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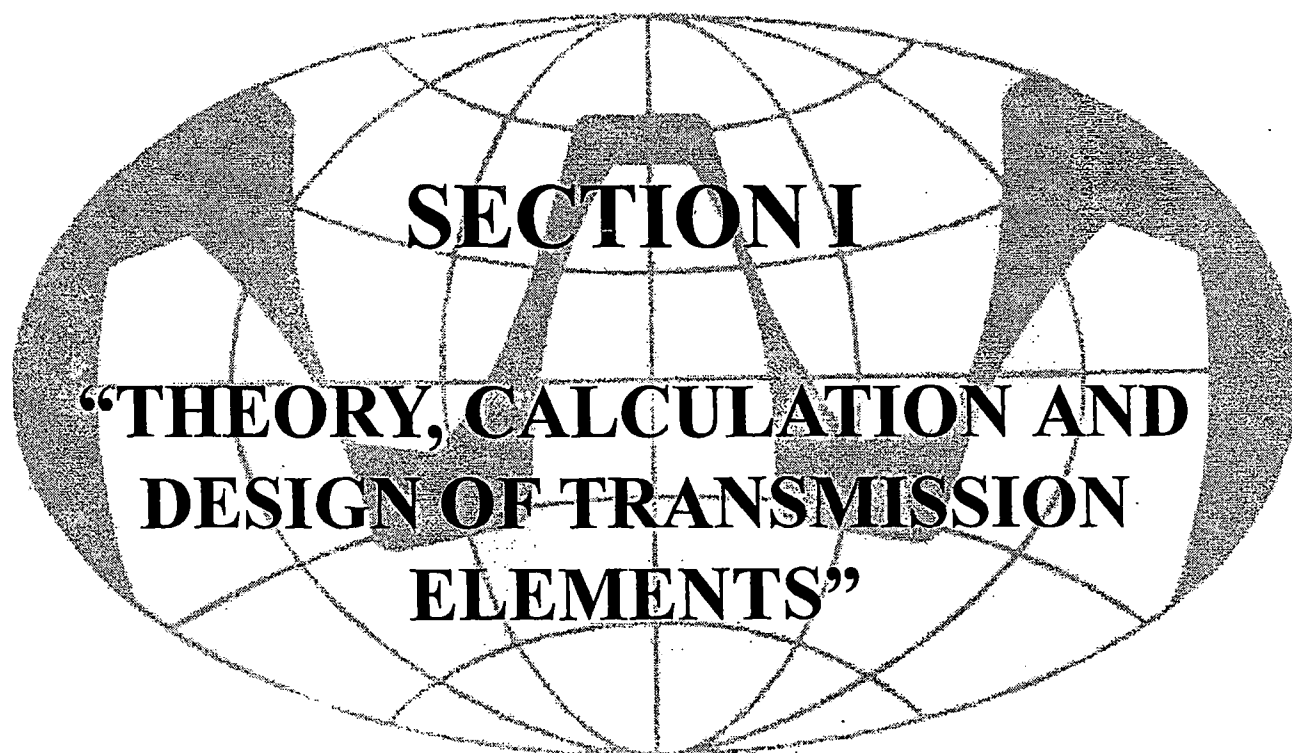
**INTERNATIONALE TAGUNG  
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**МЕЖДУНАРОДНА КОНФЕРЕНЦИЯ  
“ТЕХНИКА ПРИВОДОВ’03”**

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**МЕЖДУНАРОДНА КОНФЕРЕНЦИЯ  
“ТЕХНИКА НА ЗАДВИЖВАНИЯТА’03”**



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**Prof. Dr. Eng. Kiril Arnaudov**  
**President of the Balkan Association**  
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# PRINCIPLES OF AUTOMATIC TRANSMISSION MODELING AND SIMULATION

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**Abstract:** Automatic transmission is one of the most complex subsystems of motor vehicle. It consists of three parts i.e. electronic control unit, hydraulic control unit and mechanical components. The main objective of application of automatic transmission in passenger cars is to reduce strokes during a gearshift i.e. to achieve, as much quality in gear shifting as it is possible. In most cases, experimental method for investigation of quality and parameters of gearshift, as well as development of control algorithms are used. However, experiment demand great material means and time, so it is convenient to develop a mathematical model that can simulate the whole transmission, save time and money and confirm experimental results.

A dynamic model of automatic transmission system that has the objective to simulate the real operating conditions of all system components is presented in this paper.

**KEYWORDS:** AUTOMATIC TRANSMISSION, MATHEMATICAL MODEL, GEAR SHIFT

## 1. INTRODUCTION

Automatic transmissions with torque converter are widely used for passenger cars because they can provide advantages such as: smooth start, damping characteristics for the disturbance of output torque, easy operation and safety. The development process of modern-time vehicles and their intelligent high-tech components is characterized by the need to minimize the time and costs from the vision of a new automobile to its serial production.

Modern cars can be seen as heterogeneous systems, which are composed of elements from different physical domains. For this reason a general model of a road vehicle may include components such as: mechanic components, hydraulic or electronic control.

In this study is proposed dynamic model of a power transmission system for gear ratio changed. The automatic transmission is modeled in detail for each component. The simulation was performed on the basis of the dynamic modeling with MATLAB-Simulink.

## 2. MATHEMATICAL MODEL

The main parts of automatic transmission are torque converter, planetary gear sets, hydraulic and electronic control units. To investigate the shifting characteristic through numerical simulation, dynamics of engine that generate their transmitted input torque and road load which react a driving output shaft torque are necessary in addition to the basic parts [2], [4], [5].

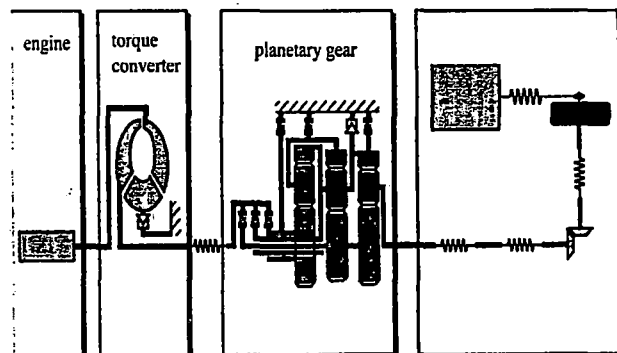


Figure 1. Schematic description of the powertrain

## 2.1 Mathematical model of torque converter

A torque converter is a common part in automatic gear vehicles. The main purpose of the converter is to multiply the output torque to the transmission during starting conditions. Also provide other benefits such as: stepless variation in torque and speed, smooth torque transmission between engine and automatic transmission, absorption of vibration isolation and torque peaks. The torque converter consists of three parts, a impeller (P), a turbine (T) and a reactor (R), respectively, using oil as the working fluid for torque transmission. The impeller is connected to the engine shaft and the turbine is connected to the gearbox.

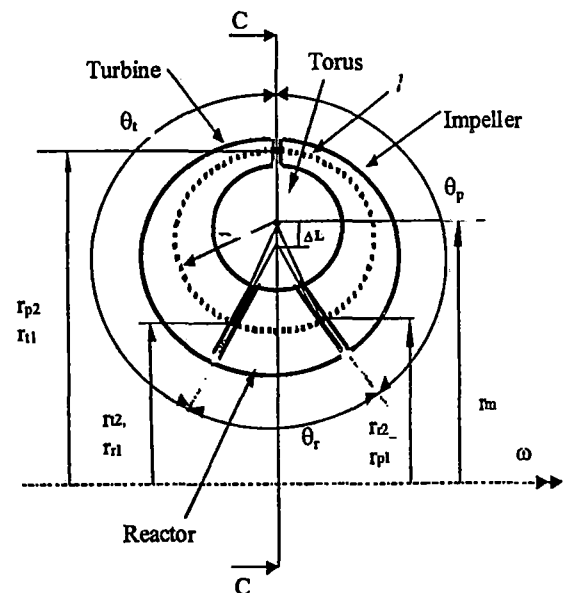


Figure 2. Schematic description of a three element torque converter (P-impeller, T-turbine, R-reactor) (1-inlet, 2-outlet)

Presented model includes the effects of torque converter fluid inertia along the flow streamlines as well as the more standard angular momentum effects and steady state effects of flow at shock losses. The underlying assumptions are: the cross-section net torus flow area is constant, the spacing between the torques

converter parts is neglected and the blade thickens effects are neglected and thermal effects are not considered [1], [3].

Angular momentum equation is:

$$M_p = \int_A (r \times \vec{e}) \cdot \rho \cdot (\vec{e} \times dA) + \frac{\partial}{\partial t} \int_V (r \times \vec{e}) \cdot \rho \cdot dV \quad (1)$$

$$\int_A (\vec{F} \times \vec{e}) \cdot \rho \cdot (\vec{e} \times dA) = (\vec{F} \times \vec{e})_2 \cdot \dot{m}_2 - (\vec{r} \times \vec{e})_1 \cdot \dot{m}_1 \quad (2)$$

where is:  $\vec{e} = \vec{u} + \vec{w}$  - absolute fluid velocity,  $\rho$  - fluid density,  $A$  - net flow area,  $r$  - radius,  $V$  - volume,  $\dot{m}$  - mass flow,  $I$  - moment of inertia,  $V$  - volume flow,  $S$  - static torque,  $\omega$  - angular velocity.

$$\frac{\partial}{\partial t} \int_V (\vec{F} \times \vec{e}) \cdot \rho \cdot dV = \omega_p \cdot \int_V r^2 \cdot \rho \cdot dV + \rho \cdot S_p \cdot \frac{dV}{dt} = I_p \cdot \dot{\omega}_p + \rho \cdot S_p \cdot \frac{dV}{dt} \quad (3)$$

Energy balance equation is:

$$E = \frac{1}{2} \cdot (I_p \cdot \omega_p^2 + I_i \cdot \omega_i^2 + I_r \cdot \omega_r^2 + \rho \cdot L_f \cdot \dot{V}^2) \quad (4)$$

$$+ \rho \cdot V \cdot (S_p \cdot \omega_p + S_i \cdot \omega_i + S_r \cdot \omega_r)$$

where:  $L_f = \int \frac{dl}{A \cdot \sin^2 \beta}$  - equivalent fluid inertia length,  $\beta$  - blade angles.

There are two major contributors to power loss i.e. the flow losses ( $\Delta p_{fr}$ ), and the shock losses ( $\Delta p_u$ ). The flow loss results from shear stresses in the boundary layer and a contribution due to possible pressure drag:

$$\Delta p_{fr} = \frac{1}{2} \cdot \rho \cdot \sum_j (c_{frj} \cdot w_{frj}^2) \quad (5)$$

where:  $c_{frj}$  - friction coefficient, and  $w_{frj}$  - fluid relative velocities.

The shock losses are due to, in general, non-ideal speed condition at the interface between any two torque converter elements.

$$\Delta p_u = \frac{1}{2} \cdot \rho \cdot \sum_j c_{uj} \cdot v_{uj}^2 \quad (6)$$

where:  $c_{uj}$  - shock loss coefficient ideally equals one, in practice determined experimentally [3],  $v_{uj}$  - shock velocities.

## 2.2 Mathematical model of planetary gear

The planetary gear set used in the system, and consists sun gear (1), annulus gear (2), pinion (3) and carrier (4). Fig. 3 show the schematic diagram of planetary gear set. The equations for the torque ( $M$ ) and angular velocities ( $\omega$ ) for this gear set as follows [4]:

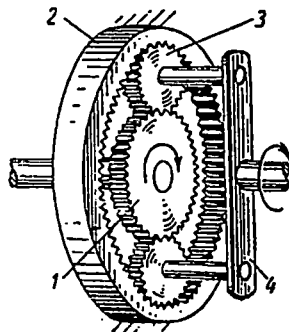


Figure 3. Schematic description of planetary gear set

$$(1 - i_{12}^4) \cdot \omega_4 = \omega_1 - i_{12}^4 \cdot \omega_2$$

$$\omega_4 = \frac{z_1}{z_1 + z_2} \cdot \omega_1 + \frac{z_2}{z_1 + z_2} \cdot \omega_2$$

$$\omega_4 = i_s \cdot \omega_1 + i_u \cdot \omega_2$$

$$i_s = \frac{z_1}{z_1 + z_2}, \quad i_u = \frac{z_2}{z_1 + z_2}$$

$$M_1 = i_s \cdot M_4$$

$$M_2 = i_u \cdot M_4$$

$$\frac{M_1}{M_2} = \frac{z_1}{z_2}$$

$$\frac{M_1}{M_2} = \frac{z_1}{z_2}$$

## 2.3 Mathematical model of friction clutch

The clutch is an element which constraints the motion of power transmitting elements. It is assumed that the torque, transmitted through the clutch plates, and the coefficient of friction at the contact surfaces. The torque transmitted ( $M_c$ ) through the multi-disc clutch can be expressed as follows:

$$M_c = R_{cm} \cdot F_c \cdot \mu \cdot z_c$$

where:  $z_c$  - number of contact area,  $F_c$  - normal force of contact area,  $R_{cm}$  - medium radius of the friction plates,  $R_{c1}$  - min. radius of plates,  $R_{c2}$  - max. radius of friction plates:

$$R_{cm} = \frac{2 \cdot (R_{c2}^3 - R_{c1}^3)}{3 \cdot (R_{c2}^2 - R_{c1}^2)}$$

$\mu$  - is a coefficient of friction and variation of the coefficient can be specified according to the next relation [2]:

$$\mu = 0.1316 + 0.0001748 \cdot |\Delta \omega_c|$$

$$\Delta \omega_c = \omega_1 - \omega_2$$

## 2.4 Mathematical model of electronic control unit

The electronic control unit is composed of sensors, solenoid valves and transmission control unit. It performs the clutch control, shift pattern control, hydraulic control during shift process and so on. The transmission control unit calculates the value of parameters on the basis of data from sensors and generates the signal to operate the solenoid valves.

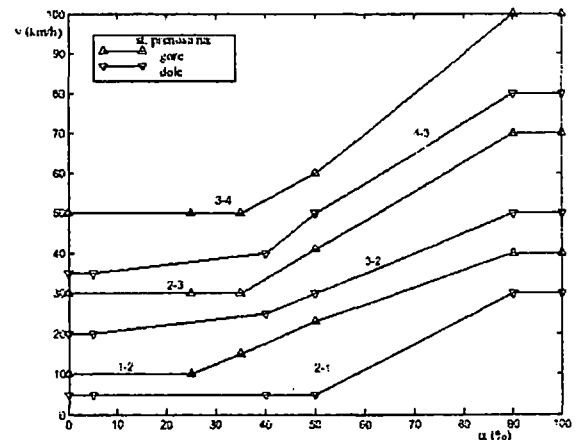


Figure 4. Shift schedule

In Fig. 4 is presented shift schedule map which, for a given throttle position in a given gear, vehicle speed at which an upshift or downshift takes place.

## 2.5 Mathematical model of engine

The high frequency vibrations of the engine are not of great interest for investigation gearshift comfort. The engine can therefore be modeled as a rotating rigid body. Its torque describes the drive train excitation caused by the engine:

$$I_m \cdot \omega_m = ((1 - \beta) \cdot M_m - M_p) \quad (13)$$

where is  $\beta$  - coefficient, [2],  $I_m$  - engine moment of inertia,  $\omega_m$  - engine angular velocity and  $M_m$  - engine torque.

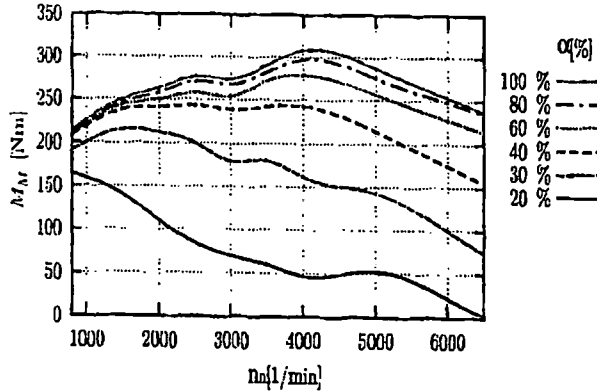


Figure 5. Engine map

The engine subsystem consists of a two-dimensional map that interpolates engine torque as function throttle and engine speed (Fig.5). In accordance with equation 13, the model subtracts the impeller torque, divides the difference by the inertia and then numerically integrates the quotient to compute the engine speed.

The engine in this study is MAN. D 2866 UM power 177kW / 2200 rev/min and torque 865 Nm / 1300 rev/min.

## 2.6 Mathematical model of road load

This submodel represents a simplified model for the output train from the cardan shaft to the wheels and chassis of the vehicle. The wind force ( $F_v$ ), rolling force ( $F_r$ ) and the inclination force ( $F_\alpha$ ) are represent as follows [2]:

$$F_k = G \cdot f_o \cdot (1 + a \cdot v^2) \cdot \cos \alpha \quad (14)$$

$$F_v = c_x \cdot \frac{\rho_v}{2} \cdot A \cdot v^2 = k \cdot \frac{A \cdot v^2}{13} \quad (15)$$

$$F_\alpha = G \cdot \sin \alpha \quad (16)$$

The load torque ( $M_o$ ) results from the wind force, rolling force and the inclination force:

$$M_o = (F_k + F_v + F_\alpha) \cdot \frac{r_d}{i_d} \quad (17)$$

with:  $r_d$  - the wheel radius,  $i_d$  - the transmission ratio of the rear differential,  $\rho_v$  - the air density,  $c_w$  - the air resistance coefficient,  $\alpha$  - the road gradient,  $A$  - the vehicle front surface,  $G$  - the vehicle mass,  $f_o$  - the rolling coefficient.

The final drive ratio ( $i_o$ ), inertia of vehicles ( $I_v$ ) and dynamically varying load ( $M_o$ ) constitute the vehicle dynamics (eq. 18):

$$I_v \cdot \omega_v = i_o \cdot M_2 - M_o \quad (18)$$

## 3. NUMERICAL MODELING

In this paper is modeled automatic transmission which is composed of modules such as: the engine, torque converter, multi-plates disc planetary gear train and shift logic block to control the transmission ratio.

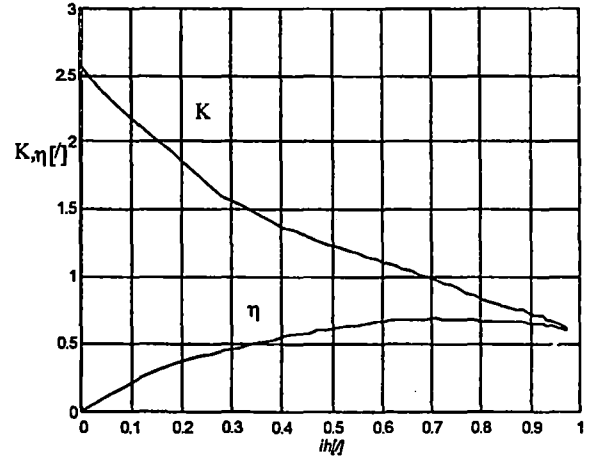


Figure 6. Torque ratio  $K$  and efficiency  $\eta$  as function speed ratio  $i_h$

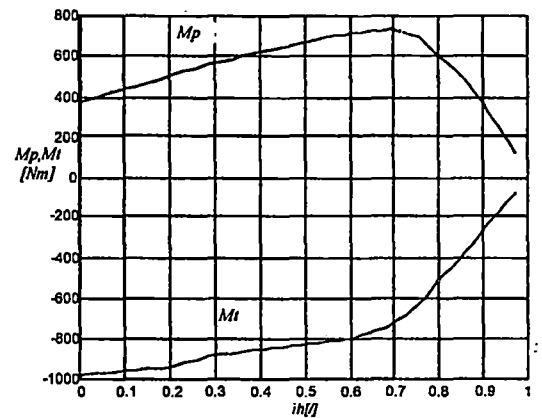


Figure 7. Impeller  $M_p$  and turbine torque  $M_t$  as function speed ratio  $i_h$

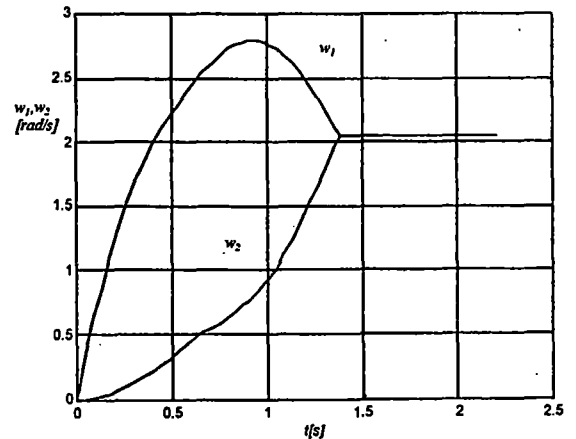


Figure 8. Angular speed of the engagement and disengagement multi-disc clutch

Parameters relevant to steady-state characteristics, represented by torque ratio  $K$  and efficiency  $\eta$  (Fig. 6 and 7) at each given are calculated.

Fig. 8 demonstrates the use of MATLAB and Simulink to simulate a rotating clutch subsystem. Modeling a clutch system is very difficult because of topological changes in the system dynamic during lockup. The clutch system in this example consists of two plates that transmit torque between the impeller and transmission. There is distinct model of operation slipping, where the two plates have difference angular velocities and lockup where the two plates rotate together.

#### 4. CONCLUSION

Major function of automatic transmission is automatic gear shifting in order to change driving torque that is appropriate in view of vehicle speed and drivers intention. Changing gear in automatic transmission results is not only transmitted torque ratio but also rotational speed ratio.

Nowadays, as the performance requirement, especially on shift quality, of automatic transmission become rigorous in recent years, most automatic transmission of passenger car adopt an electronically controlled shifting mechanism to improve fuel economy as well as to enhance the driveability.

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