TORQUE CONVERTER MODELLING FOR ACCELERATION SIMULATION

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ABSTRACT – Simulation plays an important role in optimising the performances of the drive line of the off-road vehicles. The models of the engine, torque converter, transmission and running gear are assembled into a large model used for estimation of the vehicle acceleration performances.
The paper presents a proposal for modelling of the torque converter dimensionless characteristics using segmented continuous function taking as variable the inverse of the kinematic ratio. There are analysed the variants of the characteristics offered by the literature as well as the implementation of the torque converter model into the vehicle model.

1. INTRODUCTION

The simulation of the acceleration and of the tractive performances of vehicles plays an important role in optimization of the driveline and running gear. The accuracy of the simulation depends dramatically of the models used for engine and transmission.
Usually, the steady state characteristics of the engine are known from the technical specifications provided by the supplier in terms of variation of the effective power and torque versus engine speeds or, at least, the values of the maximum power and maximum torque and the speeds of the engine.

Contrary, the technical specification of the transmission provide the only the ratios of the gearbox and, sometimes, the maximum torque ratio of the torque converter. The modelling of the driveline imposes the usage of the torque converter characteristics (the variation of the efficiency, torque ratio and absorbed torque versus slippage) (1).

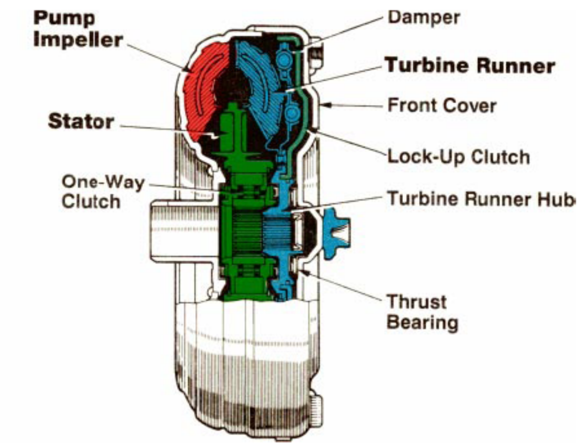

Figure 1 Torque converter components

The effort made for modelling the torque converter characteristics have focussed on the usage of cubic Spline for fitting data (2) and on the estimation of the torque converter characteristics taking into account the geometry of the pump and turbine rotors (3), (4). Nevertheless, the paper dealing with the vehicle performances or with the analysis of the automatic transmissions include models based on the fitting of the torque characteristics provided by the supplier (1), (6) or using appropriate tools implemented in specialized software such as Dymola (7).

Very often, the designers have to roughly estimate the longitudinal dynamics of the vehicles for different configuration of the driveline based on commercial presentations of the engines and transmissions; such presentations usually provide only the ratios of the gearbox and the maximum torque ratio of the torque converter.

2. DIMENSIONLESS CHARACTERISTICS OF TORQUE CONVERTER

The dimensionless characteristic of the torque converter represents graphically the variation of the efficiency and torque transformation ratio versus the inverse of the speed ratio, defined as:

$$i'_h = \frac{n_{turbine}}{n_{impeller}}, \quad i'_h \in [0,1]; \; K_h = \frac{n_{turbine}}{n_{impeller}} \tag{1}$$

Figure 2 presents a typical dimensionless characteristic of torque converter (Allison TC-418)/

It may observe that the efficiency curve presents two typical sections:
1. A parabolic shape crossing the origin.
2. A linear shape followed by an abrupt drop for high values of the inverse of the speed ratio.

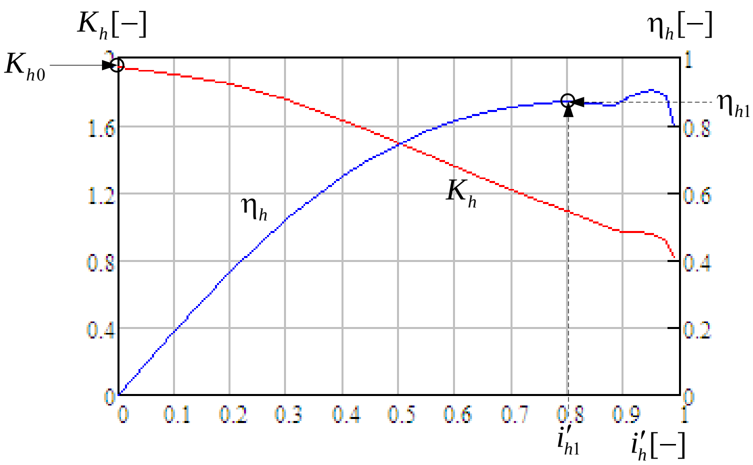

Figure 2 Typical torque converter dimensionless characteristics

The evolution of the torque transformation ratio shows two sections too, the shapes being quasi-linear.

The two sections mentioned above correspond to the two stages of torque converter work: as torque converter and as hydro-coupler, respectively; the shifting between the two stages is automatic due to the release of the stator (see **Figure 1** for the components of the torque converter) for K_h □ 1.0

The efficiency of the torque converter is defined as the ratio of the output power by the input power:

$$\eta_h = -\frac{P_{input}}{P_{output}}. \tag{2}$$

Using the following definitions:

$$P_{input} = M_{impeller} \cdot \omega_{impeller}; \; P_{output} = M_{output} \cdot \omega_{output}$$

as well as the relations (1), from the relation (2) finally results:

$$\eta_h = K_h \cdot i'_h \tag{3}$$

It is reasonable to model the efficiency curve according to the shapes of the two sections described above. The following conditions apply in modelling the efficiency curve:
A. The curve must cross the origin:

$$\eta_h(0) = 0 . \tag{4}$$

B. The maximum of the efficiency occurs for the value of the inverse of the speed ratio noted i'_{h1}:

$$\eta_h(i'_h) = \eta_{h1} ; \tag{5}$$

$$\left. \frac{\mathrm{d}\eta_h(i'_h)}{\mathrm{d}i'_h} \right|_{i'_h = i'_{h1}} = 0 . \tag{6}$$

C. The maximum value of the torque transformation ratio, noted K_{h0} may be calculated using the relation (3):

$$K_{h0} = \lim_{i'_h \to 0} \frac{\eta_h(i'_h)}{i'_h} . \tag{7}$$

Imposing the conditions A and B, three parameters may be calculated, while using all the 3 conditions allows calculating 4 parameters. Consequently, it is possible to model the efficiency first section as a polynomial 2$^{\text{nd}}$ order or as polynomial 3$^{\text{rd}}$ order, respectively; both cases will be analysed further.

For the modelling of the efficiency as a 2$^{\text{nd}}$ order polynomial, the general form of the expression is:

$$\eta_h(i'_h) = A_2 \cdot (i'_h)^2 + B_2 \cdot (i'_h) + C_2 , \tag{8}$$

and from the relation (4) results immediately: $C_2 = 0$.

The relations (5) and (6) give the following simultaneous equations:

$$\begin{cases} A_2 \cdot (i'_{h1})^2 + B_2 \cdot (i'_{h1}) = \eta_{h1} \\ 2A_2 \cdot (i'_{h1}) + B_2 = 0 \end{cases} \tag{9}$$

Solving (9) with respect to the parameters A_2 and B_2 permits to obtain the final expression of the efficiency:

$$\eta_h(i'_h) = \frac{\eta_{h1}}{i'_{h1}} \cdot i'_h \cdot \left(2 - \frac{i'_h}{i'_{h1}} \right) \tag{10}$$

For the second case, considering the modelling as a 3$^{\text{rd}}$ order polynomial, following similar steps of calculation, the final expression results:

$$\eta_h(i'_h) = \left(K_{h0} \cdot i'_{h1} - 2 \cdot \eta_{h1} \right) \cdot \left(\frac{i'_h}{i'_{h1}} \right)^3 + \left(3 \cdot \eta_{h1} - 2 \cdot K_{h0} \cdot i'_{h1} \right) \cdot \left(\frac{i'_h}{i'_{h1}} \right)^2 + K_{h0} \cdot \left(\frac{i'_h}{i'_{h1}} \right) \tag{11}$$

The typical values for the torque converters utilized for the hydrodynamic transmissions of heavy vehicles are:

$$\eta_{h1} = 0.84...0.88;\; i'_{h1} = 0.80...0.86\,.$$

The value of the maximum torque transformation ratio is provided by the torque converter producer and represents a key factor in selecting the appropriate type.

For the hydro-coupler section the following linear relation is proposed:

$$\eta_h(i'_h) = 0.95\cdot i'_h \qquad (12)$$

Finally, the all sections are described by one of the following functions:

$$\eta_h(i'_h) = \max\left\{\frac{\eta_{h1}}{i'_{h1}}\cdot i'_h\cdot\left(2-\frac{i'_h}{i'_{h1}}\right),\quad 0.95\cdot i'_h\right\}; \qquad (13)$$

$$\eta_h(i'_h) = \max\left\{\left(K_{h0} i'_{h1} - 2\eta_{h1}\right)\left(\frac{i'_h}{i'_{h1}}\right)^3 + \left(3\eta_{h1} - 2K_{h0}\cdot i'_{h1}\right)\left(\frac{i'_h}{i'_{h1}}\right)^2 + K_{h0}\cdot\left(\frac{i'_h}{i'_{h1}}\right),\quad 0.95\cdot i'_h\right\} \qquad (14)$$

The graphical representation of the actual curve and of the modelled curves is shown in Figure 3; it may observe that for the normal region of the torque converter ($i'_h = 0...0.95$) both modelled curves provide an acceptable accuracy.

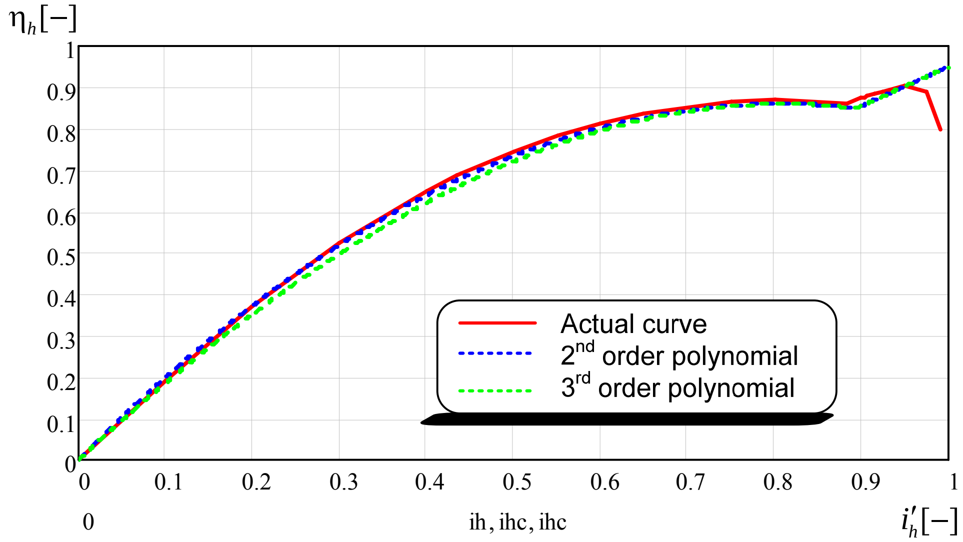

Figure 3 Comparison of actual curve with modelled curves

Further comparison is needed in order to conclude what the best solution is. The torque transformation ratio is calculated based on relation (3) in which one of the relations (10) or (11) is inserted:

$$K_h(i'_h) = \frac{\eta_h(i'_h)}{i'_h} = \frac{\eta_{h1}}{i'_{h1}}\cdot\left(2-\frac{i'_h}{i'_{h1}}\right) \qquad (15)$$

$$K_h(i'_h) = \frac{K_{h0}\cdot i'_{h1} - 2\cdot\eta_{h1}}{i'_h}\cdot\left(\frac{i'_h}{i'_{h1}}\right)^3 + \frac{3\cdot\eta_{h1} - 2\cdot K_{h0}\cdot i'_{h1}}{i'_h}\cdot\left(\frac{i'_h}{i'_{h1}}\right)^2 + \frac{K_{h0}}{i'_h}\cdot\left(\frac{i'_h}{i'_{h1}}\right) \qquad (16)$$

The graphical representation of the torque transformation ratio is presented in **Figure 4**. It results that the 2nd order polynomial does not provide an accurate estimation of the maximum torque transformation ratio but the precision of the model is fully acceptable above $i_h' = 0.2$. Contrary, the use of the 3rd polynomial curve assure the accurate estimation of the maximum torque transformation ratio but the approximation becomes acceptable only above $i_h' = 0.5$.

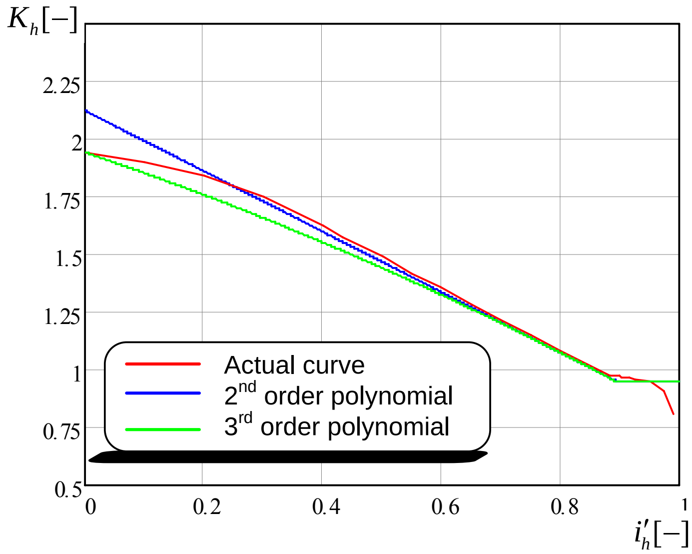

Figure 4 Torque transformation ratio modelling

A detailed analysis of the modelling errors is graphically presented in **Figure 5**.

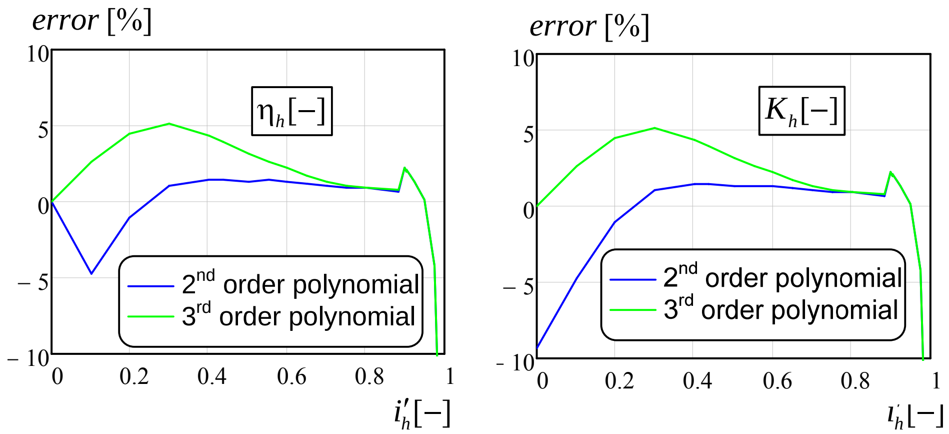

Figure 5 The estimation of the errors of modelling

3. ABSORPTION CHARACTERISTICS OF TORQUE CONVERTER

The torque converter "absorbs" the torque provided by the engine in variable amount depending on the slip between the impeller and the turbine. The absorbed torque is calculated using the relation:

$$M_{impeller} = \frac{n^2}{K(i_h')}. \qquad (17)$$

The typical evolution of the absorption coefficient, noted K, is presented in **Figure 6**. Usually, the producers of torque converters provide only the value of "stall absorption coefficient", noted K_0. In order to approximate the variation of the absorption coefficient, the following mathematical expression is proposed based on the observation that $K(0) = K_0$:

$$K(i_h') = K_0 + \frac{e^{i_h'}}{1 - i_h'}. \qquad (18)$$

Using the relation (18) the graph presented in **Figure 6** has resulted. The graph includes also the relative error due to the modelling. It may be emphasised that the relative errors are below 2% for inverse of speed ratio up to 0.4; the relative errors increase up to about 7% for i'_h □ 0.75 , but increase dramatically at the edge of the hydro-coupling section.

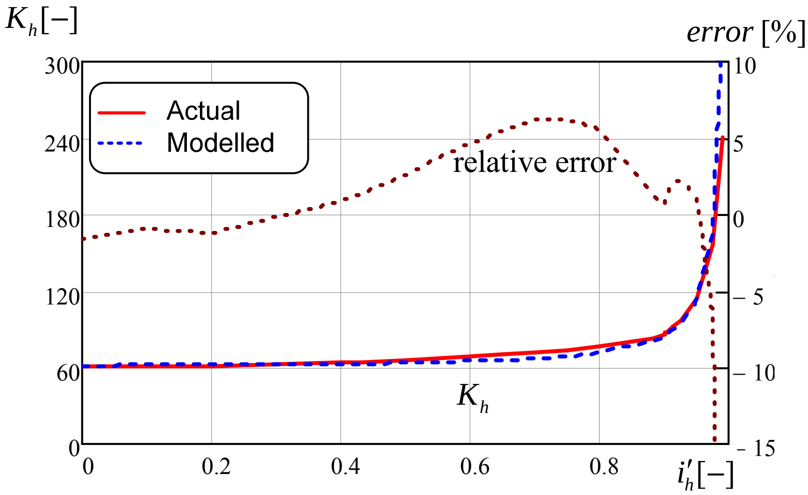

Figure 6 The absorption characteristic of the torque converter

In order to better evaluate the accuracy of the modelling, the tractive characteristics of a 14 tonnes vehicle have been calculated. The powerpak includes a Diesel engine delivering 195 kW at 2300 rpm and a planetary gear box with 6 ratios for forward movement. The methodology for calculating the tractive coefficient is detailed in (1), and the results are included into the graph presented in Figure 7.

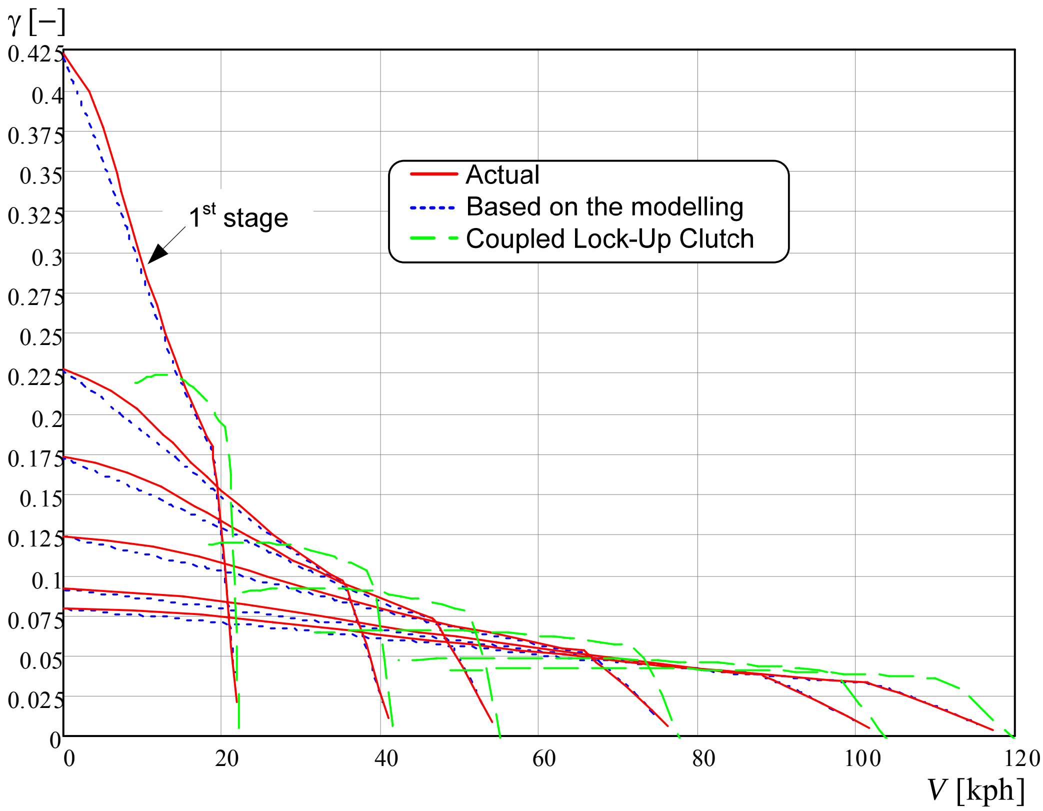

Figure 7 Tractive coefficient characteristics

From Figure 7 results two main aspects which are influenced by the errors due to the modelling of the torque characteristics;

- the estimated speed for movement on slopes;

* the setting of shifting speeds.

The first aspect is detailed in Figure 8; the resulted error is as high as 0.52 kph repre-sentting about 5.1%. This error is inferior to those of estimation of the rolling resistance or the driveline efficiency. Consequently, the modelling of the torque converter characteristics does not affect significantly the estimation of the speed the vehicle can achieve climbing the slope.

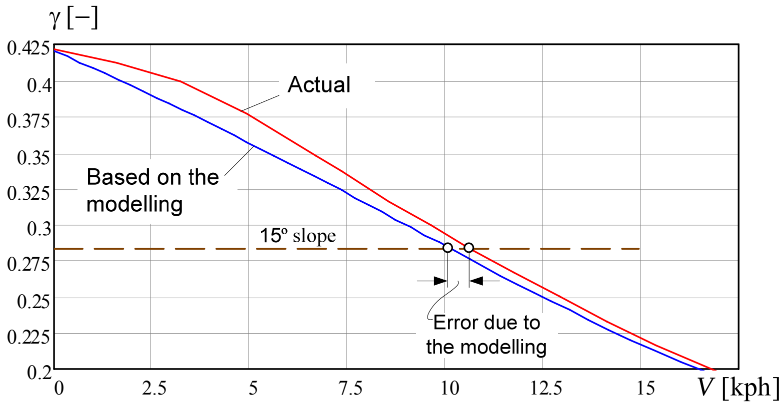

Figure 8 Detail for emphasising the error in the estimation of the speed for climbing 15º slope

The second relevant aspect regarding the influence of the torque converter characteristics modelling refers to the estimation of the speeds for optimal shifting the ratio of the gear box. The following shifting sequencing was adopted: 1st stage with torque converter (TC) working, then the lock-up clutch is coupled end the next shift occurs at the speed which corresponds to the nominal speed of the engine (2300 rpm).

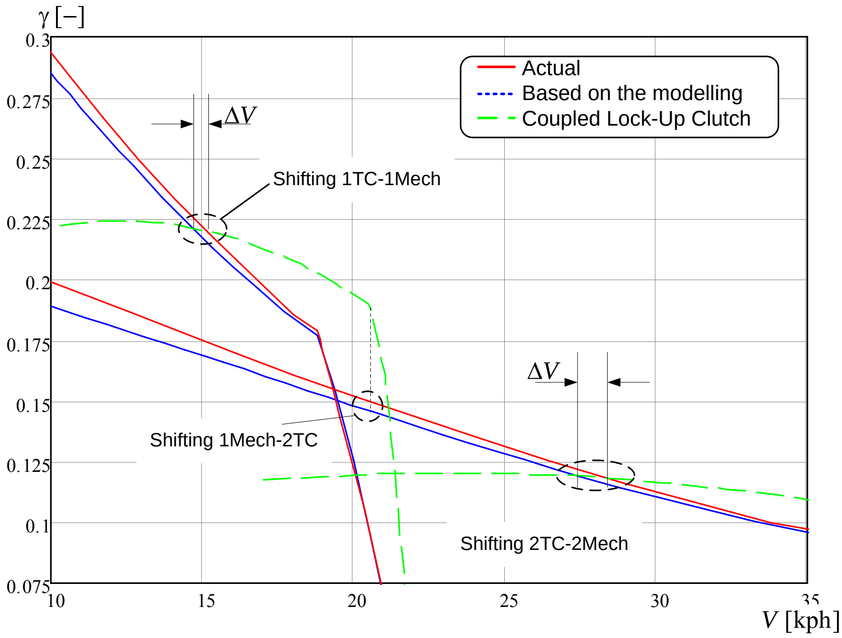

Figure 9 Shifting speed detailed

The process is detailed in **Figure 9** with emphasis of the errors due the modelling of the torque converter modelling. It may conclude that these errors do not affect the acceleration process estimation of the vehicle.

For the stages 3 to 6, the tractive coefficient characteristics presented in Figure 7 indicates that is more economic to use only mechanical regime of the hydrodynamic transmission (the lock-up clutch remains coupled).

4. CONCLUSIONS

The proposed modelling method proposed use minimal data provided by the torque converter producers: the maximum torque transformation ratio and the stall absorption coefficient.

Based on these inputs, the proposed modelling methods allow the estimation of the efficiency, torque transformation ratio and absorption coefficient characteristics.

The estimation of the errors due to the modelling of the torque characteristics as well as of the influences on the tractive coefficient characteristics has demonstrated that their level is fully acceptable. Consequently, the proposed modelling is useful as a first estimation of the following aspects which represent a real interest in design of powerpack and driveline of heavy vehicles:

- estimation of the matching of the engine and torque converter;
- estimation of the tractive coefficient characteristics which allow the checking of the difficult movement situation such as the motion on slopes;
- estimation of the acceleration performances of the vehicle.

Thus, the proposed methods may be applied to obtain first estimation of the vehicle's performances within the decision making process for adopting the appropriate configuration of the powerpack based on commercial data regarding the torque converter.

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