BALKAN ASSOCIATION OF POWER TRANSMISSIONS

SCIENTIFIC-TECHNICAL UNION OF MECHANICAL ENGINEERING

INTERNATIONAL CONFERENCE
"POWER TRANSMISSIONS`03"

INTERNATIONALE TAGUNG
"ANTRIEBSTECHNIK`03"

МЕЖДУНАРОДНАЯ КОНФЕРЕНЦИЯ
"ТЕХНИКА ПРИВОДОВ'03"

МЕЖДУНАРОДНА КОНФЕРЕНЦИЯ
"ТЕХНИКА НА ЗАДВИЖВАНИЯТА'03"

SECTION I

"THEORY, CALCULATION AND DESIGN OF TRANSMISSION ELEMENTS"

1

ISBN 954-90272-9-5

It is confirmed herewith that the present paper is included in the program of the International Conference "Power Transmissions`03" and is published in the Conference Proceedings.

Hiermit wird bescheinigt, dass dieser Vortrag im Programm der Internationalen Tagung "Antriebstechnik`03" aufgenommen war und im Vortragsband veröffentlicht ist.

Этот доклад был включен в программе Международной конференции "Техника приводов'03" и опубликован в трудах конференции.

K. Arnaudov

Prof. Dr. Eng. Kiril Arnaudov
President of the Balkan Association of Power Transmissions

CONTENTS

SECTION I
"THEORY, CALCULATION AND DESIGN OF TRANSMISSION ELEMENTS"

I - 14	ON STRAIN AND STRESS STATE OF THE METALLIC MEMBRANES AT FLEXIBLE COUPLING	Prof. Dr. Eng. Dobre G., Dipl. Eng. Dobre D., Dr. Eng. Mirica R.F., Dr. Eng. Sorohan S.	Romania
I - 15	INTERNET-REPRESENTATION OF THE RUSSIAN EXPERTS IN THE FIELD OF TRANSFERS OF GEARING www.gears.ru	d.t.n.,prof. A.V.Verhovsky students N.S. Rams, O.M.Bugaev, S.V.Dubejkovsky, V.V.Naumenko, V.S.Florida	Russia
I - 16	NATURE AND MECHANISM OF MODAL DISPLACEMENT EXCITATION IN GEAR HOUSING WALLS	Snezana Ciric-Kostic, Milosav Ognjanovic	Serbia and Montenegro
I - 17	PARAMETRICAL OPTIMIZATION OF INVOLUTE CYLINDRICAL GEAR DRIVES	Prof. Ph.D. Eng.M.Sc. Nenov P.	Bulgaria
I - 18	FACE GEAR DRIVE WITH CONICAL PINION: DESIGN, GENERATION AND TOOTH CONTACT ANALYSIS	Jose I. Pedrero, Claudio Zanzi	Spain
I - 19	INFLUENCE OF THE PRESSURE ANGLE ON THE EFFICIENCY AND THE SLIDING VELOCITY OF HYPERBOLIC GEAR DRIVES	Ass. Prof. Dr. Eng. García-Masiá C. Prof. Dr. Eng. Pedrero J. I. Ass. Eng. Morillas, J. D.	Spain
I - 20	AN INVESTIGATION ON THE DYNAMICS OF A STEPLESS AUTOMATIC INERTIAL MECHANICAL TRANSMISSION BY MEANS OF THE APPLICATION OF THE SECOND SORT LAGRANGE'S DIFFERENTIAL EQUATIONS	Assistant Prof. Dr. Eng. Petrov D.	Bulgaria
I - 21	THEORETICAL INVESTIGATION OF TRACTION CHARACTERISTIC OF BELT TRANSMISSIONS	Eng. Stojanov S., Assoc. Prof. PhD A. Dobreva	Bulgaria
I - 22	SISTEMATIC APPROACH TO TECHNOLOGICAL LEARNING	Dr. Eng. Koleva N.	Bulgaria
I - 23	DEVELOPMENT OF THE IMPROVED DESIGNS OF SPIROID AND HELICAL SPIROID GEAR UNITS	Prof. Dr. Georgiev A., Eng. Samigullin R., Eng. Kotova I.	Russia
I - 24	ELASTIC-PLASTIC STRESS-STRAIN ANALYSIS OF NOTCHED COMPONENTS UNDER COMBINED SYNCHRONOUS FATIGUE LOADING	Ass. Prof. Dr.-Ing. Savaidis G., Dr.-Ing. Savaidis A.,Dipl.-Ing. Pitatzis N., Ass. Prof. Dr.-Ing. Mihailidis A.	Greece
I - 25	CONTRIBUTIONS TO MODERNISING LARGE SIZE BALL SCREW TRANSMISSIONS	Prof. Dr. Dobre G., Dipl. Eng. Dascaliuc M., Dr. Florin Mirica R.F.1	Romania
I - 26	EINFLUß DER FORM UND DER EINBAULAGE AUF DIE WAHL DER GRÖßE VON UNIVERSAL-SCHNECKENGETRIEBEN	S. Kuzmanović, A. Vulić, R. Topić, A. Miltenović	Serbia and Montenegro
I - 27	ABOUT DYNAMIC OF THE VARIABLE-SPEED DRIVES AS NONHOLONOMIC POWER TRANSMISSIONS	Prof. Dr. Miodrag Zlokolica , Mr. Milan Kostic, Mr. Maja Cavic, Vladimir Miskov	Serbia and Montenegro

I - 73	SYNTHESIS OF SPATIAL GEARINGS BY AID OF MESHING SPACE	Dr-Eng. Lagutin S.A.,	Russia
I – 74	EHD LUBRICATION IN JOURNAL BEARING USING A NON-NEWTONIAN LUBRICANT	Ass. Prof. Dr Eng. Javorova J.G.	Bulgaria
I – 75	RESIDUAL STRESSES PLANE MODELLING FOR I8 PROFILE BEAMS PURE BENDING STRESSES IN THE ELASTO-PLASTIC DOMAIN	Reader Dr. Eng. Bal N., Reader Dr. Eng. Ispas V., Reader Dr. Eng. Julean D.	Romania
I – 76	THE APPLICATION OF SIGNAL FLOW GRAPHS TO THE KINEMATIC ANALYSIS OF THE PLANETARY GEAR TRAINS	Prof.Dr.Emilija Vetadzokoska	Macedonia
I – 77	LARGE GEAR WITH ROLL INTERMEDIATE BODIES OF LONG LIFE	Prof. Dr. tech. sn. Beljev A.	Russia
I – 78	GEOMETRICAL SYNTHESIS OF HIGH-SPEED TRANSMISSIONS WITH CLOSE SYSTEM OF ROTATING LINKS	Prof. Dr. Eng. Syzrantsev V., Ass. Prof. Cand. Eng. Krokhmal N., Eng. Fedchenko E.	Russia
I – 79	ON INFLUENCE OF STRESS CONCENTRATION ON FATIGUE STRENGTH FROM GEAR TOOTH ROOT	Jankovic Miodrag, Maksimovic Stevan	Serbia and Montenegro
I - 80	FATIGUE LIFE PREDICTION OF GEAR TEETH UNDER GENERAL LOAD SPECTRUM	Nikolic V., Maksimovic K, & Maksimovic S.	Serbia and Montenegro
I - 81	A GENERAL PROGRAM FOR AUTOMATED FUNCTIONAL DESIGN OF PLANAR LINK MECHANISMS OF II-nd CLASS	Res.Fell. Yankova T., Kulev K., Ass.Prof.Dr.Eng. Varbanov H., Res.Fell.Eng. Lilov S.	Bulgaria
I - 82	RELIABILITY MODELING AND ANALYSIS OF AN ENGINE LATHE HEADSTOCK GEARBOX	Eng. Kamberov K., Dr. Eng. Todorov G., Prof. Eng. Todorov N.	Bulgaria
I - 83	DEVICE WORKING AS NO-EXTENT VARIATION AND TRANSMISSION OF ROTATION AND COUPLING SYSTEM	Dr.Eng. Peter Stoyanov	Bulgaria

PRINCIPLES OF AUTOMATIC TRANSMISSION MODELING AND SIMULATION

Sustersic V.[1]., MSc, Assoc. Prof. Jovicic N[2], Prof. Babic M.[3], Assoc. Prof. Gordic D[4].

University of Kragujevac, Faculty of Mechanical Engineering,
Kragujevac, Serbia & Montenegro

E-mail: [1] – vanjas@knez.uis.kg.ac.yu
[2] – njovicic@knez.uis.kg.ac.yu
[3] – nastasia@ptt.yu
[4] – gordic@knez.kg.ac.yu

Abstract: *Automatic transmission is one of the most complex subsystems of motor vehicle. It consists of three parts i.e. electronic contro unit, hydraulic control unit and mechanical components. The main objective of application of automatic transmission in passenger cars i. 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, sav 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 itheir 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 us: 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 basic of the dynamic modeling with MATLAB-Smiling.

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].

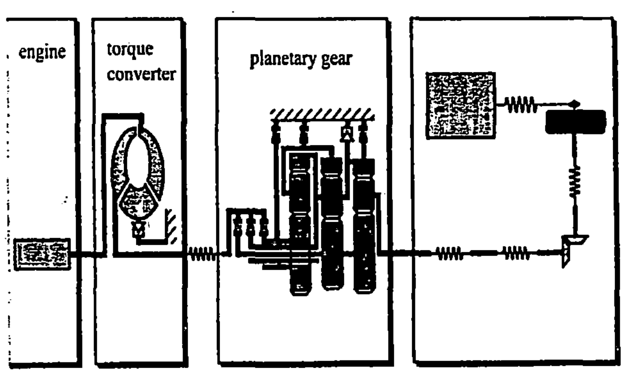

Figure 1. Schematic description of the powertrain

2.1 Mathematical model of torque converter

A torque converter is a common part in automatically geare vehicles. The main purpose of the converter is to multiply th output torque to the transmission during starting conditions. Als provide other benefits such as: stepless variation in torque an speed, smooth torque transmission between engine and automati transmission, absorption of vibration isolation and torque peacks.

The torque converter consist of three parts, a impeller (P), a turbin (T) and a reactor (R), respectively, using oil as the working flui for torque transmission. The impeller is connected to the engin shaft and the turbine is connected to the gearbox.

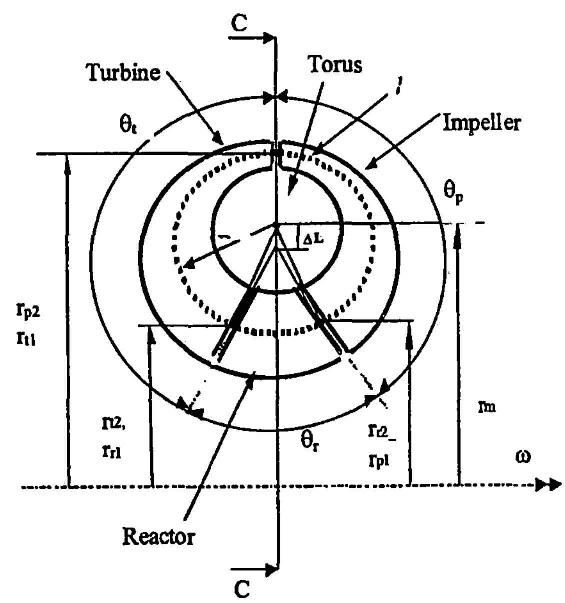

Figure 2. Schematic description of a three element torque converter (P-impeller, T-turbine, R-reactor) (1-inlet, 2-outlet)

Presented model includes the effects of torque converter flu inertia along the flow streamlines as well as the more standa angular momentum effects and steady state effects of flow a shock losses. The underlying assumptions are: the cross-section net torus flow area is constant, the spacing between the torq converter parts is neglected and the blade thickens effects are neglected and thermal effects are not considered [1], [3].

Angular momentum equation is:

$$M_p = \int_A (\vec{r}\times\vec{c})\cdot\rho\cdot(\vec{c}\times d\vec{A}) + \frac{\partial}{\partial t}\int_V (\vec{r}\times\vec{c})\cdot\rho\cdot dV \qquad (1)$$

$$\int_A (\vec{r}\times\vec{c})\cdot\rho\cdot(\vec{c}\times d\vec{A}) = (\vec{r}\times\vec{c})_2\cdot\dot{m}_2 - (\vec{r}\times\vec{c})_1\cdot\dot{m}_1 \qquad (2)$$

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

$$\frac{\partial}{\partial t}\int_V (\vec{r}\times\vec{c})\cdot\rho\cdot dV = \dot{\omega}_p\cdot\int_V r^2\cdot\rho\cdot dV + \rho\cdot S_p\cdot\frac{d\dot{V}}{dt} = I_p\cdot\dot{\omega}_p + \rho\cdot S_p\cdot\frac{d\dot{V}}{dt} \qquad (3)$$

Energy balance equation is:

$$E = \frac{1}{2}\cdot(I_p\cdot\omega_p^2 + I_t\cdot\omega_t^2 + I_r\cdot\omega_r^2 + \rho\cdot L_f\cdot\dot{V}^2) + \rho\cdot\dot{V}\cdot(S_p\cdot\omega_p + S_t\cdot\omega_t + S_r\cdot\omega_r) \qquad (4)$$

where: $L_f = \int_1^2 \frac{dl}{A\cdot\sin^2\beta}$ - ecvivalent fluid inertia length, β - blade angles.

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

$$\Delta p_{tr} = \frac{1}{2}\cdot\rho\cdot\sum_j (c_{trj}\cdot w_{trj}^2) \qquad (5)$$

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

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

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

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 (ω) for this gear set as follows [4]:

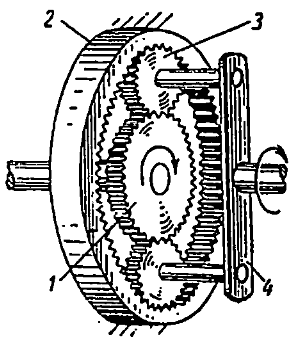

Figure 3. Schematic desription of planetary gear set

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

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

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

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

$$M_1 = i_s\cdot M_4$$

$$M_2 = i_u\cdot M_4$$

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

2.3 Mathematical model of friction clutch

The clutch is an element wich constraints the motion of po\ transmiting elements. It is assumed that the torque, transmitted the clutch plates, and the coefficient of friction at the cont surfaces. The torque transmitted (M_c) through the multi-disc clu 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 cont area, R_{cm} – medium radius of the friction plates, R_{c1} – min. rac of plates, R_{c2} – max. radius of friction plates:

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

μ - is a coefficient of friction and variation of the coefficient car 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 units is composed of sensors, soler valves and transmission control unit. It performes the dan clutch control, shift pattern control, hydraulic control during s process and so on. The transmission control units calculates value of parameters on the basic of data from sensors and gener the signal to operate the solenoid valves.

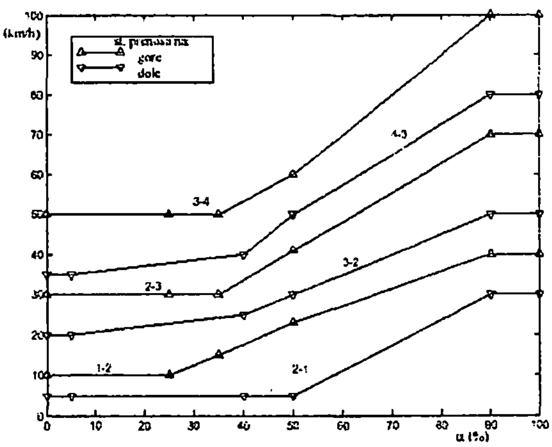

Figure 4.Shift schedule

In Fig. 4 is presented shift shedule map which, for a given throt in a given gear, vehicle speed an upshift or downshift takes plac

2.5 Mathematical model of engine

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

$$I_m \cdot \dot{\omega}_m = ((1-\beta)\cdot M_m - M_P) \tag{13}$$

where is β - coefficient, [2], I_m – engine moment of inertia, ω_m – engine angular velocity and M_m – engine torque.

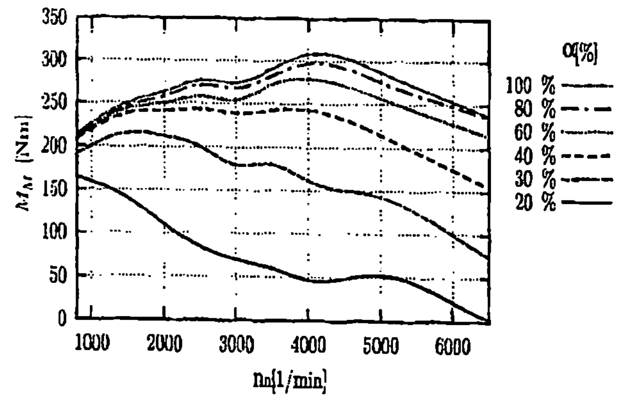

Figure 5. Engine map

The engine subsystem consists of a two-dimensional map that interpolates engine torque us 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_α) are represent as follows [2]:

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

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

$$F_\alpha = G \cdot \sin\alpha \tag{16}$$

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

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

with: r_d – the wheel radius, i_d – the transmission ratio of the rear differential, ρ_v – the air density, c_w – the air resistance coefficient, α - 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):

$$Iv \cdot \dot{\omega}_v = i_o \cdot M_2 - M_o \tag{18}$$

3. NUMERICAL MODELING

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

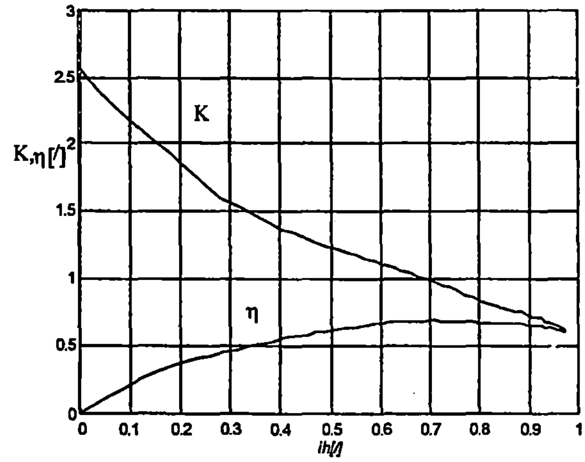

Figure 6. Torque ratio K and efficiency η as function speed ratio i

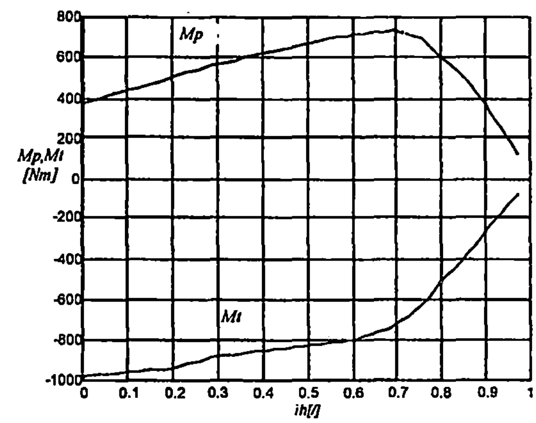

Figure 7. Impeller M_p and turbine torque M_t as function speed ratio i_h

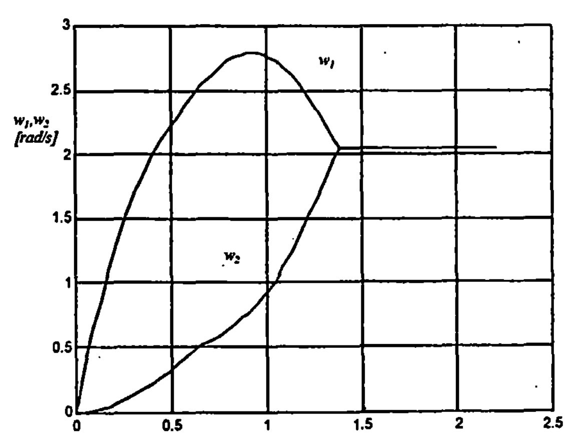

Figure 8. Angular speed of the engagement and disengagement multi-disc clutch

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

Fig. 8 is 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.

5. REFERENCES

/1/ Christer Ramdén: "Modeling and Simulation of a Hydrodynamic Torque converter", Linkoping University, XI 1996

/2/ Haj-Fraj A., Pfeiffer F.: "Optimal Control of Gear Shift Operations in Automatic Transmissions", Journal of Franclin Institute, 2001

/3/ Hrovat D. and Tolber W. E.: "Bond graph modeling and computer simulation of automotive torque converters", J. Franklin Inst., Special Issue on Physical Structure in Modeling, Vol. 319, No 1/2, pp. 93-114, Jan. 1985

/4/ Zahary Rubin, Scott Munns, John Moskwa: "The Development of Vehicular Powertain System Modeling Methodologies: Philosophy and Implementation", SAE paper 971089

/5/ Keum-Shik Hong at all: "An Object-Oriented Modular Simulation Model for Integrated Gasoline Engine and Automatic Transmission Control", SAE paper 1999-01-0750