - \frac{1}{2}\Delta\varphi_2\right),$$
$$\Delta\varphi_i = \varphi_i(t) - \varphi_i(0), i = 0,2,3,$$
$$r_1 = r_2 = r_s$$

Here r_s is the side gear radius.

It can be shown that the torques acting on the crown housing and axle shafts are

$$M_0 = -c r_s \Delta,$$
$$M_1 = c\frac{r_s}{2}\Delta,$$
$$M_2 = c\frac{r_s}{2}\Delta,$$

and the natural frequency of free vibration of the mechanism is equal to

$$\omega = \sqrt{c r_s^2\left(\frac{1}{J_0} + \frac{1}{4J_1} + \frac{1}{4J_2}\right)}.$$

A damping constant value is computer by the given damping ratio like in the case of the planetary gearing.

1.2.2. Description of driveline elements in UM Input program

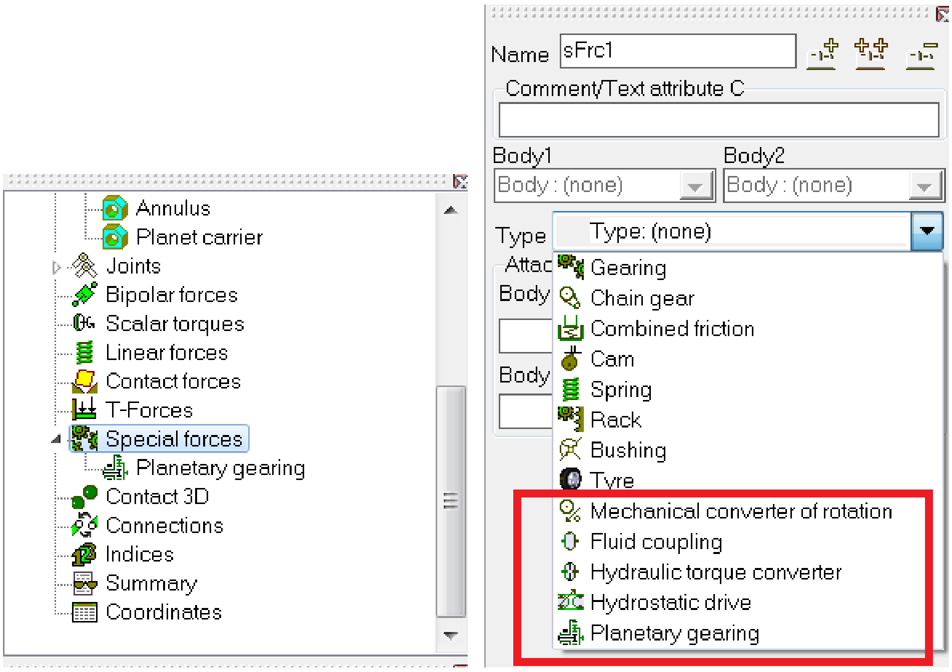


Figure 1.7. Driveline elements in the list of special forces

Driveline elements described in the previous section are available in the section of special forces of the **UM Input** program, Figure 1.7.

1.2.2.1. Data input for mechanical rotation converter

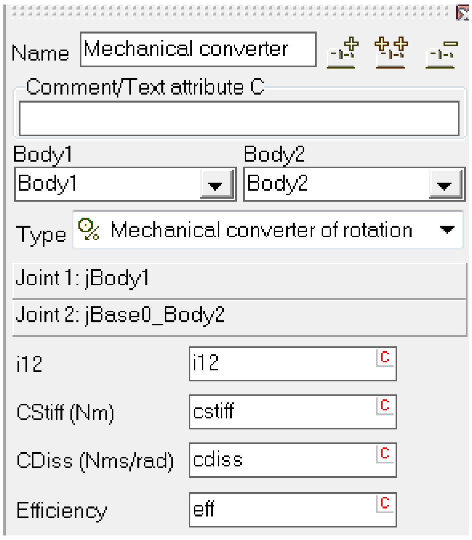


Figure 1.8. Parameters of mechanical rotation converter

Mathematical model of the element: Sect. 1.2.1.1. *"Mechanical rotation converter"*, p. 1-5.

Description of the element includes the following data, Figure 1.8.

- Assignment of two bodies: *Body1* as the input shaft and *Body2* as the output shaft. Each of the shafts must be connected to a third body by a rotational joint. A joint for the input shaft can describe the rotation as explicit time function. The program detects the corresponding joints and shows their names in the element window, *Joint* 1 and *Joint* 2 in Figure 1.8.
- Setting converter parameters (numerical values or parameterized constant expressions):
    - transmission ratio: $i_{12} = \frac{\omega_1}{\omega_2}$. If it is necessary to change the direction of rotation of the output , the sign of the ratio can be changed;
    - stiffness constant reduced to the input shaft, Nm;
    - damping constant reduced to the input shaft, Nms/rad;
    - element efficiency (energy loss in the element), $0 < \eta \leq 1$.

Models:
{UM Data}\SAMPLES\LIBRARY\Driveline\MechConverter;
{UM Data}\SAMPLES\LIBRARY\Driveline\MechConverter 1t.

1.2.2.2. Data input for fluid coupling

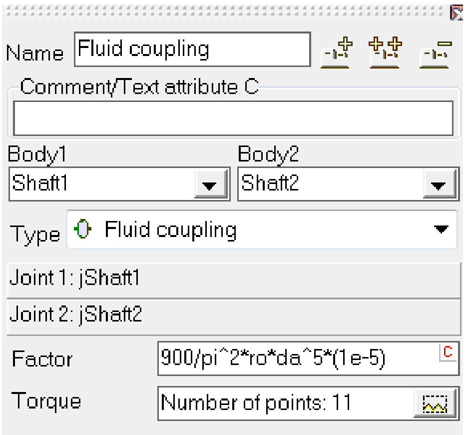


Figure 1.9. Parameters of fluid coupling

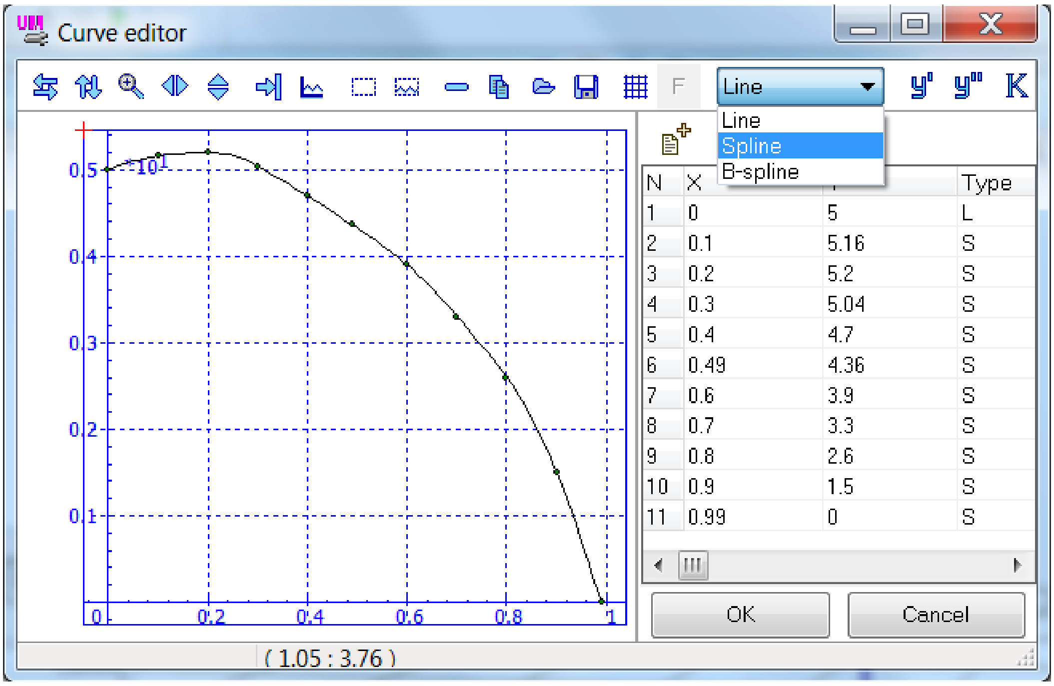


Figure 1.10. Fluid coupling characteristic: torque coefficient vs. speed ratio

Mathematical model of the element: Sect. 1.2.1.2. *"Fluid coupling"*, p. 1-5.

Description of the element includes the following data, Figure 1.9.

- Assignment of two bodies: *Body1* as the input shaft (impeller) and *Body2* as the output shaft (turbine). Each of the shafts must be connected to a third body by a rotational joint with one degree of freedom. The program detects the corresponding joints and shows their names in the element window, *Joint* 1 and *Joint* 2 in Figure 1.9.
- **Factor** k_M.

Description of the torque coefficient versus speed ratio $i = n_2/n_1$ in the curve editor, Figure 1.10 available by the button.

Model: {UM Data}\SAMPLES\LIBRARY\Driveline\Fluid Coupling.

1.2.2.3. Data input for hydraulic torque converter

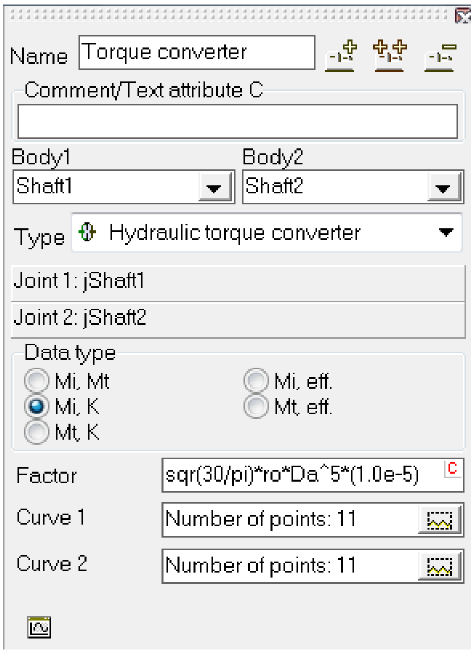


Figure 1.11. Parameters of torque converter

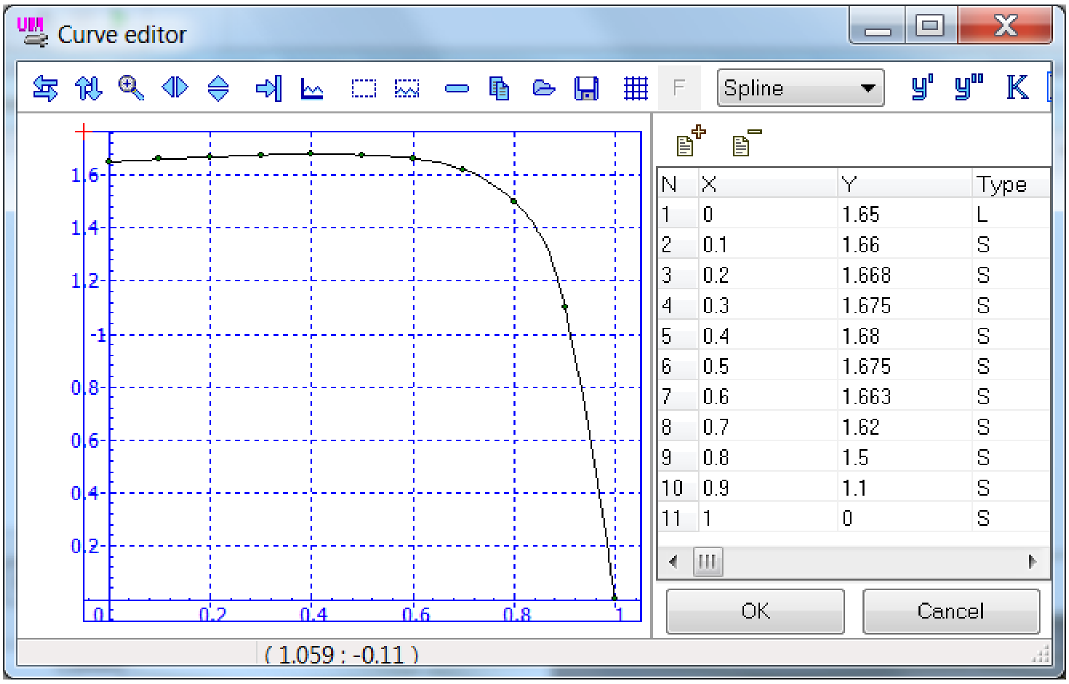


Figure 1.12. Example of the impeller torque coefficient vs. speed ratio

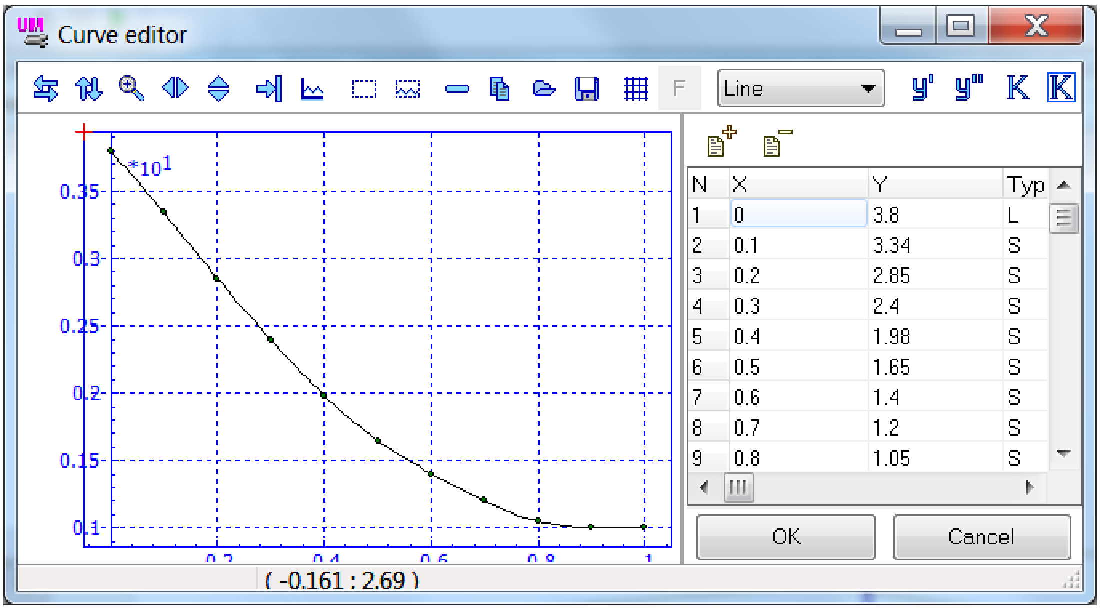


Figure 1.13. Example of torque ratio vs. speed ratio

Mathematical model of the element: Sect. 1.2.1.3. *"Hydraulic torque converter. Combined torque converter"*, p. 1-6.

Description of the element includes the following data, Figure 1.11.

- Assignment of two bodies: *Body1* as the input shaft (impeller) and *Body2* as the output shaft (turbine). Each of the shafts must be connected to a third body by a rotational joint with one degree of freedom. The program detects the corresponding joints and shows their names in the element window, *Joint* 1 and *Joint* 2 in Figure 1.11.
- Data type: Selection of a variant of data description where Mi, Mt stand for the impeller or turbine torque coefficients, *K* is the torque ratio, and *eff*. is the efficiency.
- **Factor** k_M.
- One of five possible description of torque converter characteristics versus speed ratio $i = n_2/n_1$ in the curve editor available by the buttons. For instance, the curves for the impeller torque coefficient and the torque ratio are shown in Figure 1.12, Figure 1.13.

Use the button to view plots of all converter characteristics, Figure 1.14.

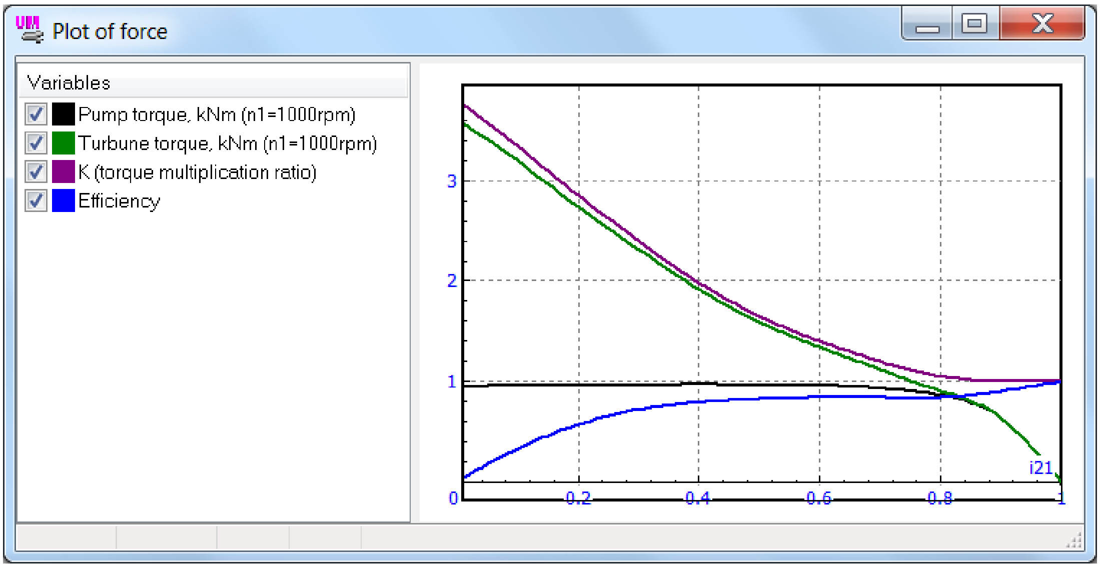


Figure 1.14. Characteristics of a combined torque converter

Model: {UM Data}\SAMPLES\LIBRARY\Driveline\Hydraulic Torque Converter.

1.2.2.4. Data input for hydrostatic drive

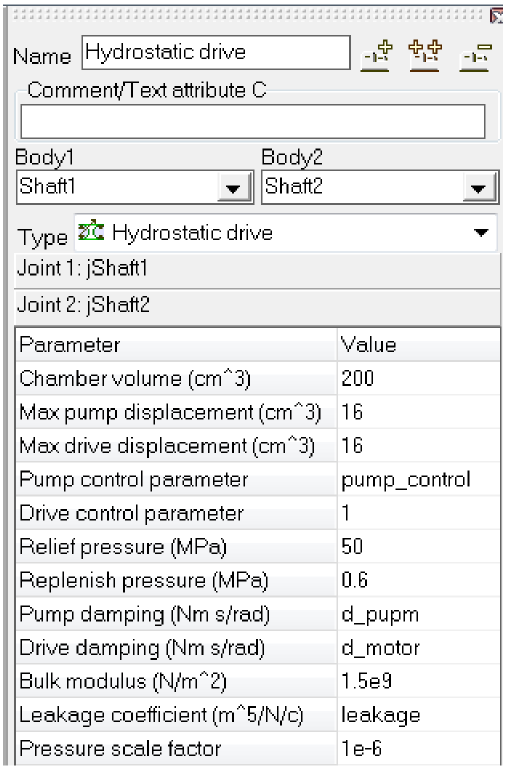


Figure 1.15. Parameters of hydrostatic drive

Mathematical model of the element: Sect. 1.2.1.4. *"Hydrostatic drive"*, p. 1-8.

Description of the element includes the following data, Figure 1.15.

- Assignment of two bodies:
  - *Body1* as the input (pump) shaft
  - *Body2* as the output (motor) shaft

Each of the bodies must be connected to a third body by a rotational joint. The program detects the corresponding joints and shows their names in the element window, *Joint* 1 and *Joint* 2 in Figure 1.16. Rotation of the input shaft can be an explicit time function.

- Setting the drive parameters (numerical values or parameterized constant expressions):
  - Chamber volume V (cm^3);
  - Max pump displacement q_p (cm^3);
  - Max drive displacement q_m (cm^3);
  - Pump control parameter $e_p \in [-1,1]$; it is recommended to use an identifier for parameterization of the parameter in case of the variable displacement pump;
  - Drive control parameter $e_m \in [-1,1]$; it is recommended to use an identifier for parameterization of the parameter in case of the variable displacement motor;
  - Relief pressure (MPa);
  - Charge pressure (MPa);
  - Pump damping v_p (Nm s/rad);
  - Drive damping v_m (Nm s/rad);

- Bulk modulus E (N/m^2);
- Leakage coefficient r (m^5/N/s);
- Pressure scale factor in equations of the drive during simulation; it is recommended to compute the pressure value in MPa; in this case the factor is equal to 10^{-6}.

Model: {UM Data}\SAMPLES\LIBRARY\Driveline\Hydrostatic drive.

1.2.2.5. Data input for planetary gearing

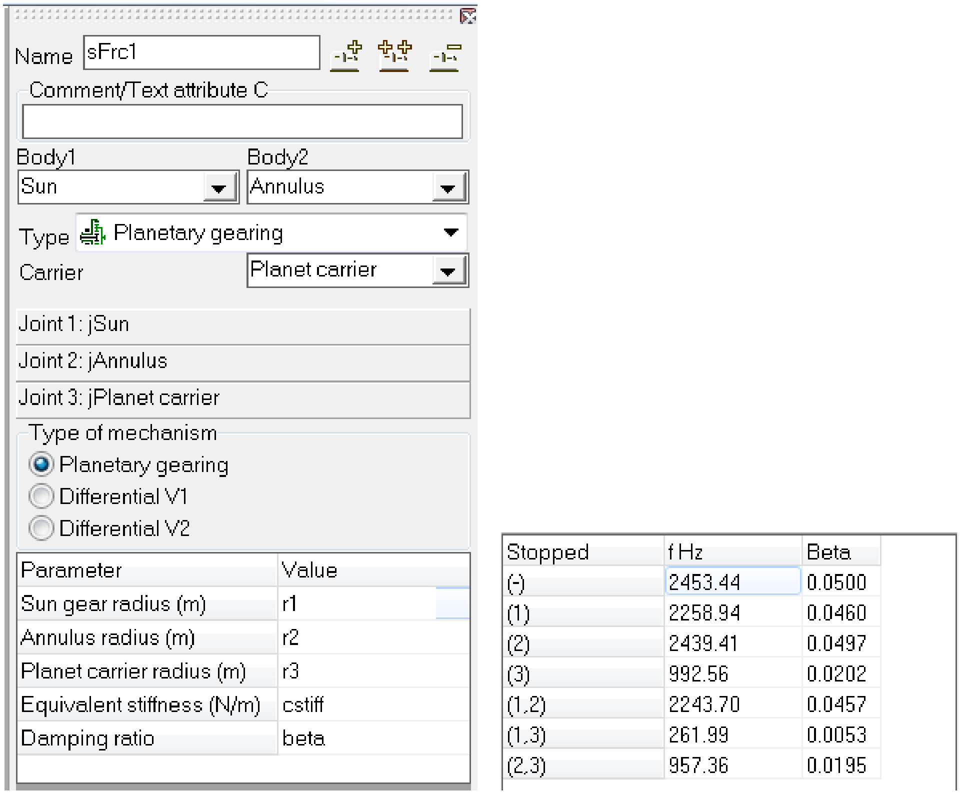


Figure 1.16. Parameters of planetary gearing

Mathematical model of the element: Sect. 1.2.1.5. *"Epicyclic or planetary gearing"*, p. 1-10.

Description of the element includes the following data, Figure 1.16.

- Assignment of three bodies:
  - *Body1* as the sun wheel
  - *Body2* as the annulus
  - Carrier body

Each of the bodies must be connected to a third body by a rotational joint. The program detects the corresponding joints and shows their names in the element window, *Joint* 1, *Joint* 2 and *Joint* 3 in Figure 1.16.

- Setting the gearing parameters (numerical values or parameterized constant expressions):
  - wheel radii, m;
  - equivalent stiffness constant, N/m;

- damping ratio.

After input of the gearing train parameter, the program computes natural frequency of the mechanism as well as damping ratio for seven possible cases (Figure 1.16): all shafts are free (-), one or two shafts are locked, e.g. (2) – locked the annulus, (1,2) – locked the sun wheel and the annulus.

Model: {UM Data}\SAMPLES\LIBRARY\Driveline\Planetary Gearing.

1.2.2.6. Data input for differential

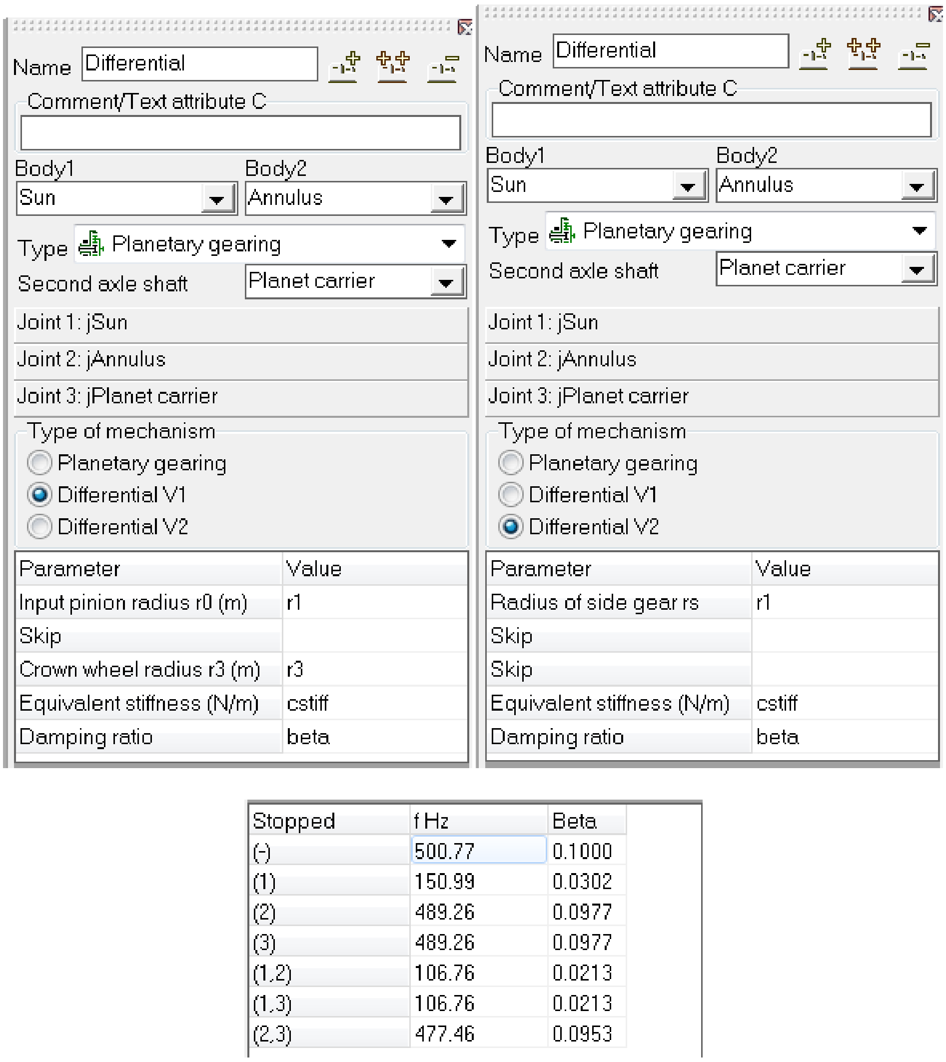


Figure 1.17. Parameters of differential

Mathematical model of the element: Sect. 1.2.1.6. *"Differential"*, p. 1-12.

In UM the differential is implemented as a particular case of the planetary gearing. Description of the element includes the following data.

- Assignment of three bodies:
  - *Body1* as the input shaft;
  - *Body2* as first output shaft;
  - Second output shaft instead of the carrier.

Each of the bodies must be connected to a third body by a rotational joint. The program detects the corresponding joints and shows their names in the element window, *Joint* 1, *Joint* 2 and *Joint* 3 in Figure 1.16.

- Type of differential model V1 or V2, Sect. 1.2.1.6. *"Differential"*, p. 1-12.
- Setting the gearing parameters (numerical values or parameterized constant expressions):
  - input shaft gear (pinion) radius, m (Differential V1);
  - crown wheel radius, m (Differential V1);

- radius of side gear, m (Differential V2);
- equivalent stiffness constant, N/m;
- damping ratio.

After input of the gearing parameter, the program computes natural frequency of the mechanism as well as damping ratio for seven possible cases (Figure 1.16): all shafts are free (-), one or two shafts are locked, e.g. (2) – locked the first output shaft, (2,3) – locked both first output shafts.

1.2.3. Use of standard UM Base elements in transmission model

Many components of a driveline can be modeled by standard UM elements.

1.2.3.1. Gear trains

The *Gearing* special force element allows the user to describe gear trains consisting of any number of gear, see Chapter 2, Sect. *Special forces* | *Gearing*.

A planetary gearing based on modeling each of the gears and toothings is considered in the model {UM Data}\SAMPLES\LIBRARY\Driveline\PlanetaryGearing MBS.

Similar model for a differential is {UM Data}\SAMPLES\Mechanisms\final_drive.

Usually such models have much more degrees of freedom than the simplified elements *Mechanical rotation converter, Epicyclic or planetary gearing and Differential* described in this chapter.

1.2.3.2. Chain gear

The *Chain gear* special force element allows the user to describe the corresponding element of a transmission, see Chapter 2, Sect. *Special forces* | *Chain gear*.

Model: {UM Data}\SAMPLES\LIBRARY\ChainGear.

1.2.3.3. Cardan shaft

A cardan joint is modeled either by a six d.o.f joint or by generalized joint, see Chapter 2, Sect. *Description of joints* | *Six d.o.f joint, Description of joints* | *Generalized joint*.

Model: {UM Data}\SAMPLES\Mechanisms\universal_joint.

1.2.3.4. Friction clutch

A friction clutch is modeled by the friction scalar force element as a joint torque or as a scalar torque, see Chapter 2, Sect. *Force elements* | *Joint forces and torques*, *Force elements* | *Scalar torque*, *Force elements* | *Types of scalar forces*| *Friction force*.

Model: {UM Data}\SAMPLES\Mechanisms\clutch.

1.3. Internal combustion engine

1.3.1. Mathematical model of internal combustion engine

1.3.1.1. ICE parameters

The UM model of internal combustion engine (ICE) is a torque Me applied to the engine crankshaft. The value of the torque depends on the shaft speed $n_e \in [n_{min}, n_{max}]$, position of the throttle $\beta \epsilon [0{,}100\%]$,

$$M_e = M_e(M_e, \beta)$$

The following notations for engine characteristics are used below:

N – engine output power;

n_N – output power speed, rpm;

$M_N = \frac{N}{n_n \pi / 30}$ – torque at output power (Nm);

$M_{\max}$ – maximal torque (Nm);

n_M – maximal torque speed, rpm;

$n_{\min}$ – minimal speed, rpm;

n_0 – idle engine speed without load (spark ignition engine), rpm;

m_{cyl} – number of cylinders;

m_{str} – number of strokes;

V_h – capacity (L);

l_{str} – piston stroke (m);

p_{fmep} – friction main effective pressure (MPa);

$k_M = \frac{M_N}{M_{\max}}$ – maximal torque ratio;

$k_n = \frac{n_M}{n_N}$ – maximal torque/power speed ratio.

1.3.1.2. Full load torque-speed curve

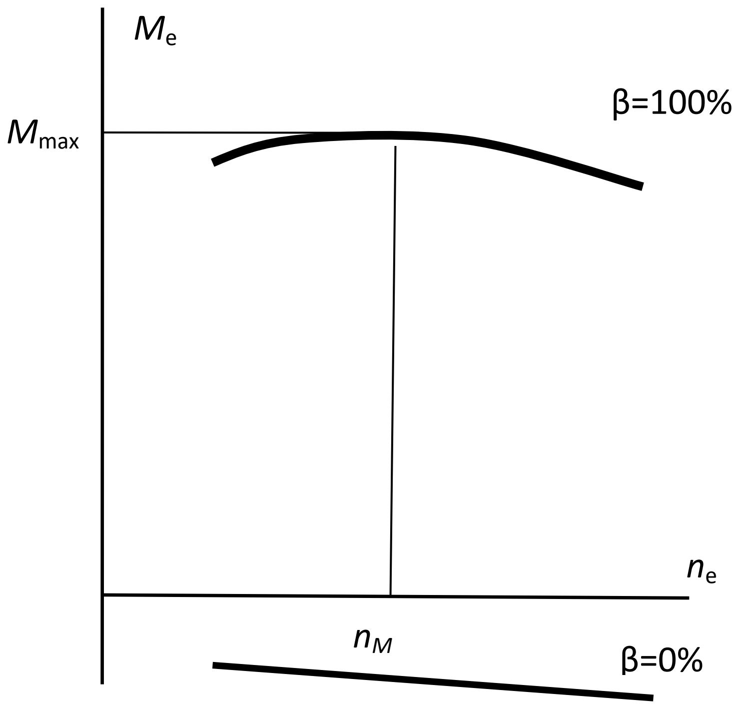


Figure 1.18. Full load torque (β=100%) and lost torque to friction (β=0%)

The full load torque-speed characteristic of ICE is the quasi-static dependence $M_e = M_e(n_e, 100\%)$, which corresponds to $\beta = 100\%$, Figure 1.18.

The full load torque curve can be described by a set of points from the experimental data of approximately according to the analytic formula for the power (Leiderman's formula)

$$N_e = N\frac{n_e}{n_N}\left(a + b\frac{n_e}{n_N} - c\left(\frac{n_e}{n_N}\right)^2\right).$$

This formula can be writer for the torque as

$$M_e = M_N\left(a + b\frac{n_e}{n_N} - c\left(\frac{n_e}{n_N}\right)^2\right). \quad (1.1)$$

Recommended values of a, b, c parameters in Eq. (1.1)

- Petrol engine:

$$a = b = c = 1, k_n = 1$$

- Diesel engine:

1. General formula, Variant 1

$$c = 1, b = \frac{2 - k_M - k_n^2}{1 - k_n}, a = 2 - b;$$

Where

$$k_n = \frac{n_M}{n_N}.$$

In this variant $M_e(n_N) = M_N, M_e(n_M) = M_N/k_M$, but $M_e(n_M)$ is not the maximal output torque.

2. General formula, Variant 2

$$b = \frac{1/k_M - 1}{0.5/k_n + 0.5k_n - 1}, c = \frac{b}{2k_n}, a = 1 + c - b$$

In this variant $M_e(n_N) = M_N$, $M_e(n_M) = M_N / k_M$, and $M_e(n_M)$ is the maximal output torque, i.e.

$$\left.\frac{dM_e}{dn_e}\right|_{n_M} = 0 .$$

3. Direct injection diesel
   $a = 0.87,\ b = 1.13,\ c = 1$
4. Prechamber diesel
   $a = 0.6,\ b = 1.4,\ c = 1$
5. Swirl-chamber diesel
   $a = 0.7,\ b = 1.3,\ c = 1$

Example. Diesel Kamaz 740.10,
Figure 1.19.

$N = 154\,\text{kW},\ n_N = 2600\text{rpm},\ M_{\max} = 667\,\text{Nm},\ n_N = 1600\text{rpm},\ m_{\text{cyl}} = 8,\ m_{\text{str}} = 4,$
$V_h = 10.85\text{L}, l_{\text{str}} = 120$

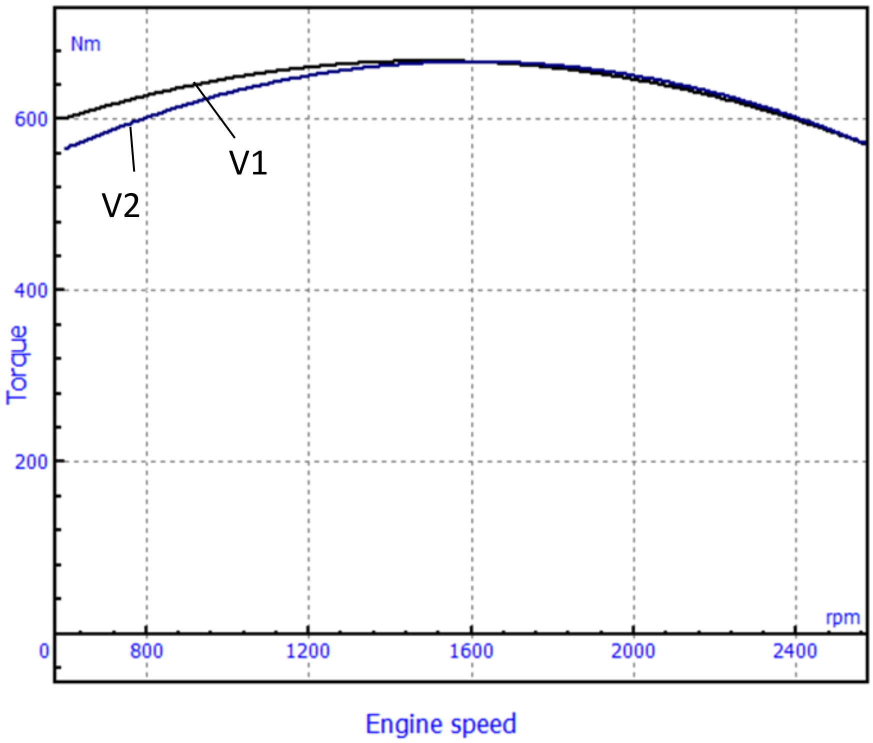


Figure 1.19. Analytic model of load torque for diesel Kamaz 740.10, Variant 1 (V1) and Variant 2 (V2)

1.3.1.3. Torque lost

Torque lost M_{f} corresponds to frictional resistance of engine. It is measured at the throttle position β=0%, Figure 1.18. Friction torque is expressed in term of *friction mean effective pressure* (fmep) p_{fmep} as

$$M_f = \frac{1000 V_h}{\pi m_{\mathrm{str}}} p_{\mathrm{fmep}}.$$

Here m_{str} is the number of strokes; V_h is the engine capacity (L).

Usually a linear dependence of the fmep on the engine speed ω (rad/s) is accepted

$$p_{\mathrm{fmep}} = p_0 + p_1 \frac{l_{\mathrm{str}}}{\pi} \omega.$$

If the stroke length parameter is unknown, its approximate value can be obtained from the formula

$$l_{\mathrm{str}} \approx 0{,}108 \cdot \sqrt[3]{\frac{V_h}{m_{cyl}}}.$$

Recommended values of parameters p_0, p_1 are

- petrol engine

  $p_0 = 0.045,\ p_1 = 0.015$

- diesel engine

  $p_0 = 0.105,\ p_1 = 0.013$

If experimental data on torque lost M_{f} are available, the direct formula can be applied

$$M_{\mathrm{f}} = M_{\mathrm{fa}} + M_{\mathrm{fb}} \omega,$$

which requires evaluation of parameters $M_{\mathrm{fa}}, M_{\mathrm{fb}}$

1.3.1.4. Engine torque map

Engine torque map is the torque/speed dependences for different values of throttle position. Examples of the maps are shown in Figure 1.20, Figure 1.21, Figure 1.22.

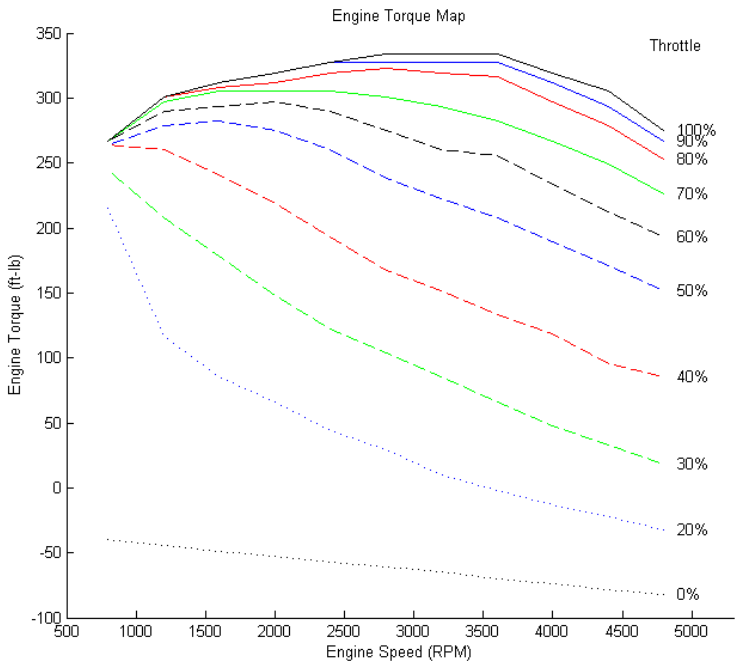


Figure 1.20. Example: spark ignition engine map [2]

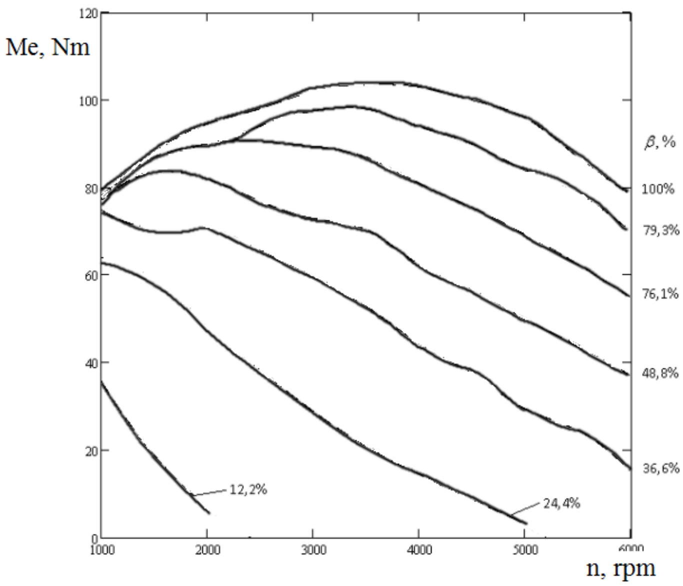


Figure 1.21. Example: spark ignition engine map [3]

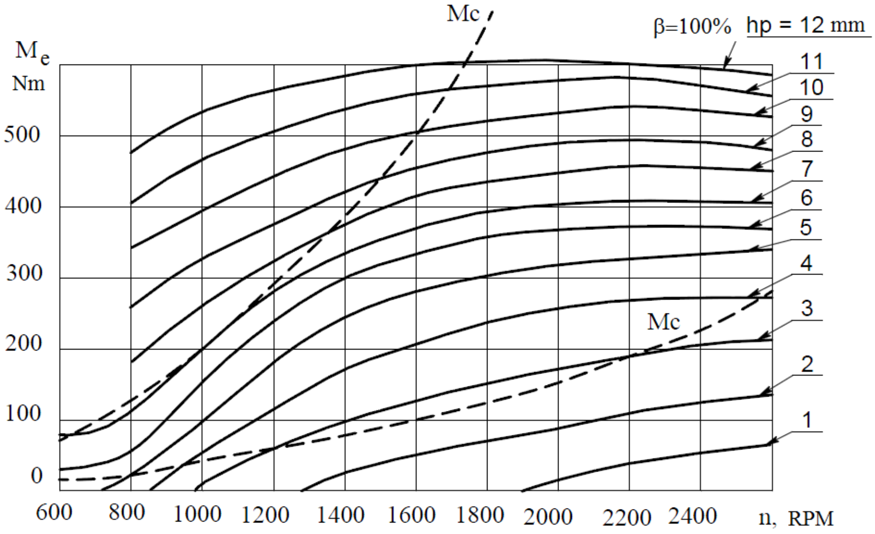


Figure 1.22. Example: diesel engine map [4]

Engine torque map in UM is described by two different ways. First, if experimental curves like in Figure 1.21, Figure 1.22 are known, a set of curves is specified in a special editor. Otherwise, empirical analytic expression specifies an approximate dependence $M_e = M_e(n_e, \beta)$ basing on the analytic full load and lost torque curves.

1.3.1.4.1. Pointwise description of engine map

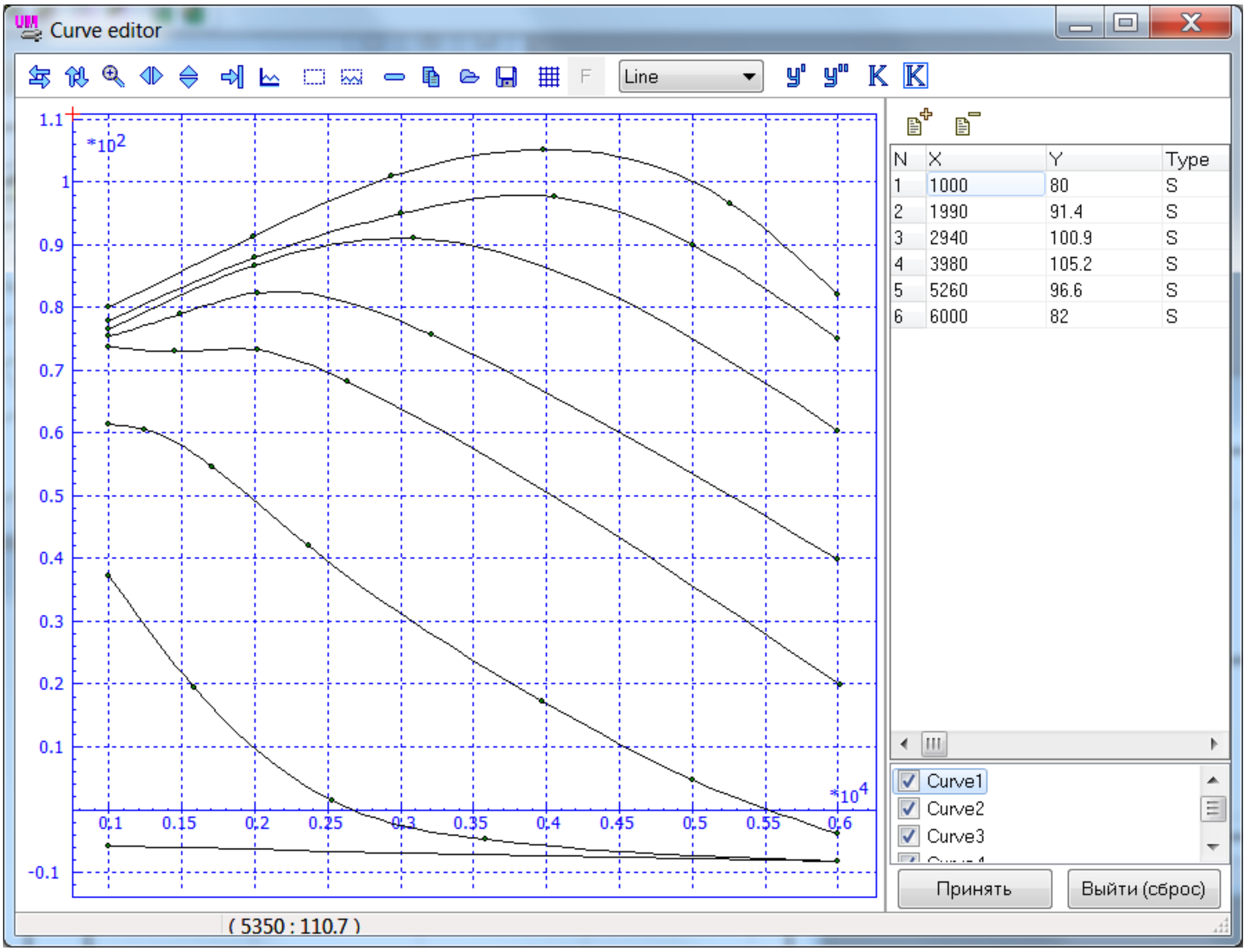


Figure 1.23. Example: pointwise engine map, β=100%, 79%, 76%, 49%, 37%, 24%, 12%

According to this type of the engine map description, a set of torque-speed curves are specified in the curve editor

$$M_e = M_e(n_e, \beta_i),\ i = 1,2..m_M$$
$$\beta_1 = 100\%,\ \beta_{m_M} = 0\%,$$

The curves are supposed to be ordered in decrease of the throttle position, and the set of curves must include both the full load and lost torque curves. A spline interpolation is recommended for smoothing the curves. A linear interpolation is used for evaluation of torque for arbitrary values of the throttle position β

$$M_e(n_e,\beta) = \frac{M_e(n_e,\beta_i)(\beta_{i-1} - \beta) + M_e(n_e,\beta_{i-1})(\beta - \beta_i)}{\beta_{i-1} - \beta_i},\ \beta \in [\beta_i,\ \beta_{i-1}]$$

In Figure 1.23 we have some simplified implementation of experimental data in Figure 1.21, and the interpolated curves are shown in Figure 1.24

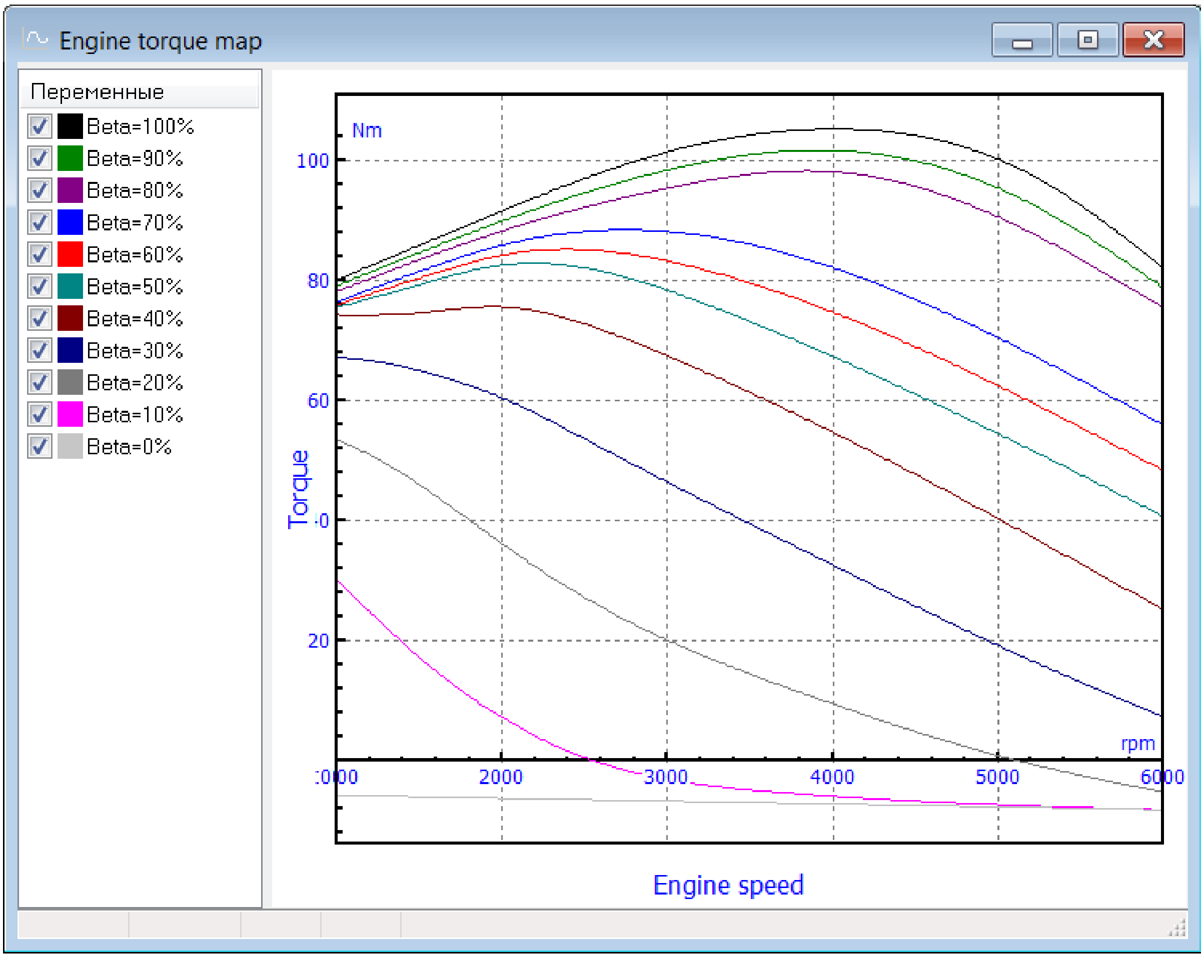


Figure 1.24. Example: interpolation of map in Figure 1.23

Remark. In the case of a spark ignition engine, the curve for the idle throttle position β_0 must be included in the set of curves, Sect. 1.3.1.5.3. Otherwise, the accuracy of the interpolation for small β values is too low, and the engine stalls often by the start.

1.3.1.4.2. Analytic engine map for spark ignition engine

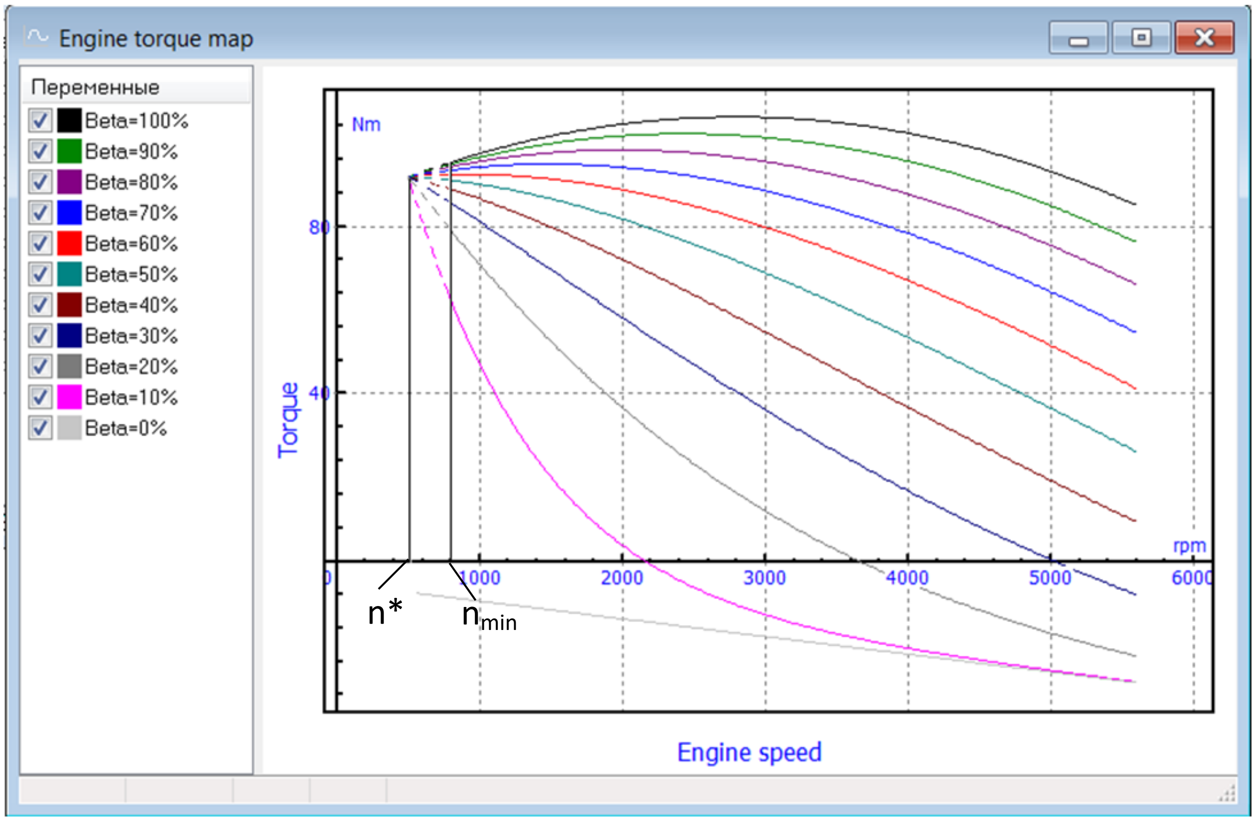
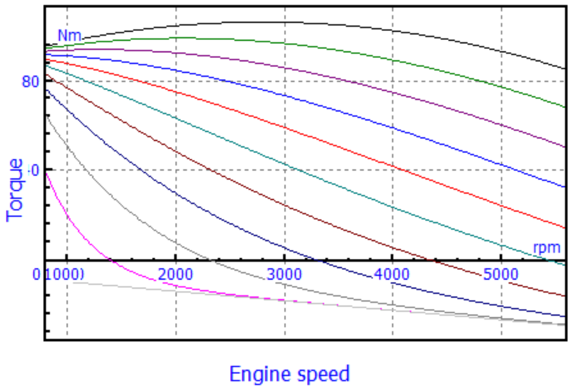
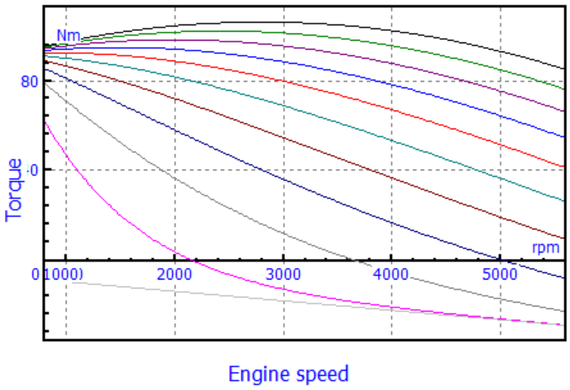


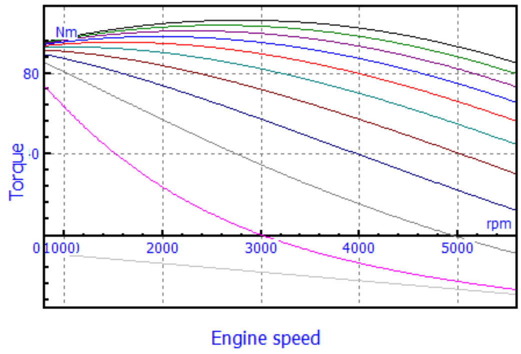
Figure 1.25. Example of analytic engine map for spark ignition engine, the form factor s=0.5



s=1



s=0.5



s=0.3

Figure 1.26. Influence of the form factor s

Analytic expression for spark ignition engine uses analytic representation of the full load and lost torque curves described in Sect. 1.3.1.2, 1.3.1.3. The map model depends on two additional parameters *n** and *s*

$$M_e = M_e(n_e, \beta, n^*, s).$$

The first parameter *n** is the special engine speed, for which extrapolation of the entire map curves in the region of low speeds intersect in one point, see Figure 1.25. The second parameter is the map form factor *s*. Its influence on the map curves is clear from Figure 1.26. Varying of the two parameters allows obtain a good approximation to the experimental map, which is appropriate for many applications, cf. Figure 1.20, Figure 1.21.

1.3.1.4.3. Analytic engine map for diesel engine

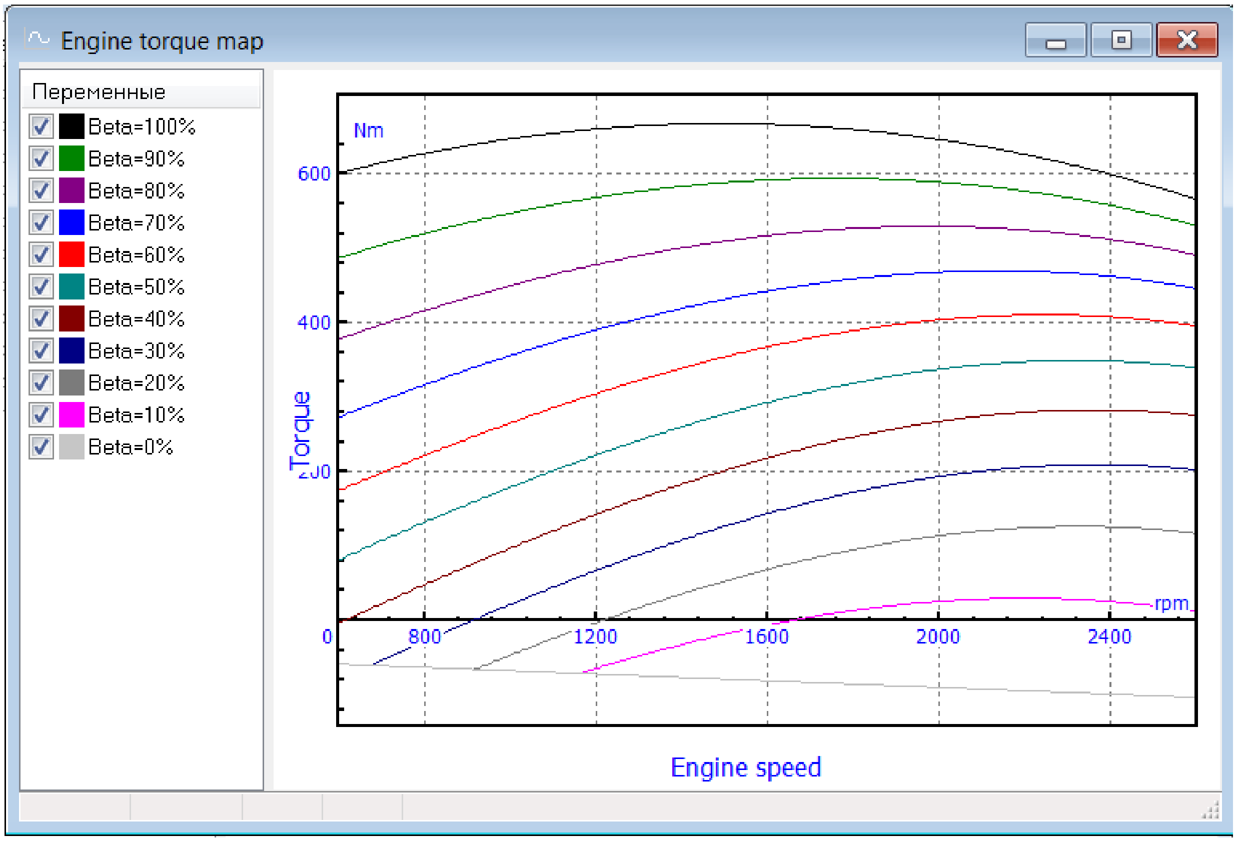
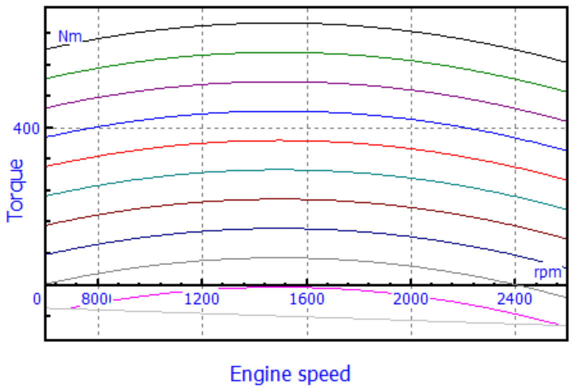
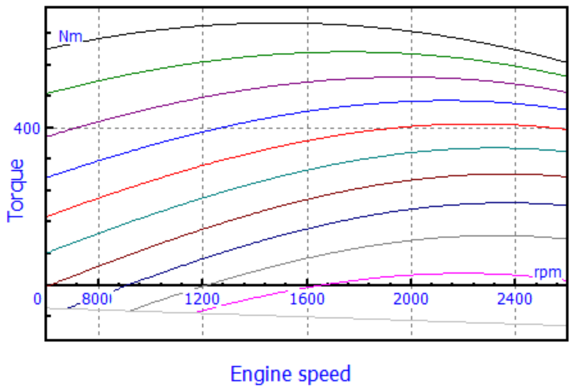


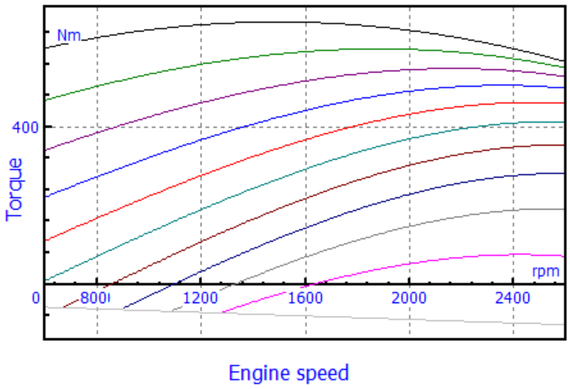
Figure 1.27. Example: analytic engine map for diesel engine Kamaz 740.10, the form factor s=1



s=0



s=1



s=1.5

Figure 1.28. Influence of the form factor s on diesel torque map

Analytic expression for diesel engine uses analytic representation of the full load and lost torque curves described in Sect. 1.3.1.2, 1.3.1.3. The map model depends on one additional parameter *s* (the map form factor *s*)

$$M_e = M_e(n_e, \beta, s).$$

Varying of the form parameter allows the user to obtain a good approximation to the experimental map, cf. Figure 1.22.

1.3.1.5. Engine governors

1.3.1.5.1. One-speed governor for spark ignition engines

Usually no speed governors or one-speed governors are used for road vehicles with a spark ignition engine. The one-speed governor regulates the maximal engine speed, Figure 1.29. The governor model requires one parameter: the speed droop in the controlled interval

$$\delta_{max} = \frac{\Delta n_N}{n_N} \cdot 100\%$$

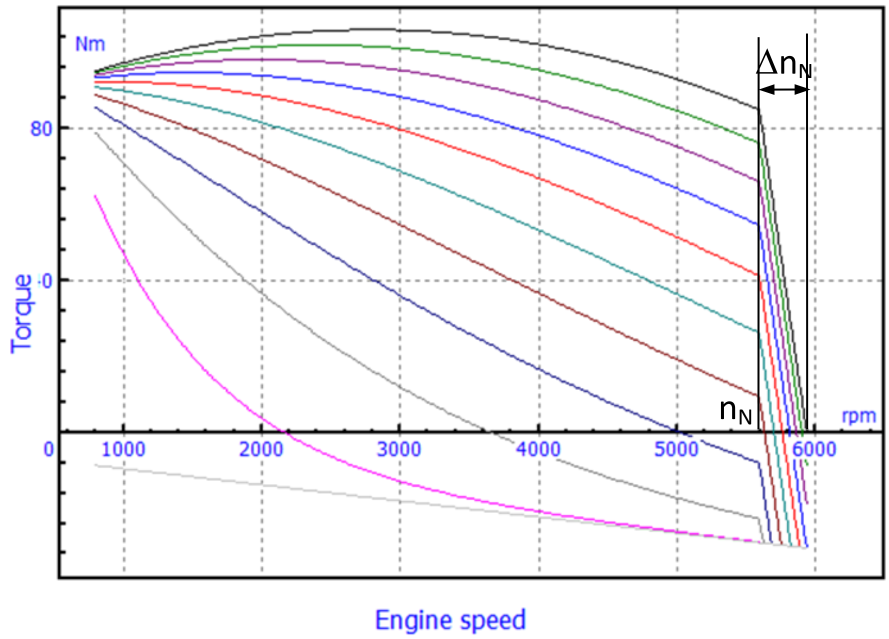


Figure 1.29. Torque map for engine with one-speed governor

1.3.1.5.2. Two-speed and all-speed governors for diesel engine

Two-speed (min-max) governors of diesel engines are usually used with cars, whereas all-speed (variable speed) governors are applied for engines on trucks, tractors, tracked vehicles, Figure 1.30, Figure 1.31.

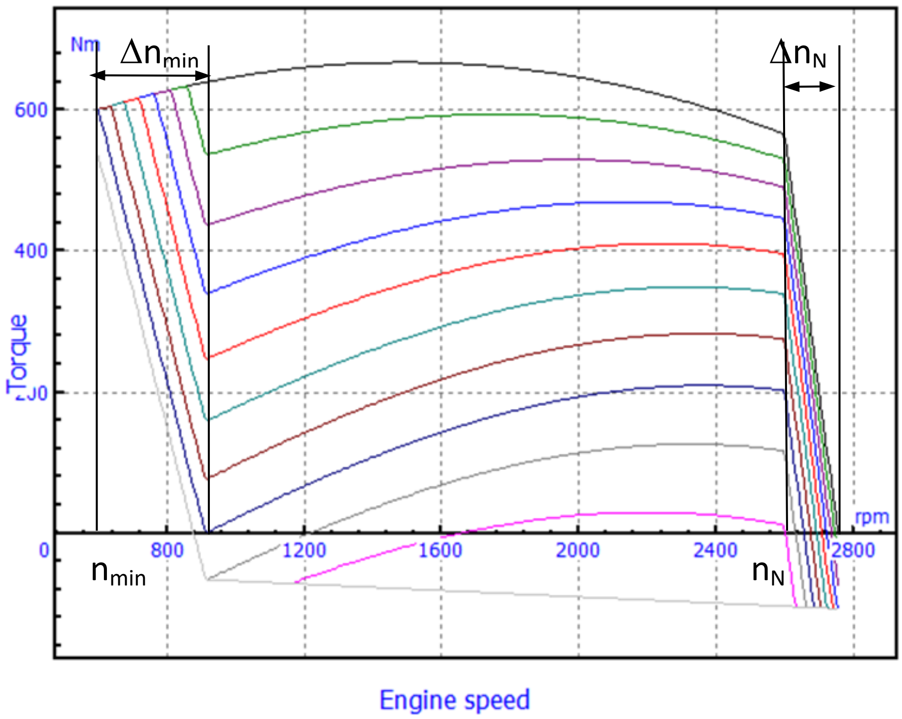


Figure 1.30. Torque map for diesel engine with two-speed governor

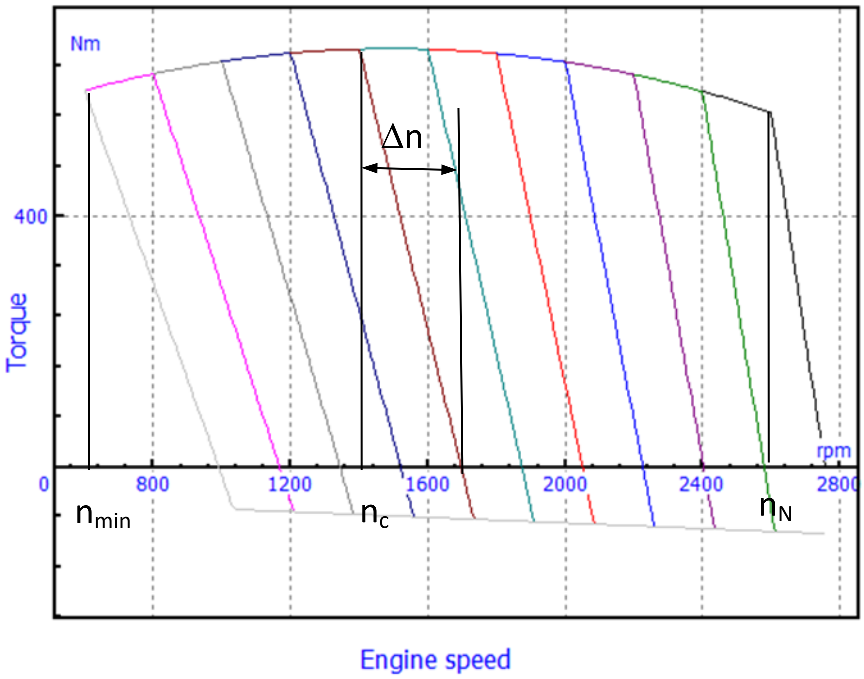


Figure 1.31. Torque map for diesel engine with all-speed governor

Both two- and all-speed governor models require two parameters: the speed droop for the maximal and minimal speeds.

$$\delta_{max} = \frac{\Delta n_N}{n_N} \cdot 100\%, \ \delta_{min} = \frac{\Delta n_{min}}{n_{min}} \cdot 100\%$$

In the case of the all-speed governor, the controlled speed value n_c depends on the accelerator pedal position $\psi \in [0,1]$

$$n_c = n_{min}(1-\psi) + n_N \psi$$

The speed droop for an arbitrary value of the controlled speed value n_c is approximately computed as linear interpolation

$$\delta = \frac{\Delta n}{n_c} \cdot 100\% = \frac{\delta_{min}(n_N - n_c) + \delta_{max}(n_c - n_{min})}{n_N - n_{min}}$$

The speed droop for the maximal speed is 5÷8%, and it increases for lower controlled speeds up to 70% for minimal speed.

1.3.1.5.3. Accelerator pedal and throttle position

Positions of the pedal and engine throttle satisfy a first order differential equation

$$\tau_\beta \dot{\beta} + \beta = \beta_{st}(\psi),$$

where $\beta_{st}(\psi)\%$ is the stationary dependence of the throttle position of the pedal state $\psi \in [0,1]$, τ_β is the lag in throttle reaction on the pedal position change. The default value is $\tau_\beta = 0.1s$.

The function $\beta_{st}(\psi)$ is

$$\beta_{st}(\psi) = 100\psi$$

for diesel engines and

$$\beta_{st}(\psi) = \beta_0(1-\psi) + 100\psi$$

for spark ignition engines. Here β_0 is the idle throttle position, which is computed from the equation

$$M_e(n_0, \beta_0) = 0$$

with n_0 as the idle engine speed without load.

1.3.1.6. Engine start and stalling

At engine start, the constant torque $M_{start} = M_e(n_{min}, \beta = 1)$ is applied to the crankshaft. The start mode is over when $n_e = n_{min}$.

In the case of manual transmission, the engine start fails if the transmission is not in neutral and the clutch is engaged.

Engine stalls if its speed decreases to a half of the minimal speed n_{min}.

1.3.2. Adding ICE to model in UM Input program

To add an internal combustion engine to a multibody system, the user must specify a body corresponding to the engine crankshaft and parameterize the engine torque by an identifier.

Engine shaft is a rigid body, which has one rotational degree of freedom relative to another rigid body. The latter can be e.g. engine case, car body in case of road vehicle or a hull for a tracked vehicle.

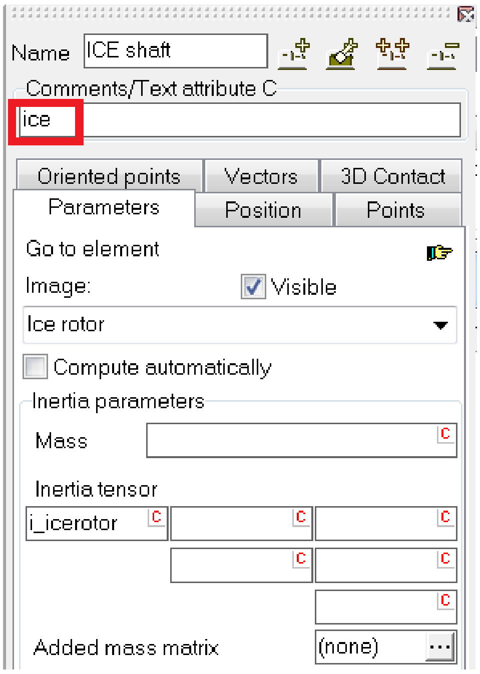


Figure 1.32. ICE crankshaft body

The program identifies a body as a shaft if the “ice” text attribute of C type is assigned to the body, Figure 1.32. An example of a rotation joint is shown in Figure 1.33. The joint force of the *Expression* type parameterizes the engine torque acting on the shaft. It is recommended to use the standard identifier **ice_torque** for parameterization of the torque.

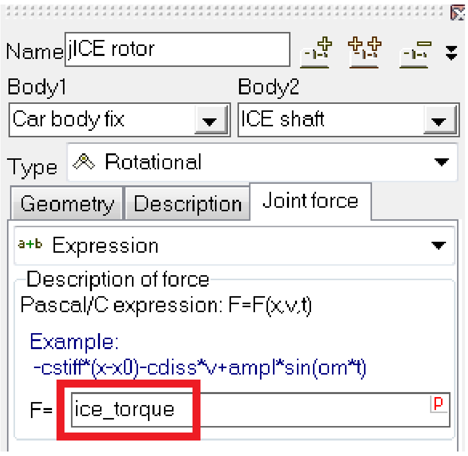


Figure 1.33. Joint specifying the shaft rotational degree of freedom

1.3.3. Setting engine parameters in UM Simulation program

In simulation of a car model, the ICE parameters are specified on the **Road Vehicle | Transmission | ICE** tab of the simulation inspector, Figure 1.34.

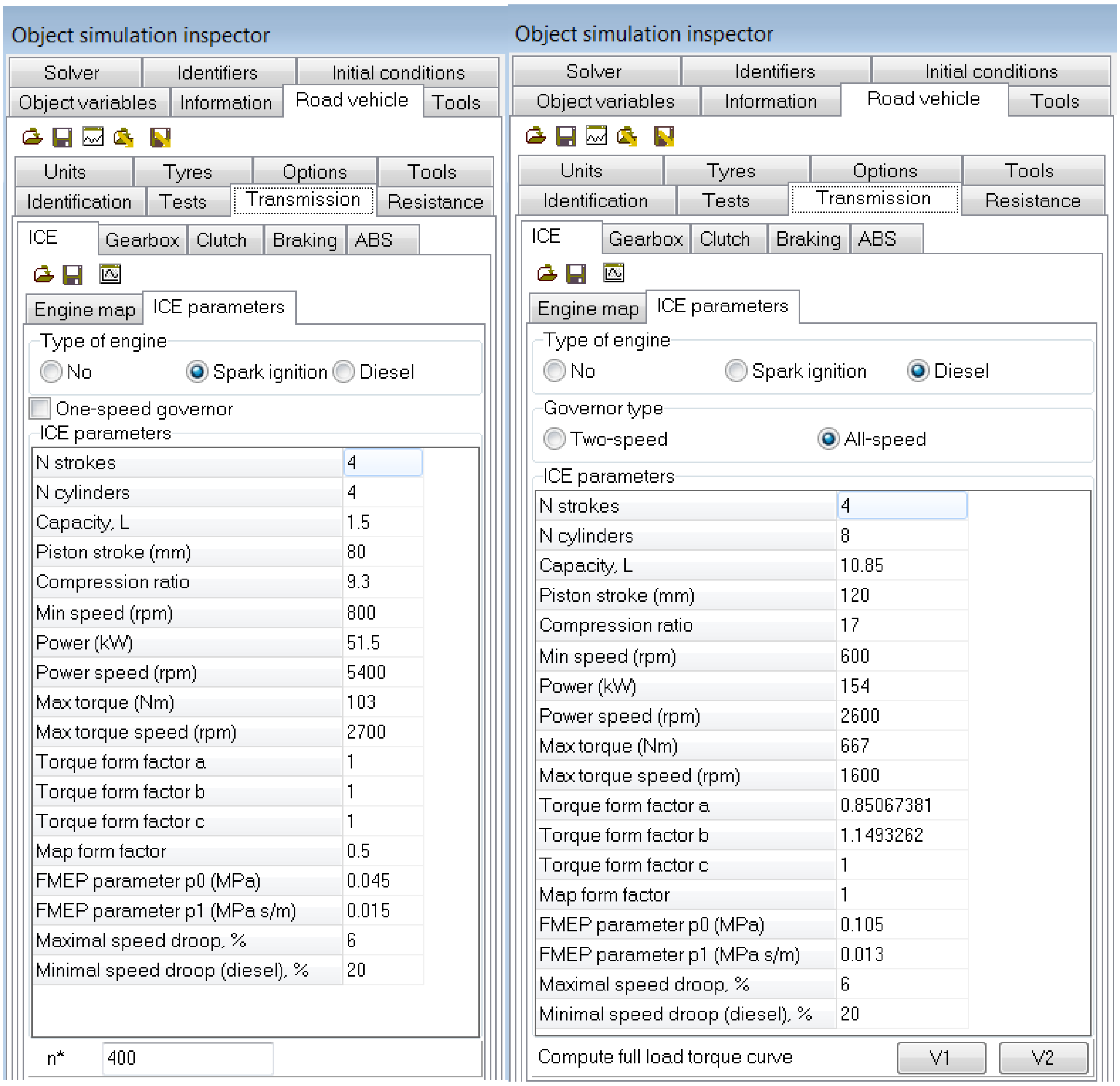


Figure 1.34. ICE parameters for spark ignition and diesel engines

Buttons in the top of the ICE tab are used to

- save ICE parameters in a *.ice text file;
- read ICE parameters from a *.ice text file;
- draw torque map in a graphic window, Sect. 1.3.1.4.

ICE parameters tab contains the following elements.

- Selection of type of engine.
- Setting governor type, Sect. 1.3.1.5.
- List of numeric parameters, specifying the engine:
  - Number of strokes m_{str} ;

- Number of cylinders m_{cyl};
- Capacity V_h ;
- Piston stroke l_{str} ;
- Compression ratio;
- Minimal engine speed $n_{\min}$;
- Engine output power N;
- Output power speed n_N ;
- Maximal torque $M_{\max}$;
- Maximal torque speed n_M ;
- Form parameters a, b, c for analytic full load torque curve, Sect. 1.3.1.2; in case of a diesel engine, the parameters can be either set directly or computed by click on the V1 or V2 buttons;
- Map form factor s, Sect. 1.3.1.4.2, 1.3.1.4.3;
- Fmep parameters p_0, p_1, Sect. 1.3.1.3;
- Minimal (diesel) and maximal speed droop in governor model, Sect. 1.3.1.5.

- Special engine speed **n*** used in the analytic torque map in case of spark ignition engine, Sect. 1.3.1.4.2.
- Buttons **V1** (Variant 1) and **V2** (Variant 2) for evaluation parameters of the diesel full load torque a, b, c, Sect. 1.3.1.2.

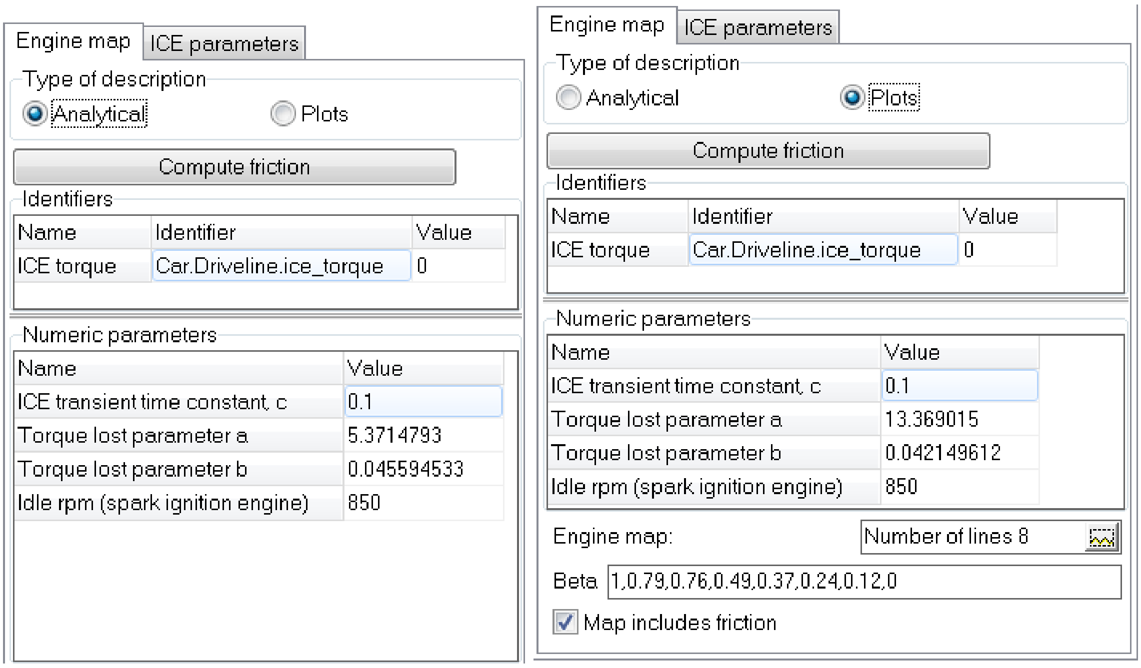


Figure 1.35. Engine map parameters

Engine map tab contains the following elements.

- Selection of type of map description: *Analytical* (Sect. 1.3.1.4.2, 1.3.1.4.3) or *Plots* (Sect. 1.3.1.4.1).

- Button **Compute friction**: click the button to compute the torque lost parameters $M_{\text{fa}}, M_{\text{fb}}$ according to the fmep parameters p_0, p_1, Sect. 1.3.1.3, Figure 1.34.
- Identifier, parameterizing the engine torque in the UM model of vehicle (ice_torque in Figure 1.35), Sect. 1.3.2.
- List of numeric parameters:
    - ICE transient time constant τ_β, i.e. the lag in throttle reaction on the pedal position change, Sect. 1.3.1.5.3;
    - Parameters $M_{\text{fa}}, M_{\text{fb}}$ in the model of torque lost, Sect. 1.3.1.3; the parameter can be set directly or computed according to the fmep parameters p_0, p_1 by the Compute friction button;
    - Idle engine speed without load (spark ignition engine) n_0, Sect. 1.3.1.5.3.

- Group of elements for description of engine map by plots, Sect. 1.3.1.4.1:
    - **Engine map:** click on the  button call the curve editor for pointwise description of the torque curves, Figure 1.23. See the user's manual Chapter 3, Sect. *2D curve editor*. It is recommended to use a spline interpolation of torque curves. Curves must be ordered in decrease of the throttle position.
    - Box **Beta** contains values of throttle positions 1÷0 for each of the torque curves.
    - Check **box Map includes friction** must be checked if the torque value includes the torque lost to friction.

1.4. Simulation of road and tracked vehicle transmissions

In this section we consider methods for development of transmissions (or drivelines) of road and tacked vehicles with UM. The model includes ICE and mechanisms for traction transmission with hydraulic and mechanical force elements. Transmission control is executed with the help of the parameterization of the force elements, e.g. the value of braking torque, and with adding special controlling identifiers such as the gearbox position, the position of accelerator, brake, clutch pedals and so on.

1.4.1. Transmission model as included subsystem

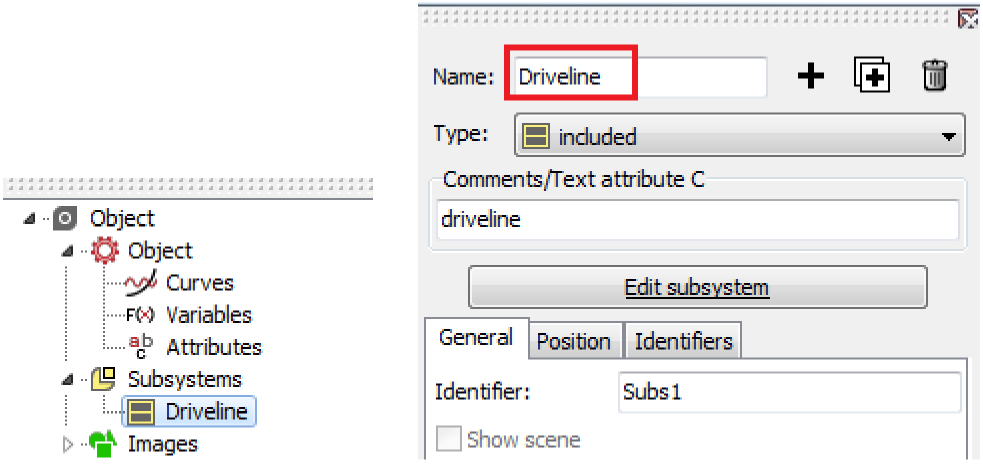


Figure 1.36. Driveline as included subsystem

It is recommended to develop a driveline as an included subsystem marked with the driveline text attribute, see Figure 1.36, right. With this method, the shift of existing transmissions to new vehicle models becomes easier. In particular, this principle is used for development of UM database of standard transmission and steering system of tracked vehicles (TV).

The common elements of drivelines are an ICE and a clutch or a hydraulic apparatus, which transmit the driving torques from ICE to the gearbox.

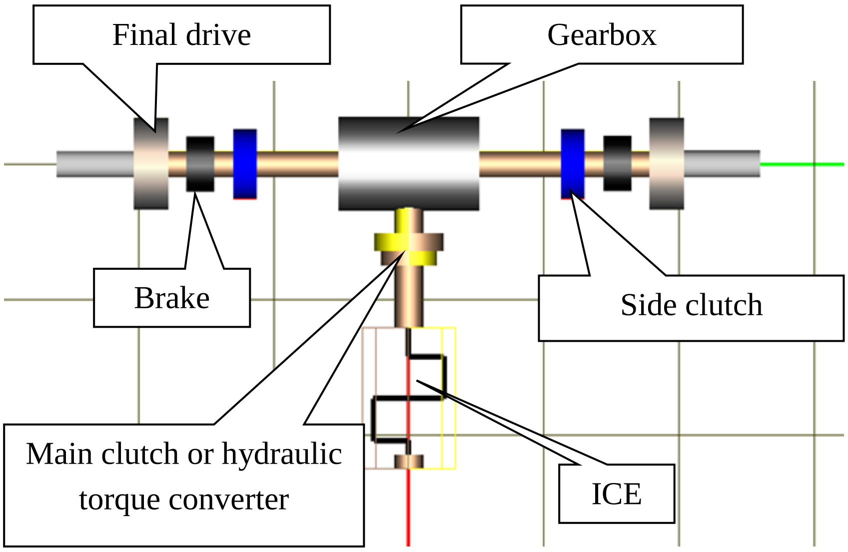


Figure 1.37. Example of transmission with Clutch-Brake steering mechanism

Driveline images in UM are simplified similar to Figure 1.37. The driveline element models are simplified as well. For example, a gear box or a final drive is modeled by force elements, which converts rotation. The model of such the element is specified by the transmission ratio, stiffness and damping constants, efficiency factor, Sect. 1.2.1.1. *Mechanical rotation converter*, 1.2.2.1. *Data input for mechanical rotation converter*. If necessary, the user can develop a detailed model of some assemblies with gearing wheels.

1.4.2. Brakes and friction clutches

A frictional scalar force elements are used for modeling the brakes and friction clutches, see (see. Chapter 2, Mechanical system as an object for modeling, file UM_Technical_Manual.pdf, Sect. *Types of scalar forces | Friction force*).

A friction clutch connects/disconnects two rotational elements (bodies) of a transmission, whereas a brake decreases the rotational speed of one body relative to another one, which usually does not rotate (for instance, a car body of a TV hull). That is why the brake is modeled as a *joint torque* acting on the rotational degree of freedom, and the clutch is considered as a *scalar torque*, see Chapter 2, Sect. *Joint forces and torques*, *Scalar torque*.

Another difference consists in the following consideration. Friction torque in a brake is variable and depends on the brake pedal stroke. In the case of a friction clutch the torque value is a given value. This difference requires different methods in parameterization of friction torques. In the case of a brake, an identifier is introduced for the torque value, and this identifier is used in the brake control. In the case of a friction clutch, the friction torque is the product of the sticking torque and a control identifier, which unit/zero values corresponds to the engagement/disengagement state of the clutch; a smooth change of the controlling identifier results in an gradual change of the torque, a stepwise change corresponds to an instantaneous switching the clutch.

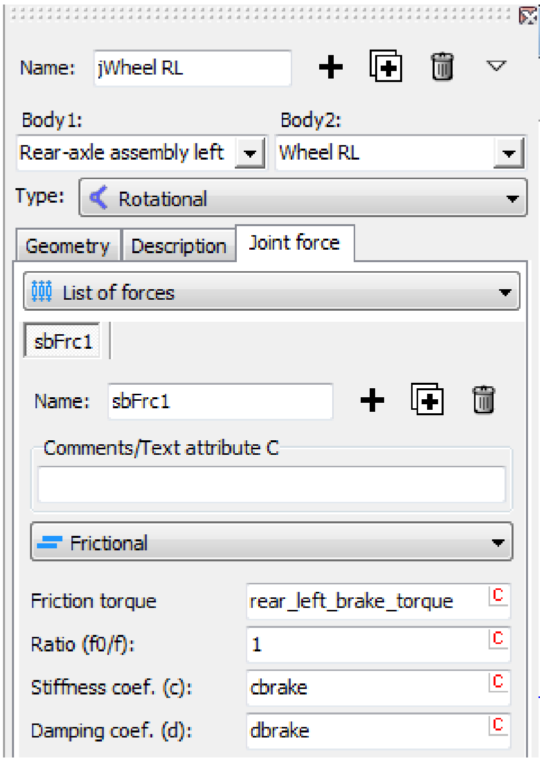


Figure 1.38. Brake model for wheel of a car

Brake model as a joint torque for the left rear wheel of a car is shown in Figure 1.38. The identifier *rear_left_brake_torque* parameterizes the torque value, see Sect. 1.4.3 *Automotive brake model*.

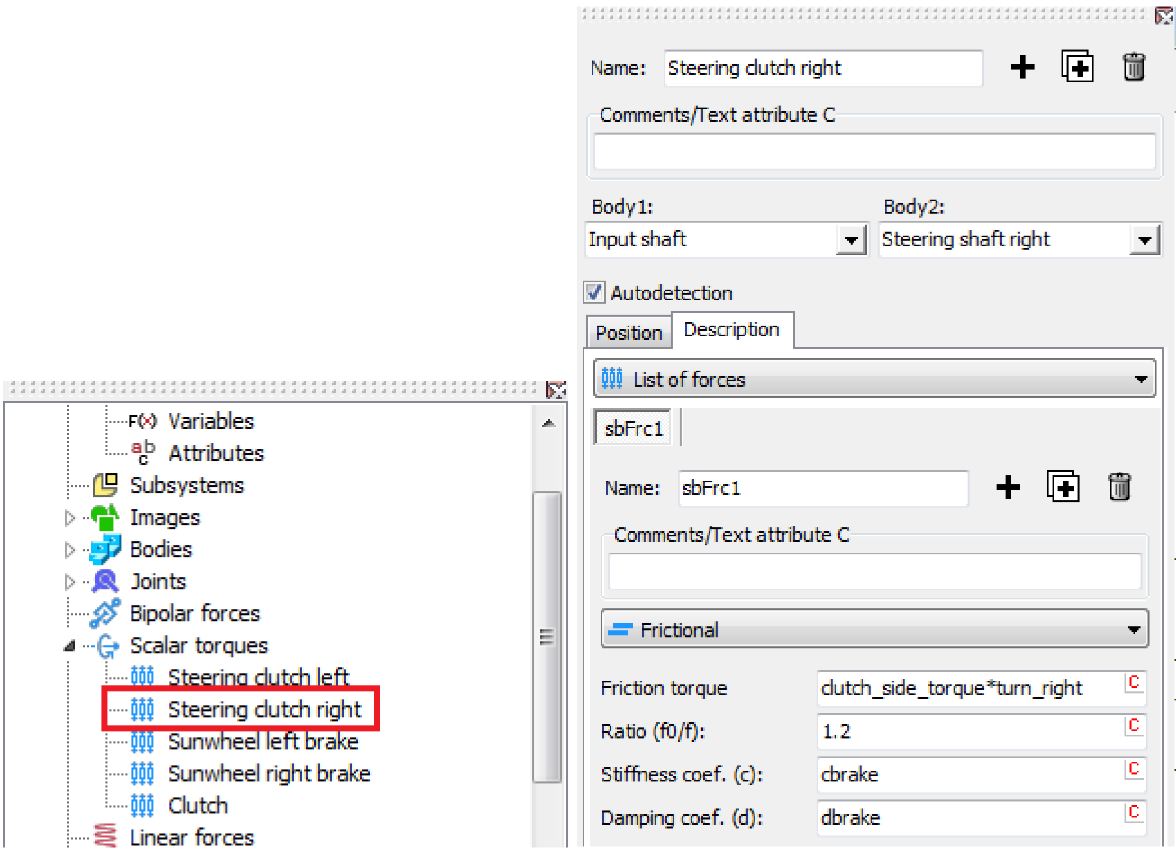


Figure 1.39. Model of friction clutch

The model of a right friction clutch of a TV steering system as a scalar torque is shown in Figure 1.39. The *clutch_side_torque* identifier parameterizes the maximal torque value. The engagement/disengagement of the clutch is executed by the controlling identifier *turn_right*. If *turn_right*=1, the right turn of the tracked vehicle occurs. The value *turn_right*=0 disengages the clutch.

Remark. Rotation axis of bodies connected by a friction clutch should be parallel, otherwise the model work will be incorrect.

1.4.3. Automotive brake model

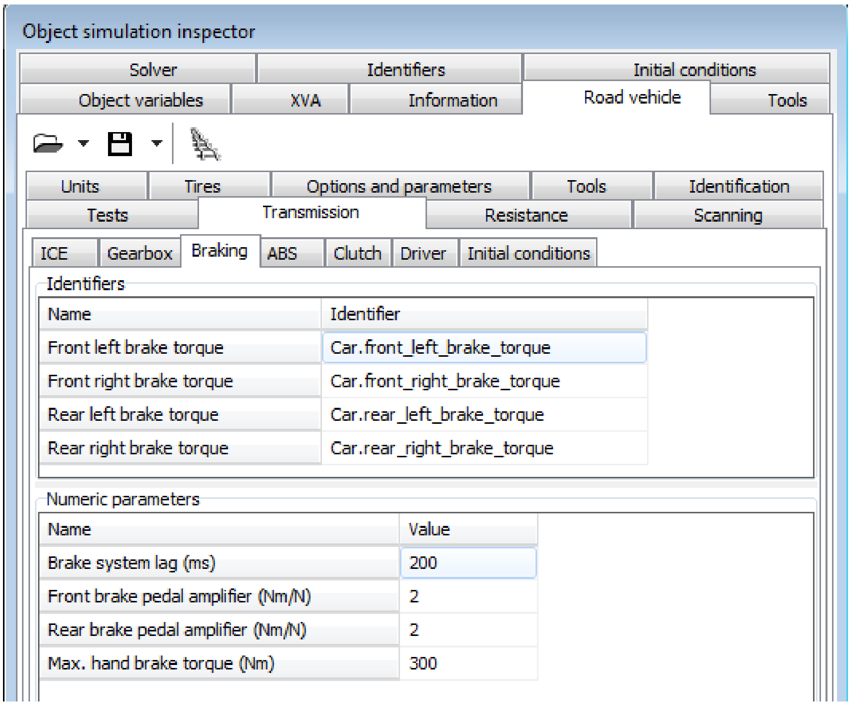


Figure 1.40. Automotive brake parameters

To control the car braking process, identifiers parameterizing the brake torque for each of the wheels should be selected like in the window shown in Figure 1.40. It is recommended to use the standard identifiers from Figure 1.40 in description of the braking forces, see Figure 1.38.

Consider the brake torque M_i for wheel *i*. The desired brake torque M_i^* depends on the force applied to the brake pedal F_b

$$M_i^* = k_i F_b$$

where k_i is the amplifiers, which can be different for the front and rear wheels, see Figure 1.40. To take into account some delay between pressing the brake pedal and the brake reaction, the **Brake system lag** parameter T_b is introduced so that the brake torque M_i is computed from the differential equation

$$T_b \frac{dM_i}{dt} + M_i = M_i^*.$$

The brake pedal force is obtained from the transmission control, see Sect. 1.4.7. *Transmission control*. In addition, the ABS can affect the torque value, see Sect. 1.4.6 *Anti-lock braking system (ABS)*.

1.4.4. Torque transfer from engine to gearbox: friction clutch and hydraulic apparatus

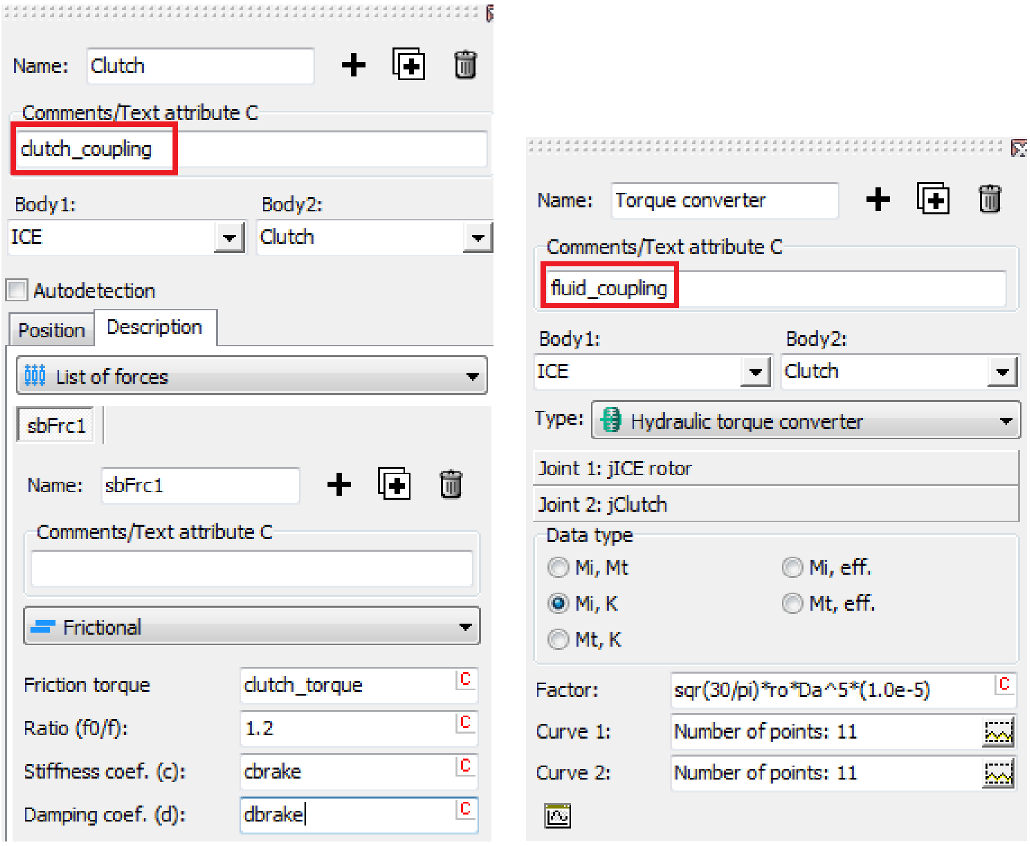


Figure 1.41. Force elements modeling friction clutch and torque converter

Either friction clutch of hydraulic apparatus are used for modeling the transfer of torque from ICE to transmission, Figure 1.41.

Friction clutch is modeled by a scalar torque, Sect. 1.4.2. *"Brakes and friction clutches"*, c. 1-47. The text attribute *clutch_coupling* should be assigned to the element, Figure 1.41 left, and the torque must be parameterized by an identifier, which default and recommended name is *clutch_torque*. This identifier is used by UM for setting the torque value depending on the clutch pedals position.

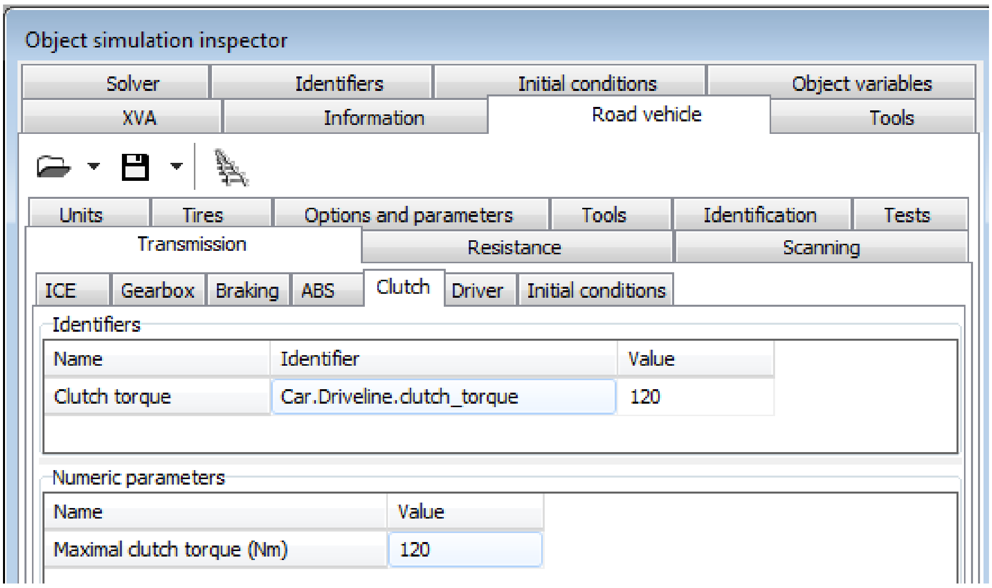


Figure 1.42. Friction clutch parameters in UM Simulation

Before start of the simulation, the user should select on the **Clutch** tab the identifier, which parameterizes the friction torque. The maximal friction torque must be set as well, Figure 1.42.

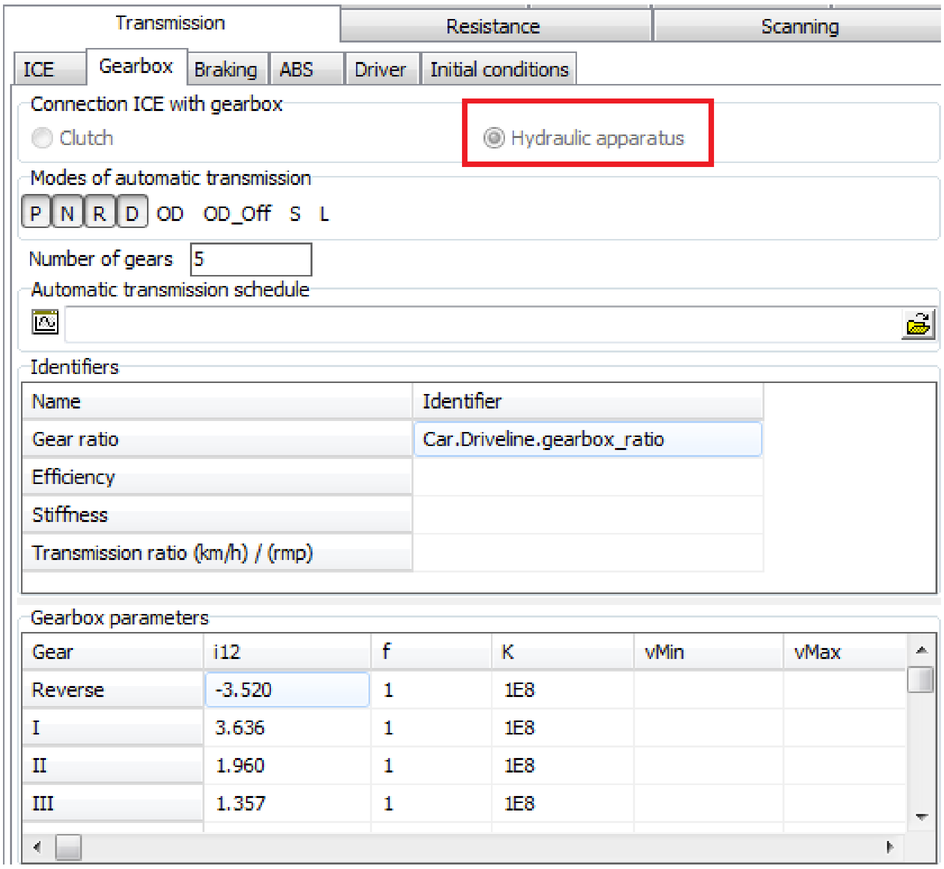


Figure 1.43. Type of engine/gearbox connection

A hydraulic apparatus is either a torque converter (Sect. 1.2.1.4. *Hydrostatic drive*), or a fluid coupling (Sect. 1.2.1.2. *Fluid coupling*). In contrary to the friction clutch, the hydraulic apparatus if not a driver controlled element. The description of the element in the UM Input must include the identifying text attribute *fluid_coupling*, Figure 1.41.

If a hydraulic apparatus in presented in a model of a road vehicle as an element connecting the ICE and the gearbox, the program detects an **automatic gearbox.**

UM Simulation program uses the text attributes of force elements *clutch_coupling* and *fluid_coupling* for automatic detection of the coupling type between the engine and the gearbox and represents it on the Gearbox tab, Figure 1.43.

As a rule, models of transmissions of tracked vehicles included in UM database mostly contain both the friction clutch and hydraulic apparatus. In this case the user can select the necessary coupling type on the **Gearbox** tab, Figure 1.44. The disabled element is ignored.

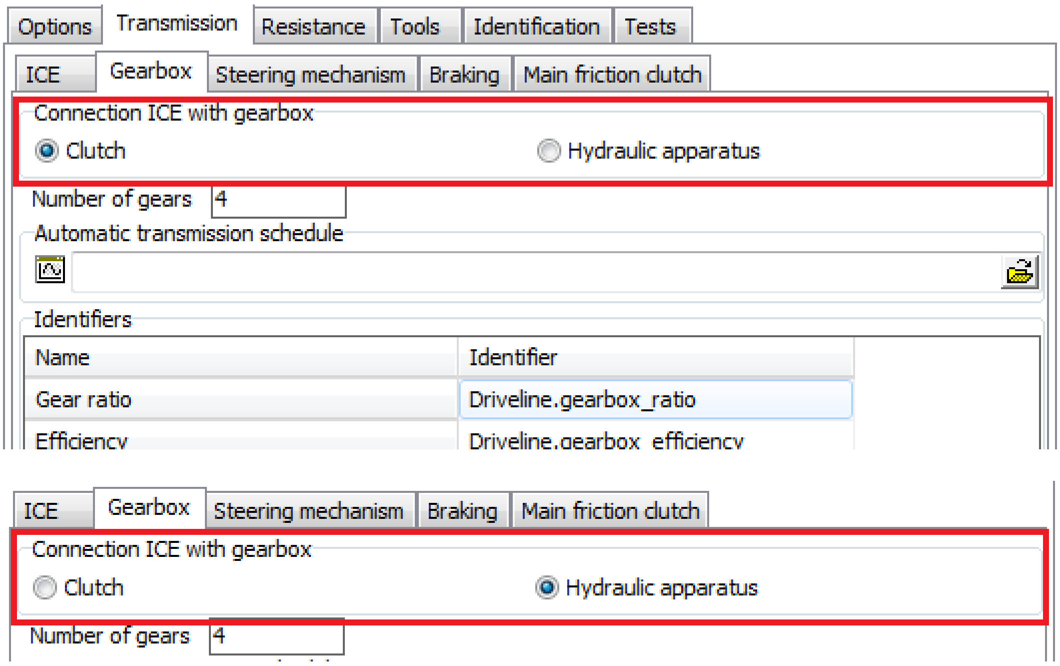


Figure 1.44. Selection of ICE/gearbox coupling type

1.4.5. Gearbox

1.4.5.1. Force element as gearbox model

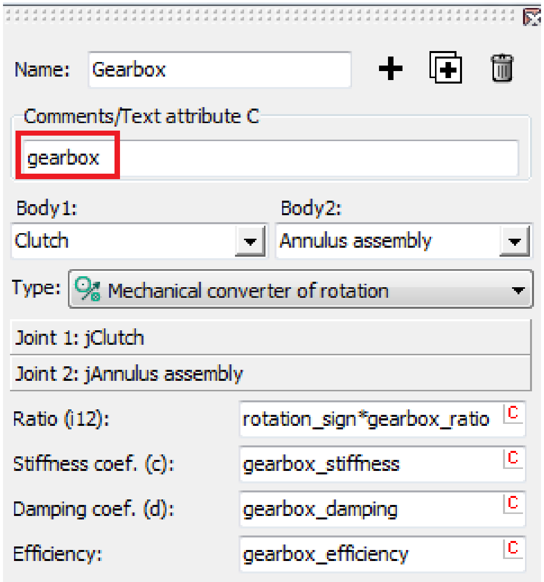


Figure 1.45. Force element modeling gearbox

A simplified model of the gearbox is presented by the force element **Mechanical converter of rotation**, Sect. 1.2.1.1 *Mechanical rotation converter*, Figure 1.45.

Successful automatic identification of the element requires setting the text attribute *gearbox*, like in Figure 1.45.

It is recommended to use the parameterization of the element parameters by the default identifies

– *gearbox_ratio*,

– *gearbox_stiffness* – stiffness constant, (Nm/rad),

– *gearbox_efficiency*.

The two last identifiers allow changing the gearbox stiffness and efficiency in dependence on the gearbox ratio. If the user do not plan to change the values of these parameters, numerical values can be set directly.

The *rotation_sign* identifies allows matching the correct rotation of the output shaft bysetting either +1 or -1 value, Figure 1.45.

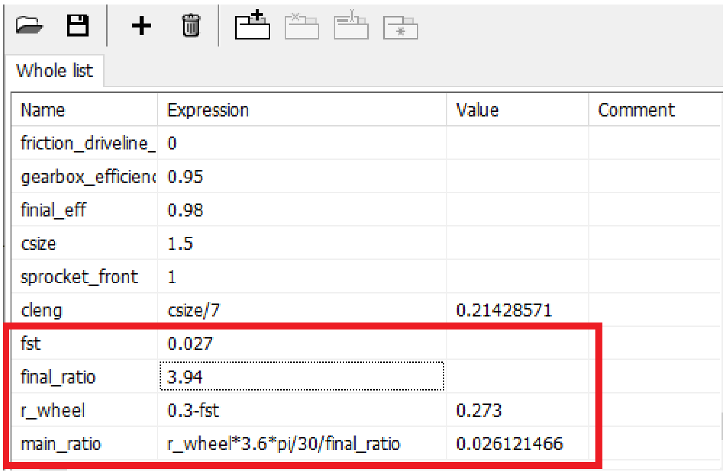
1.4.5.2. Identifier for transmission ratio

To calculate the dependence of the longitudinal speed of the vehicle on the engine speed n_e (rpm), the program uses the transmission ratio i_m (km/h / rpm), which connects the longitudinal speed of the vehicle *v* (km/h) with the frequency of the gearbox output shaft n_g (rpm):

$$v = n_g i_m = \frac{n_e i_m}{i_g},$$

where i_g is the gearbox ratio depending on the gear position.

To include this gear ratio in the model, an identifier is introduced with the default name *main_ratio*. The user can apply both the direct numerical value of this identifier and an expression. In the vaz21009 T car model, an expression is used that includes the wheel radius (*r_wheel*) taking into account the static deflection (*fst*), and the final gear ratio (*final_ratio*), Figure 1.47.



Name	Expression	Value	Comment
friction_driveline_	0		
gearbox_efficienc	0.95		
finial_eff	0.98		
csize	1.5		
sprocket_front	1		
cleng	csize/7	0.21428571	
fst	0.027		
final_ratio	3.94		
r_wheel	0.3-fst	0.273	
main_ratio	r_wheel*3.6*pi/30/final_ratio	0.026121466	

Figure 1.46. Expression for the transmission ratio

1.4.5.3. Setting gearbox parameters in UM Simulation

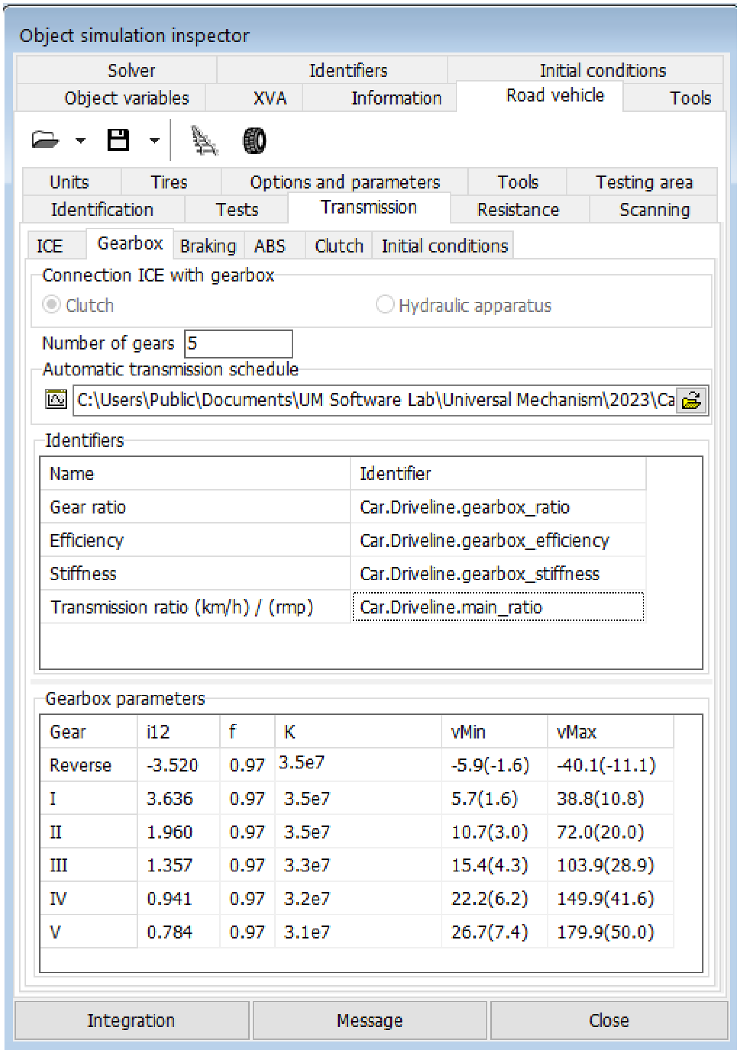


Figure 1.47. Gearbox parameters

The **Transmission | Gearbox** tab is used for setting the gearbox parameters in the **Object simulation inspector,** Figure 1.47.

The following data is set:

- Identifier for the gearbox ratio (required parameter),
- Identifier for gearbox efficiency (optional parameter),
- Identifier for gearbox stiffness constant (optional parameter),
- Identifier for transmission ratio (required parameter),
- File with gear shifting curves: automatic transmission schedule (required for automatic gearbox),
- Number of gears (required parameter),
- Gear ratio for each of the gear (i12). Negative value is assigned for the reverse gear (required parameters),
- Numerical values for efficiency (f) and stiffness (K, Nm/rad) for each of the gears (required parameters if the corresponding identifiers are assigned in Figure 1.47).

The default identifiers are assigned automatically, Sect. 1.4.5.1. *"Force element as gearbox model"*, p. 1-53. If the user set non-standard identifiers, he should select them from the identifier list after double clicking on the corresponding box of the table.

1.4.5.4. File with gear shift schedule

The *.gss files (gear shift schedule) are used for gear shifting when the automatic gearbox is used as well as in the model of the longitudinal speed control. The file contains gear upshift and downshift curves for vehicle or engine rotor speed versus throttle position.

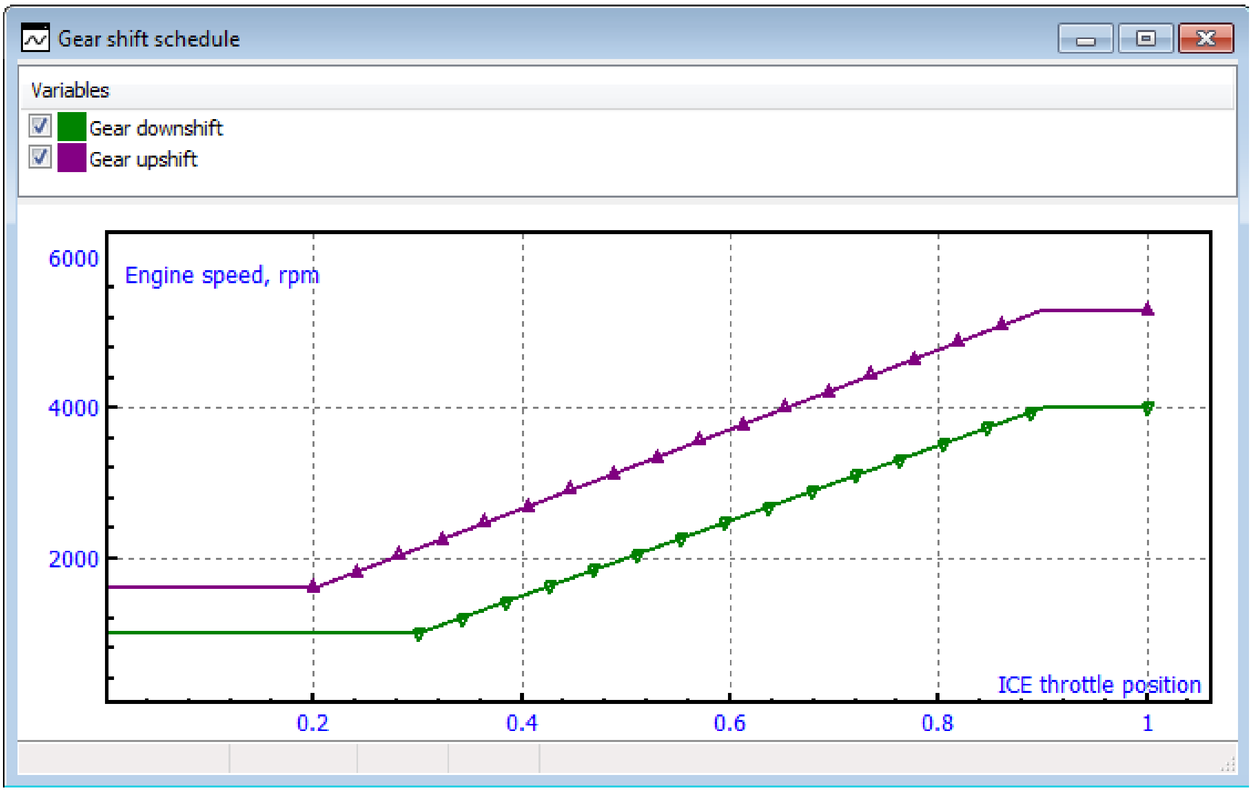


Figure 1.48. Upshift and downshift curves for engine rotor speed

Gear shift in dependence on the engine speed

In this simplified case, it is necessary to input two curves: the upshift speed (lower curve) and the downshift speed (upper curve) in dependence on the ICE throttle position, Figure 1.48.

Gear shift in dependence on the vehicle speed

The gear shift schedule in this case contains 2N-2 curves where N is the number of gears. The odd curves correspond to the gear downshift, and the even ones correspond to the gear up-shift, Figure 1.49.

Curves in the file must be strictly ordered in the growth of speed. For example, for a gearbox with four gears the curves sequence must be 2–>1, 1–>2, 3–>2, 2–>3, 4–>3, 3–>4.

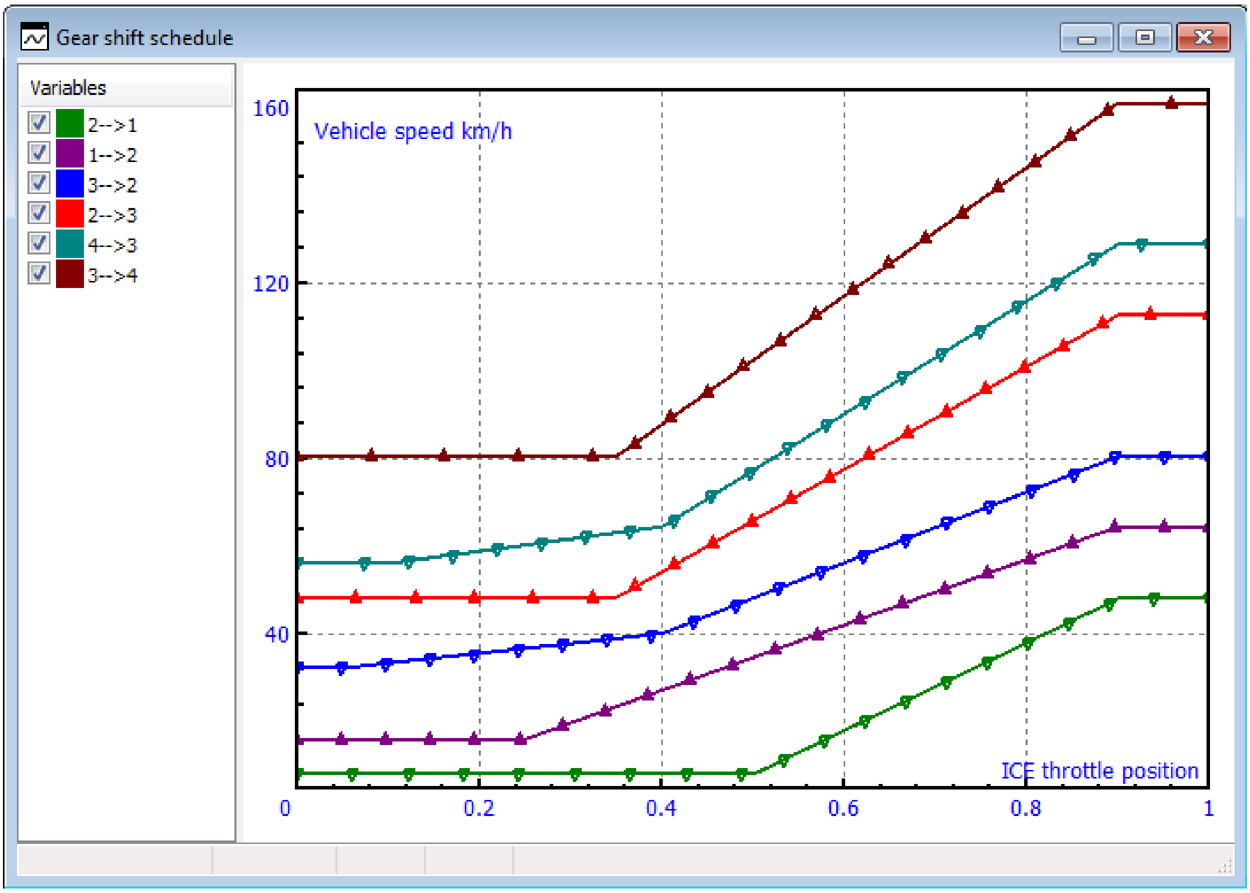


Figure 1.49. Upshift and downshift curves for car speed

The *.gss files are made with the tool **Gear shift schedule** on the **Tools** tab, Figure 1.50. The curves are entered with the curve editor, Figure 1.51.

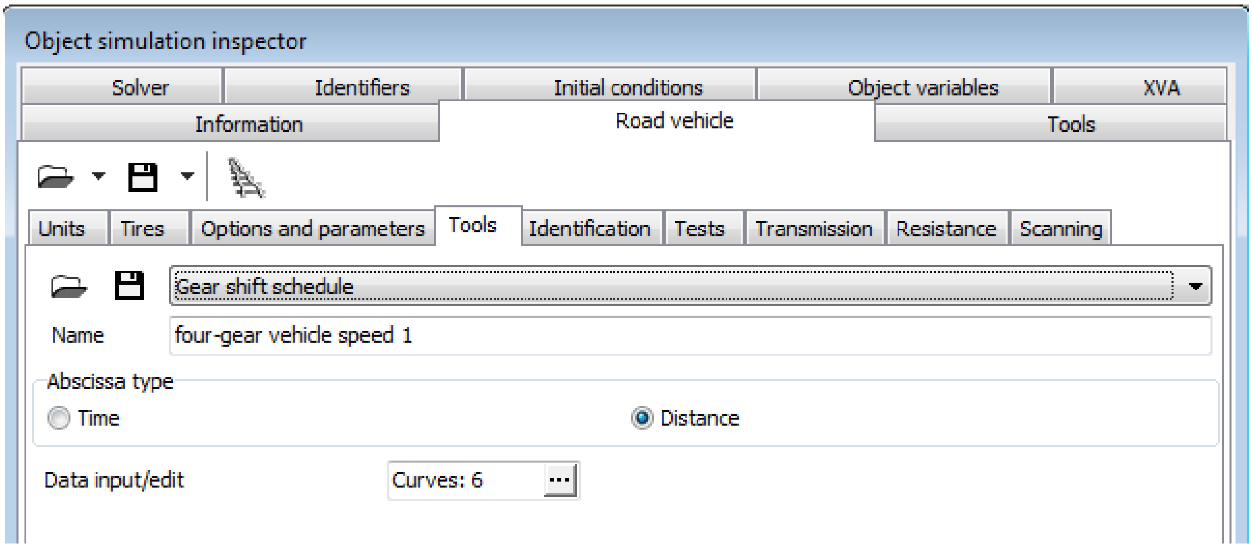


Figure 1.50. Tool for making a file with the gear shift schedule

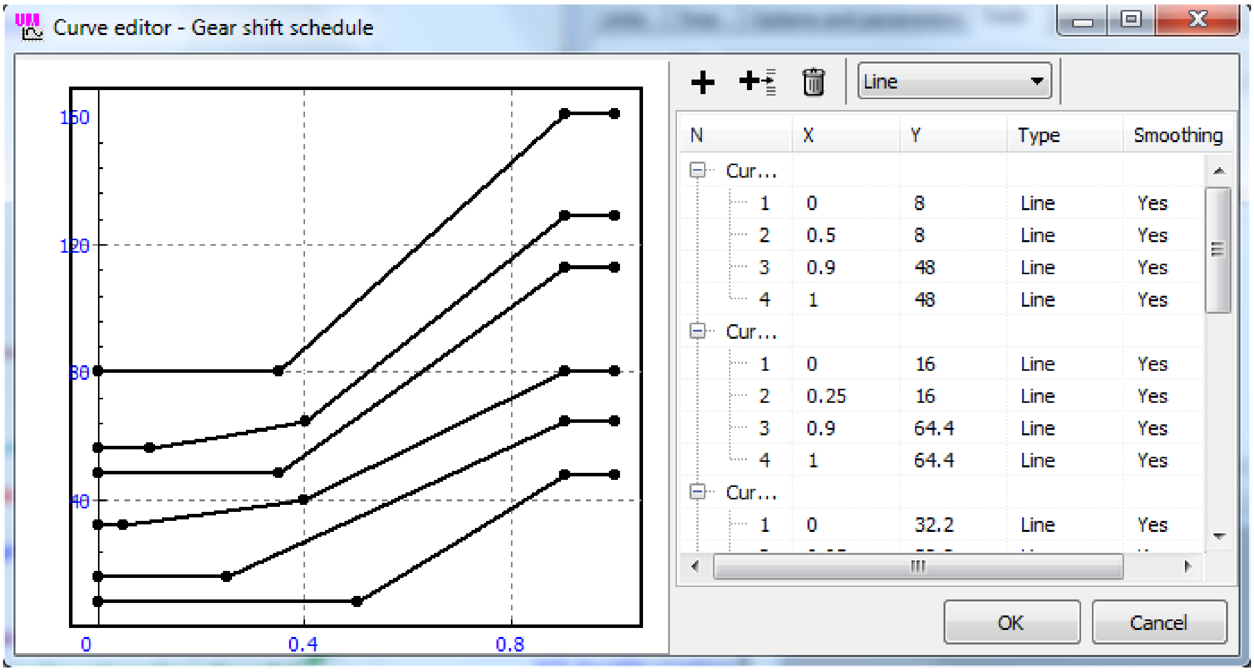


Figure 1.51. Making gear shift schedule

1.4.6. Anti-lock braking system (ABS)

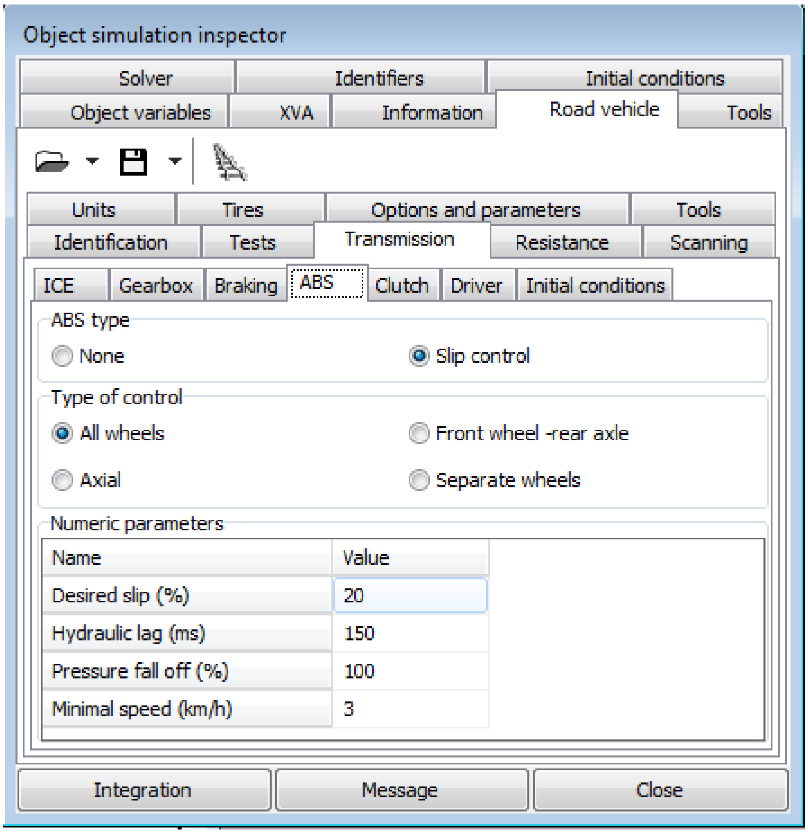


Figure 1.52. ABS parameters

A simplified model of anti-lock braking system "Sliding control" is implemented in UM. Let s* be a desired value of the wheel sliding at braking. Consider a case when the individual control for each of the wheel is realized. The ABS controlled parameter for *i*-th wheel is

$$k_i = \begin{cases} 1, s \le s^* \\ 1 - k^*, s > s^* \end{cases}$$

where $k^* \in [0,1]$ is the parameter of the pressure fall-off for the active ABS. This means, the ABS is activated if the wheel sliding is greater than s* and disabled when it is less that s*.

The braking torque is computed by the formula

$$M_i = M_{i0} f_i,$$

where M_{i0} is the given value of the braking torque according to the brake pedal stroke (see Sect. 1.4.3 *Automotive brake model*), and f_i if the reducing factor satisfying the equation

$$T\dot{f}_i + f_i = k_i.$$

Here *T* is the hydraulic delay.

The described model is easily generalized on other types of ABS shown in Figure 1.52.

Parameters of the model are shown in Figure 1.52. ABS is disabled if the speed is less than the minimal one.

1.4.7. Transmission control

For methods are used in UM for transmission control of road and tracked vehicles.

- External control with the **UM Com Server** library. This type of control is used mainly fin development of road vehicle simulators. See details in the user's manual, Chapter 20, file 20_um_com.pdf, Sect. *Interfaces for simulator of road vehicles*.
- Control panel in the **Simulator** test of road vehicle.
- Identifier control.
- Automatic control with internal driver model (in development).

1.4.7.1. Control panel

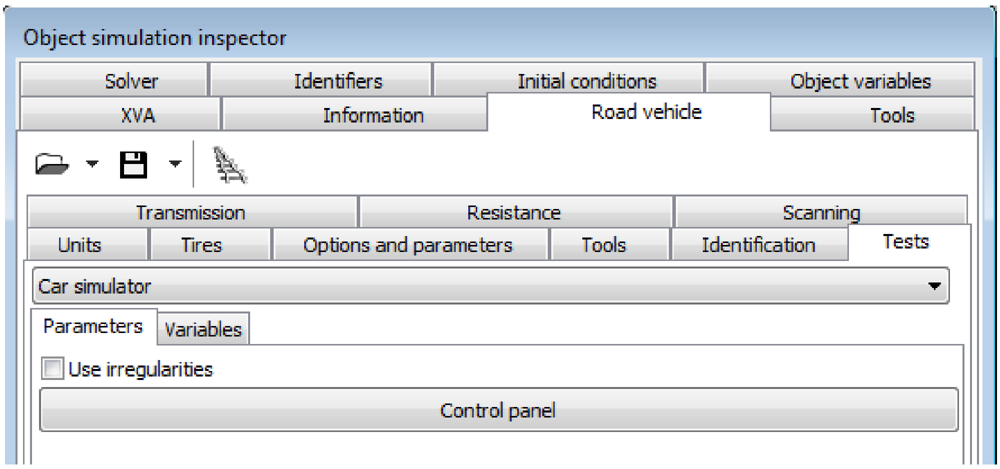


Figure 1.53. Test type: Simulator

When the **Car simulator** test type is selected, the control panel becomes available after click on the corresponding button, Figure 1.53. The panel is used for the mouse control of the road and tracked vehicles, Figure 1.54, Figure 1.55. The user can turn on and off the ignition, brakes, steer the vehicle and so on.

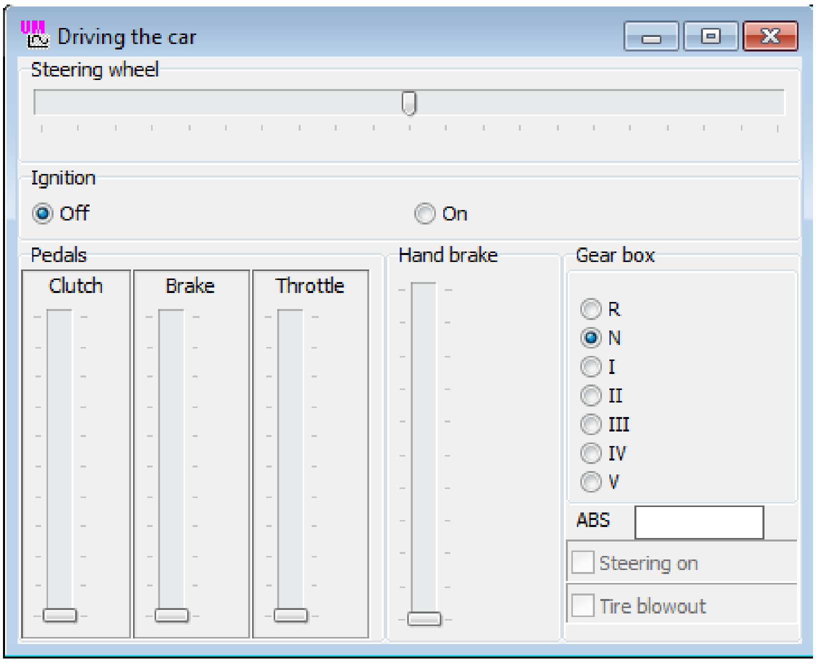


Figure 1.54. Control panel for a car

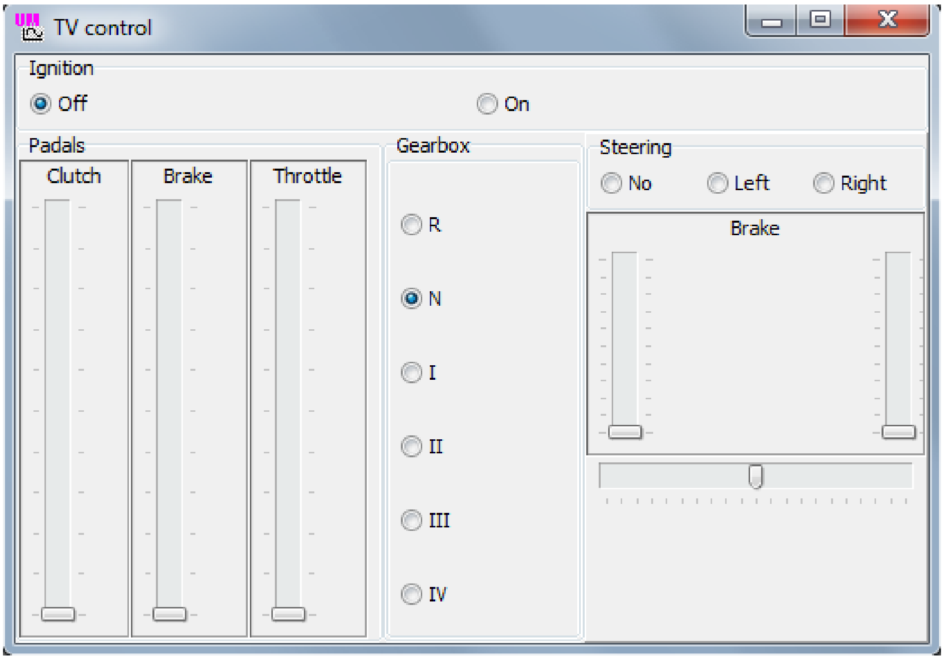


Figure 1.55. Control panel for a tracked vehicle

1.4.7.2. Identifier control

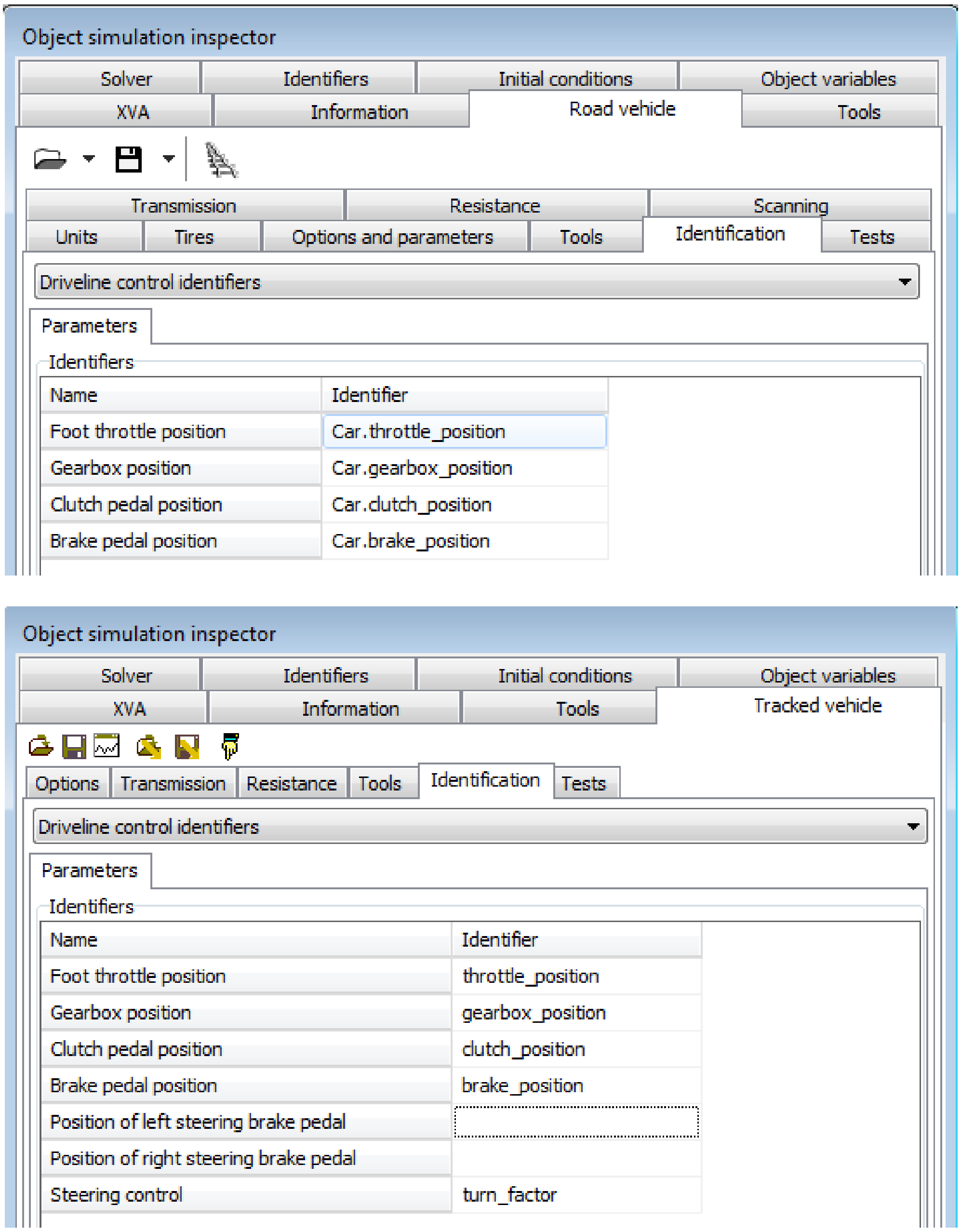


Figure 1.56. Identifiers for control of road and tracked vehicles

A set of identifiers is used for the control of a road or a tracked vehicle. These identifiers must be added to the list of identifiers in UM Input. If a TV transmission from the UM database is used, the identifiers are added automatically.

The list of identifiers for the transmission control is (the default identifiers are shown in Figure 1.56):

- Throttle position, the numerical value should belong to the interval [0, 100%]
- Gear position: -1 (R), 0 (N), 1 (the first gear) and so on
- Clutch pedal position, interval [0, 1]
- Brake pedal position, the value is zero or positive. The brake torque is zero for the null value of the identifier. Identifier value 1 corresponds to 8kN force acting on the pedal. The

braking torque is the product of the force acting on the pedal and the gain factor specified by the user.

Additional identifiers for the TV control:

- Positions of the left and rights brakes (if presented).
- Steering parameter, value belongs to the interval [-1, 1]. Zero value corresponds to the straight motion, negative and positive values correspond to the left and right turn. In the case of a hydrostatic drive the parameter is changed continuously otherwise it takes the discrete values -1, 0, 1.

The control is created with the tool **Identifiers | Identifier control** in the Object simulation inspector, Figure 1.57.

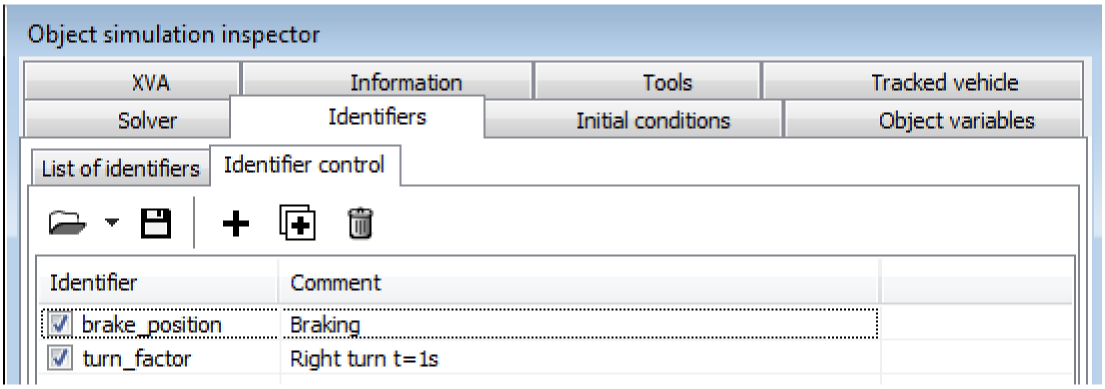


Figure 1.57. List of indentifier controls

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