

Modelling of dynamic characteristics of an automatic transmission during shift changes

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Abstract: This paper describes modelling of the transient dynamics of an automatic transmission during gear changes. A brief introduction to the automatic transmission system and the dynamic characteristics of the transmission components during the gear changes are presented. Then, detailed mathematical models of a four-speed automatic transmission manufactured by BTR Automotive, Australia, are developed. A mode description method is used to describe the transient shifting process and a modular structure of the transmission system, which consists of a torque converter module, geartrain module, hydraulic system module and modules of clutches and bands, is presented. As an application, the developed simulation system is applied to investigate the transient performance of the automatic transmission during the 1–2 shift process. The output torque profiles predicted by the model simulation correlate very well with the experimental data measured from vehicle tests.

Keywords: dynamic modelling, transient characteristics, automatic transmissions, gear shifting

NOTATION

$A(t)$	flow area
c	orifice flowrate coefficient
C_R	radial clearance of the valve
d	diameter of the valves
F	force
H	Heaviside step function
i	speed ratio
I	mass moment of inertia
k_0, k_1	clutch/band stiffness
l	flow length between ends 1 and 2
L_i	length vectors
M_i	modes of operation
n	number of contact surfaces in the multidisc clutch pack
p	pressure
\dot{p}	pressure rate
p_0	atmospheric pressure
$p_1 - p_2$	pressure drop across the ends 1 and 2
Q	flowrate
Q_i	generalized coordinates

r	radius
R_d	drum radius of the band
R_i	inner radius of the clutch friction plates
R_o	outer radius of the clutch friction plates
t	time
T	torque
$V(t)$	oil volume as a function of time
X	stroke of the valve/piston
β_B	band wrap angle
$\beta(p)$	bulk modulus as a function of pressure
β_0	bulk modulus at $p = p_0$
γ_a	air ratio at pressure p
γ_0	air ratio at $p = p_0$
ε, k	oil property constants
θ	angular velocity
$\ddot{\theta}$	angular acceleration
\mathcal{G}	dynamic viscosity of the oil
μ_C	static or dynamic friction coefficient of the clutch
μ_B	static or dynamic friction coefficient of the band

Subscripts

B	band
B1	band B1
C	clutch
C2	clutch 2

The MS was received on 8 October 2001 and was accepted after revision for publication on 26 June 2002.

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CR	carrier
E	engine
FS	forward sun
LP	long pinion
LU	lock-up clutch
OWC	one-way clutch
P	pump
R	ring gear
RS	reverse sun gear
SP	short pinion
T	turbine
V	vehicle

1 INTRODUCTION

Vehicles equipped with automatic transmissions provide advantages such as easy operation, smooth acceleration and safety. Demands for improved performance of the vehicle, such as driveability, passenger comfort, early malfunction detection and fuel economy, have been increasing dramatically in recent years. In automatic transmissions, frequent stops and starts and rapid acceleration/deceleration require frequent gear ratio changes, which perturb vehicle acceleration and engine speed. These conditions are the main causes of poor shift quality, poor fuel economy and unwanted emissions. Thus, an in-depth knowledge of the transient characteristics during a gear ratio change is very important for the design and control of an automatic transmission system.

Computer simulation is a powerful tool for investigating complex transmission systems. This can lead to shorter product design cycles, reduce development cost and allow engineers to explore many options early in the design phase. Simulating the transient characteristics of an automatic transmission is, however, complicated because many factors affect the shift quality during gear changes. These factors include the magnitude of the ratio change, clutch and band configurations, hydraulic control system, operating temperatures, shift scheduling, etc. System-level dynamic models of the transmission components, together with a complete understanding of the hydraulic system instabilities, and system-level interference forces are necessary for a model-based design and control of the automatic transmission systems.

The powertrain elements that describe an automatic transmission system consist of an engine, a hydrodynamic torque converter, a planetary gearset, friction elements and driveline elements. Most studies carried out to date on modelling have been the result of individual transmission manufacturers looking at their specific powertrain requirements and applications. As an example, Kotwicki [1] developed a dynamic model of an automatic transmission system based on the steady state operations of the torque converter. Instead of deriving model coefficients from the converter physical param-

eters, the coefficients were obtained through regression fits of the known torque converter characteristics. This model was also used in the automatic transmission model developed by Pan and Moskwa [2] to study the transient characteristics of a Ford automatic transmission. Simplified dynamic models of an engine, torque converter and vehicle are often used in the investigation of the transient characteristics of a powertrain equipped with an automatic transmission (see, for example, references [3] to [6]). In particular, Jo *et al.* [7] applied various simulation techniques in order to analyse the shift characteristics of the vehicle powertrain with automatic transmissions.

This paper focuses on the development of a simulation system to investigate the transient characteristics during gear changes in a BTR four-speed automatic transmission. The outcomes of this study will then be used in the design and optimization of the shift control elements. A state mode description method is used for describing the shifting process, and the governing equations of motion of the integrated powertrain system are derived. A modular structure of the automatic transmission, including the torque converter module, hydraulic system module, geartrain module and modules of clutches and bands, is developed. Experimental data are also used in the simulation processes to allow the model to be tuned and validated with reasonable accuracy. The transient characteristics of a 1–2 shift are presented, and the model is then used to determine optimum pressure profiles for the shifting elements.

2 AUTOMATIC TRANSMISSION SYSTEM AND THE 1–2 SHIFT PROCESS

The BTR four-speed automatic transmission system consists of a torque converter with a single-face lock-up clutch, four multiplate clutch assemblies, two brake bands, two one-way clutches and a planetary geartrain as shown in Fig. 1. In first gear, the C2 clutch is applied and the engine torque, T_E , and consequently the turbine torque, T_T , are transmitted to the forward sun (FS) gear through the C2 clutch. The clockwise rotation of the FS gear causes counterclockwise rotation of the short pinions (SP). The carrier (CR) is held by the 1–2 one-way clutch (1–2 OWC), which makes the ring (R) gear rotate around its axis in a clockwise direction. Torque is transmitted from the FS gear to the output shaft splined to the ring gear. The first gear ratio is the ratio between the FS gear speed, θ_{FS} , and the R gear speed, θ_R , i.e.

$$i_{FS1} = \frac{\theta_{FS}}{\theta_R} = \frac{r_R}{r_{FS}} \quad (1)$$

The speed ratios of other gear elements, such as the short pinions, long pinion (LP), reverse sun (RS) gear and

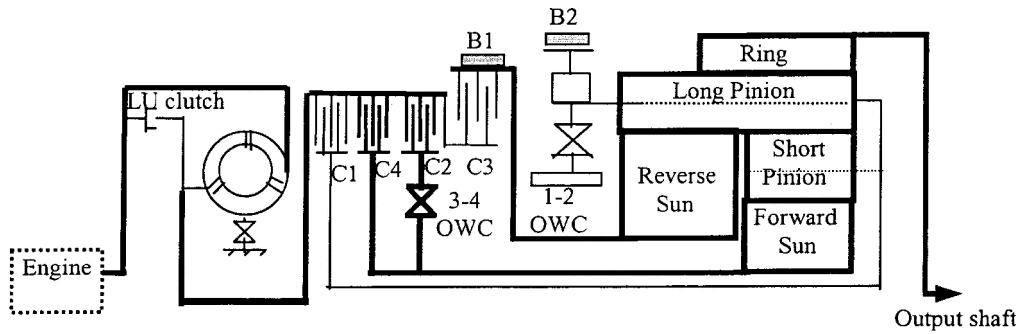


Fig. 1 The schematic diagram of the BTR four-speed automatic transmission

carrier, to the ring gear are

$$i_{SP1} = \frac{\theta_{SP}}{\theta_R} = \frac{r_R}{r_{SP}} \quad (2)$$

$$i_{LP1} = \frac{\theta_{LP}}{\theta_R} = \frac{r_R}{r_{LP}} \quad (3)$$

$$i_{RS1} = \frac{\theta_{RS}}{\theta_R} = \frac{r_R}{r_{RS}} \quad (4)$$

The 1–2 shift is accomplished by applying the B1 band to hold the RS gear and by overrunning the 1–2 OWC. Similarly, the speed ratios in the second gear are listed as follows:

$$i_{FS2} = \frac{\theta_{FS}}{\theta_R} = \frac{r_R(r_{FS} + r_{RS})}{2r_{FS}(r_{RS} + r_{LP})} \quad (5)$$

$$i_{SP2} = \frac{\theta_{SP}}{\theta_R} = \frac{r_R(r_{RS} - r_{SP})}{2r_{SP}(r_{RS} + r_{LP})} \quad (6)$$

$$i_{LP2} = \frac{\theta_{LP}}{\theta_R} = \frac{r_R}{2r_{LP}} \quad (7)$$

$$i_{CR2} = \frac{\theta_{CR}}{\theta_R} = \frac{r_R}{2(r_{RS} + r_{LP})} \quad (8)$$

where the subscripts 1 and 2 represent the first and second gears respectively.

The 1–2 shift sequence can be divided into two phases, a torque phase where the speed ratios do not change but the output torque is reduced by the application of B1 band and an inertia phase where the speed ratios begin to change. In the torque phase, the output torque changes according to the frictional characteristic of the B1 band and the inertial properties of the geartrain system. A typical torque and speed profile during the 1–2 shift is shown in Fig. 2. In general, two parameters may be used to assess the quality of the shift, *time width*, i.e. duration of the torque phase, and *torque hole*, i.e. torque drop during the torque phase. In general, the shorter the duration of the torque phase, the less the torque hole is during the torque phase.

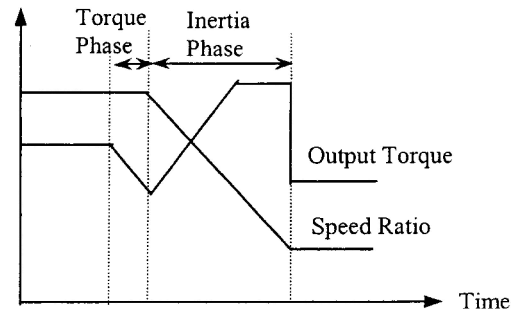


Fig. 2 Typical output torque profile with torque and inertia phases

3 MODE DESCRIPTION METHOD

An automatic transmission can be described as a parametric system due to the discrete speed changes that occur during the shifting process. In each gear, the system dynamics is determined from various combinations of active elements involved with the power flow, such as clutches, bands and gear elements. Each combination is called a ‘mode’ of operation [8] and each mode has its own distinct set of dynamic equations of motion. As a result, the number of governing equations also changes during each phase of the shift simulation. A particular shift process may be divided into several modes of operation, $\{M_1, M_2, \dots, M_n\}$, with one combination of transmission elements involved in each mode. A set of vectors of generalized coordinates, $\{Q_1, Q_2, \dots, Q_n\}$, and a set of length vectors, $\{L_1, L_2, \dots, L_n\}$, are defined for each mode of operation. The modes are arranged in increasing time steps, with M_n representing the last mode during the shifting process. The efficiency of the numerical simulation depends on the choices made on modes during the various phases of the shifting process. As an example, a poorly chosen mode with too many sets of dynamic equations will slow down the simulation. In addition, switching between modes will increase the complexity of the shift simulation.

According to the mode description method, the 1–2

shift process can be divided into four modes, i.e.

$$M = \{M_1, M_2, M_3, M_4\} \quad (9)$$

where modes 1 and 4 represent the steady state conditions of the first and second gears respectively and modes 2 and 3 define the torque and inertia phases during the 1–2 shift respectively. It is noted that in mode 4, there is slip in the torque converter before the second gear steady state condition is achieved. Referring to equation (9), the corresponding set of vectors of generalized coordinates is

$$Q = \{Q_1, Q_2, Q_3, Q_4\} \quad (10)$$

where

$$Q_1 = \{\theta_E, \theta_P, \theta_T, \theta_{FS}, \theta_{SP}, \theta_{LP}, \theta_{RS}, \theta_R, F_{SP}, F_{LP}, F_{RS}, F_R\}$$

$$Q_2 = \{\theta_E, \theta_P, \theta_T, \theta_{FS}, \theta_{SP}, \theta_{LP}, \theta_{RS}, \theta_R, F_{SP}, F_{LP}, F_{RS}, F_R\}$$

$$Q_3 = \{\theta_E, \theta_P, \theta_T, \theta_{FS}, \theta_{SP}, \theta_{LP}, \theta_{RS}, \theta_{CR}, \theta_R, F_{SP}, F_{LP}, F_{RS}, F_R\}$$

$$Q_4 = \{\theta_E, \theta_P, \theta_T, \theta_{FS}, \theta_{SP}, \theta_{LP}, \theta_{CR}, \theta_R, F_{SP}, F_{LP}, F_{RS}, F_R\}$$

Hence, the set of length vectors used in the 1–2 shift simulation is

$$L = \{12, 12, 13, 12\} \quad (11)$$

4 MODEL STRUCTURE

Figure 3 shows the modular models of an automatic transmission consisting of an engine module, torque converter module, geartrain module, hydraulic control module and the modules for clutches and bands. A brief description of these modules follows.

4.1 Engine model

In this study, a simplified engine model based on the numerical tabulation of the measured engine performance curve is used. A cubic interpolation method is used to obtain the engine torque during intermediate throttle positions and engine speeds.

4.2 Torque converter model

The torque converter provides torque multiplication during vehicle launch and smooth acceleration during gear changes. The derivation of detailed dynamic models of the torque converter that captures the accelerations of the fluid flow as well as time lags associated with the establishment of steady state flow conditions is beyond

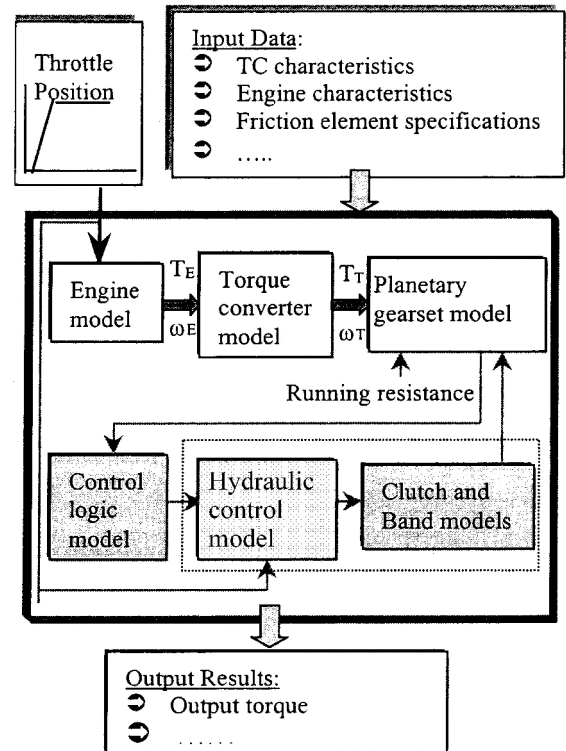


Fig. 3 Modular models of the automatic transmission

the scope of this paper. Instead, static torque converter models are used in this study; i.e. the torque converter characteristics are approximated from regression fits of actual experimental steady state performance curves.

4.3 Hydraulic control system model

The hydraulic circuit of an electronically controlled automatic transmission is complex and consists of many elements such as a pump, electromagnetic actuators, regulator and control valves, etc., as part of a complex system. It controls hydraulic pressures in clutches and bands according to the signals from the electronic control unit of the automatic transmission. It is essential for the hydraulic pressure at friction elements to be precisely controlled in relation to the amount of torque transmitted during the shift process in order to achieve optimum shift quality. The pressure fluctuations in the hydraulic system often create fatigue problems in the system components and lead to unacceptable shift qualities and low-frequency airborne hydraulic noise. Understanding the dynamic performance of each hydraulic element is therefore essential in predicting the transient characteristics of the frictional torque. A comprehensive mathematical model of the hydraulic control system is developed to calculate the pressure applied to the frictional elements during shift changes.

Because of the computational complexity of the hydraulic circuit, a pragmatic approach to the math-

emathical description of the hydraulic elements is required in order to execute the simulation. As an example, the flow characteristic (pressure drop/flow resistance) of the hydraulic circuit at low temperature is dominated by the viscosity effect rather than the turbulent orifice flow; thus the pressure drop due to viscosity effects should be incorporated at low operating temperatures. Mathematical representation of each hydraulic element can be formulated from first principles. The pressure changes in terms of various flow quantities are described by the differential equations for fluid dynamics, and the flow-rate passing through valves and orifices is described by laminar and turbulent orifice flow equations.

In particular, Newton's second law is used to derive the equation of motion of the regulator and control valves, which is primarily dictated by pressure differences, spring forces and jet forces. Fluid dynamics models are used to define the pressure changes in the hydraulic system. As an example, sudden pressure changes in a valve significantly affect the dynamics of the valve. The differential equations that describe the pressure changes in terms of various system flow quantities can be written in the following generalized form:

$$\dot{p}(t) = \frac{\dot{Q}(t)}{V(t)} \beta(p) \quad (12)$$

where the bulk modulus can be defined as a function of pressure, i.e.

$$\beta(p) = \frac{\beta_0 p k}{(1 - \gamma_a) p k + \gamma_a \beta_0}$$

and

$$\gamma_a = \frac{\gamma_0 (p_0/p)^{0.7}}{1 - \gamma_0 + \gamma_0 (p_0/p)^{0.7}}$$

The flowrate passing through the valves and orifices can be defined in terms of Reynolds number; i.e. the standard orifice flow equations of turbulent flow are

$$\dot{Q}(t) = c A(t) \sqrt{p_1(t) - p_2(t)} \quad (13)$$

and viscous flow conditions for laminar flow are

$$\dot{Q}(t) = \frac{p_1(t) - p_2(t)}{l} \frac{\pi d c_R^3}{12 \eta} \left[1 + \frac{3}{2} \left(\frac{\varepsilon}{C_R} \right)^2 \right] \quad (14)$$

The stiffness characteristics of the clutch can be divided into two regions; one is a linear region where the stiffness is governed by the clutch return spring and the second region is dominated by the compressibility of the facing material of the clutch. Based on the measured experimental data, the clutch non-linear characteristics can be written as

$$F_c(t) = k_0 X + H(X - X_0) k_1 (X - X_0)^2 \quad (15)$$

As an example, simulation results of the hydraulic system are compared with measured pressure profiles in Fig. 4. In particular, Fig. 4 compares the band pressure resulting from a typical current profile of the transmission control unit (TCU). The pressure profiles show very good correlation, with the exception of the period between 4.8 and 5.3 s. It is noted that the inertia phase begins at $t = 4.8$ s and the system enters the non-linear transient gear state.

4.4 Clutch and band model

Frictional torques of clutches and bands are obtained from empirical formulae [9]. The torque transmitted by clutches is proportional to the axial force across the clutch plates and the coefficient of friction on the contact surfaces. The torque transmitted through the multidisc

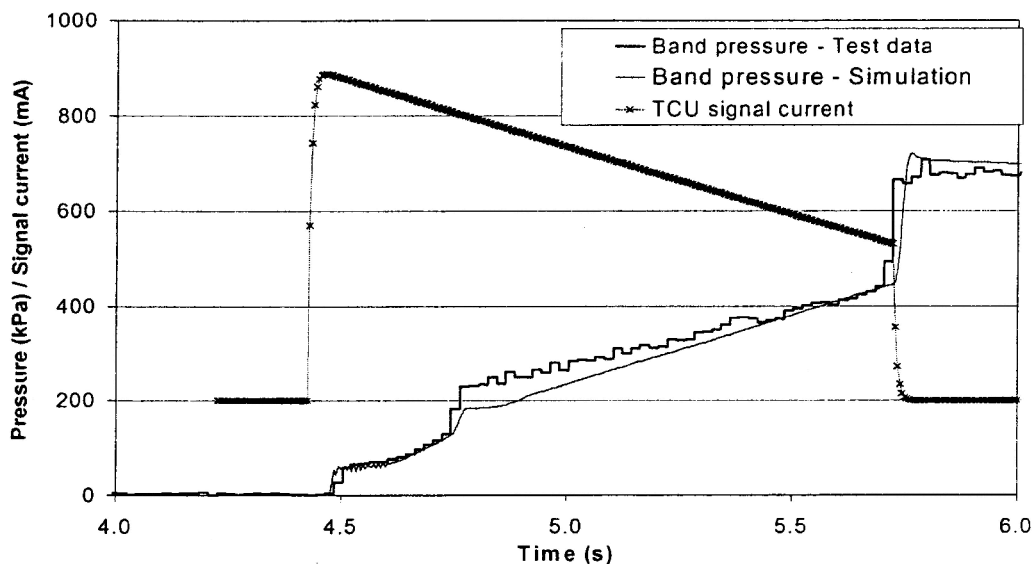


Fig. 4 Comparison of the simulated band pressure with measured data

clutch can be expressed as follows:

$$T_C = \mu_c n R_m F_c \quad (16)$$

where

$$R_m = \frac{2(R_o^3 - R_i^3)}{3(R_o^2 - R_i^2)}$$

It is noted that the dynamic coefficient of friction is generally a function of the slipping speed of the frictional couple.

The torque transmitted by the band depends on whether the band is in the energized mode or in the de-energized mode. In the energized mode, the band torque is given by [9]

$$T_B = R_d F_B (e^{\mu_B \beta_B} - 1) \quad (17)$$

and in the de-energized mode the band torque is

$$T_B = R_d F_B (1 - e^{-\mu_B \beta_B}) \quad (18)$$

Accurate values of F_B are determined from the model of the hydraulic control system described in the previous section. A typical band force profile used in the simulation of the 1–2 shift process is shown in Fig. 5.

4.5 Planetary geartrain model

The energy method and free-body analysis method are used to derive the governing equations of the geartrain during shift changes. Assuming that there is no backlash between gear meshings and the gear elements are infinitely stiff, the speed and torque relationships of the BTR Ravigneaux planetary gearset are

$$\theta_{FS} - i_{FS1} \theta_R - (1 - i_{FS1}) \theta_{CR} = 0 \quad (19)$$

$$\theta_{RS} + i_{RS1} \theta_R - (1 + i_{RS1}) \theta_{CR} = 0 \quad (20)$$

$$T_R = i_{RS1} T_{RS} - i_{FS1} T_{FS} \quad (21)$$

$$T_{CR} = -(i_{RS1} + 1) T_{RS} + (i_{FS1} - 1) T_{FS} \quad (22)$$

where equations (19) and (20) describe the speed relationship and equations (21) and (22) describe the torque relationship.

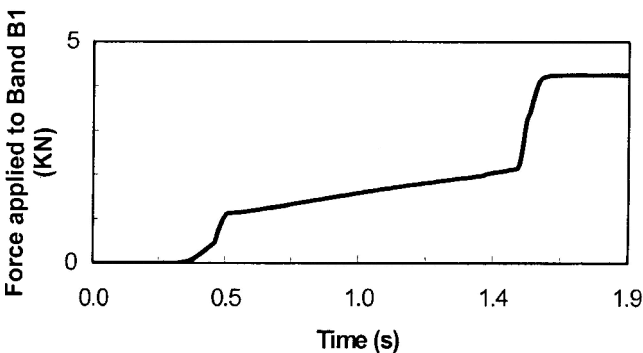


Fig. 5 Typical force applied to the B1 band during the 1–2 shift

5 MATHEMATICAL MODEL

Figure 6 shows the free-body diagram of the BTR four-speed automatic transmission system. The equations of motion of the transmission elements are developed using the following assumptions:

1. All links and rotating elements of the transmission are rigid.
2. All links have only one rotational degree of freedom.
3. Gears exhibit no backlash and bearings have no play.
4. Friction effects of rotating elements except bands and clutches are neglected.

The dynamic equations are expressed as follows:

Engine:

$$I_E \ddot{\theta}_E = T_E - T_{LU} - T_P \quad (23)$$

Torque converter shaft:

$$I_1 \ddot{\theta}_1 = T_{LU} + T_T - T_{C2} - T_{C4} \quad (24)$$

Forward sun gear:

$$I_{FS} \ddot{\theta}_{FS} = T_{C2} + T_{C4} - 3F_{SP} r_{FS} \quad (25)$$

Short pinion:

$$3I_{SP} \ddot{\theta}_{SP} = 3r_{SP} (F_{SP} - F_{LP}) \quad (26)$$

Long pinion:

$$3I_{LP} \ddot{\theta}_{LP} = 3r_{LP} (F_{LP} - F_R) - 3r_{LP} F_{RS} \quad (27)$$

Ring:

$$(I_R + I_V) \ddot{\theta}_R = 3r_R F_R - T_V \quad (28)$$

Reverse sun gear:

$$I_{RS} \ddot{\theta}_{RS} = 3r_{RS} F_{RS} - T_B \quad (29)$$

Carrier:

$$A \ddot{\theta}_{CR} = -T_{OWC1} + 3F_{SP}(r_{FS} + r_{SP}) + 3F_{RS}(r_{RS} + r_{LP}) - 3F_R(r_{RS} + r_{LP}) - 3F_{LP}(r_{SP} + r_{LP}) \quad (30)$$

The flow chart of the 1–2 shift simulation is shown in Fig. 7. Referring to the mode description method described in Section 3, modes 1, 2 and 3 represent the steady state in the first gear, torque phase and inertia phase respectively. The last mode, mode 4, is a phase after the gear ratio change. The reverse sun gear is held in the second gear, and the angular velocities and the torque are fixed by the second gear speed ratios.

6 RESULTS AND DISCUSSION

Figure 8 shows the typical speed ratio profile during a 1–2 shift for the B1 band force profiles shown in Fig. 5. The simulation is performed at a 100 per cent throttle condition at an engine speed of 4800 r/min. The 1–2 shift is initiated at $t = 0.25$ s and the speed ratio remains in

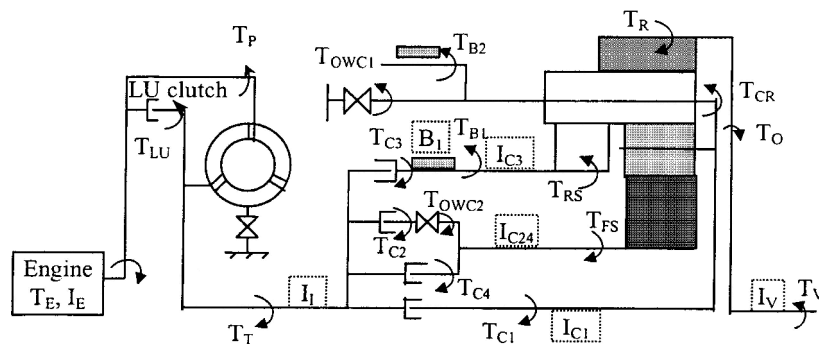


Fig. 6 Free-body diagram of the BTR four-speed automatic transmission

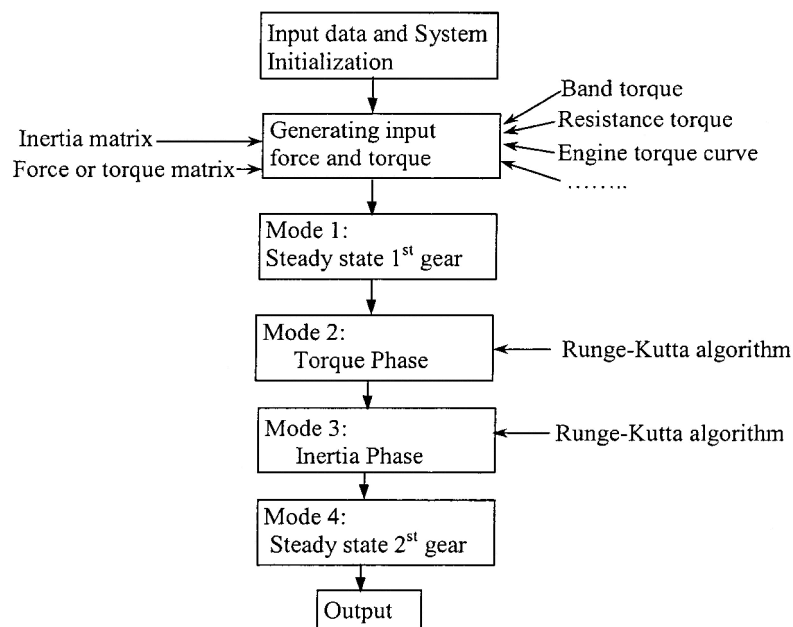


Fig. 7 Flow chart of the 1-2 shift simulation

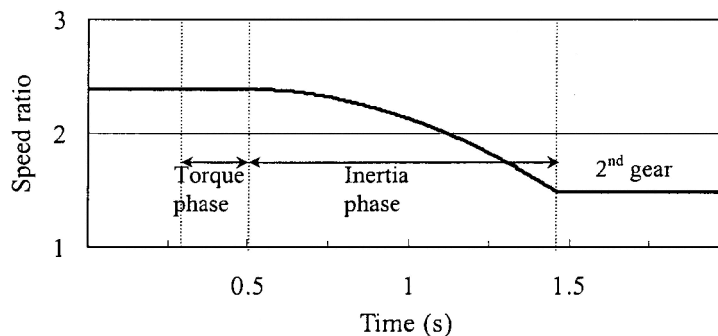


Fig. 8 Change in the speed ratio during the 1-2 shift

the first gear ratio ($i_{FS1} = 2.39$) during the torque phase between $t \approx 0.25$ s and $t \approx 0.49$ s. The speed ratio then continuously decreases to the second gear ratio ($i_{FS2} = 1.45$) during the inertia phase between $t \approx 0.49$ s and $t \approx 1.47$ s. Consequently, the engine speed also decreases during the inertia phase. Figure 9 shows that the corresponding output speed profile of the ring gear increases slightly during the 1-2 shift. Figure 10 shows the output

torque profile with the torque hole (rapid decrease in output torque) during the torque phase and gradual torque increase during the inertia phase. To examine the sensitivity of the time width and torque hole, the 1-2 shift simulation is performed with various combinations of band pressures and frictional coefficients.

Variations in band forces are common in high-volume production transmissions because of the dimensional

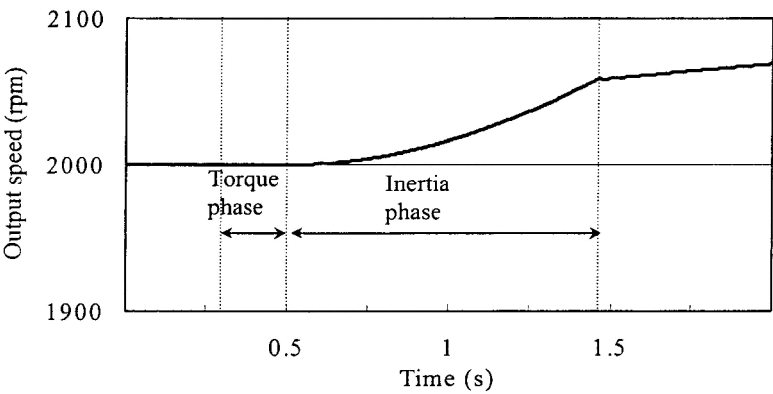


Fig. 9 Output speed profile of the ring gear

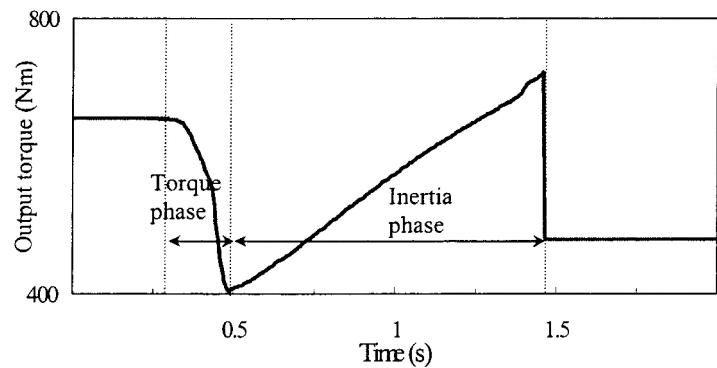


Fig. 10 Output shaft torque profile of the ring gear

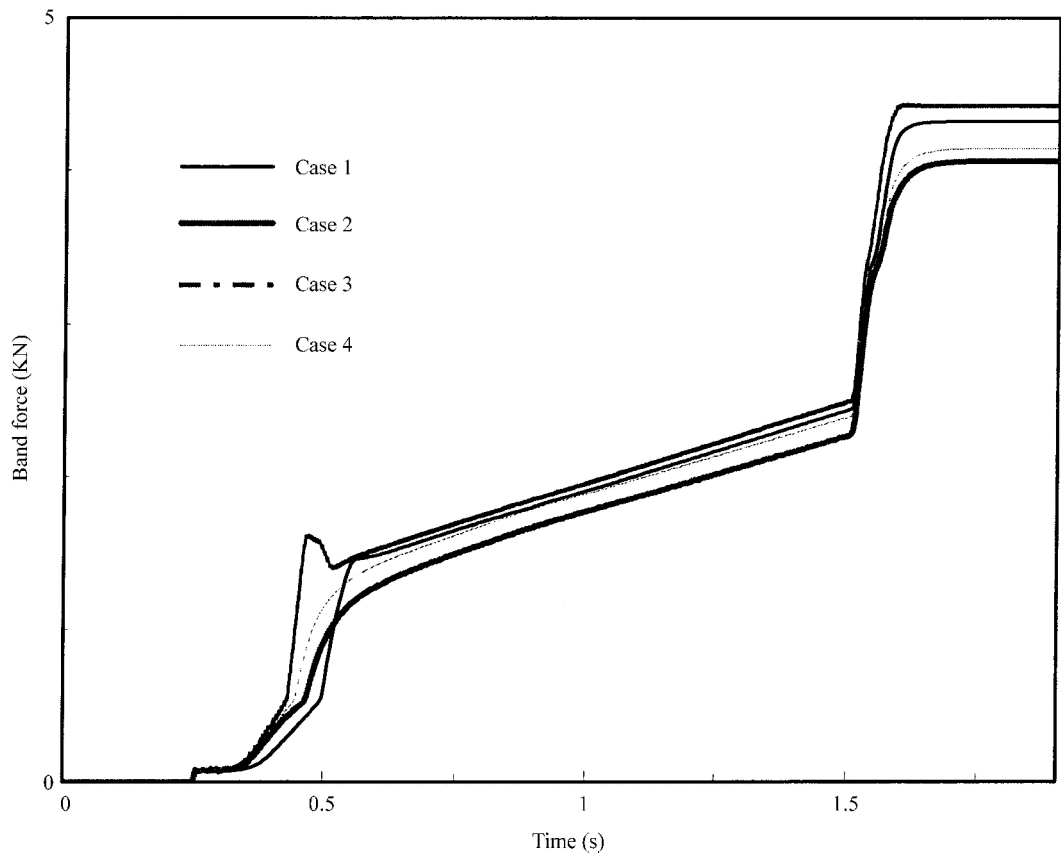


Fig. 11 Four different forces applied to the B1 band

tolerances associated with manufacturing transmission components, in particular, high-precision hydraulic elements. In addition, variations in oil properties such as frictional characteristics, oil viscosity, oil ageing, etc., alter the performance of the hydraulic system and contribute to variations in the band supply pressure. To simulate the variability of a typical production transmission, simulations are performed using the four slightly different band force profiles shown in Fig. 11. These band force profiles are obtained by changing the regulator valve (an important hydraulic system component) dimensions within the production tolerances. The output shaft torque profiles corresponding to the four input forces are compared in Fig. 12. Cases 1 and 3 show a sharp decrease in output torque at the end of the torque phase and case 2 shows the longest duration of the shift and sharp changes in output torque at the end of the inertia phase. Alternatively, case 4 shows the best torque profile; i.e. it exhibits the minimum torque hole and output torque disturbance. Results presented in Fig. 12 show that variations in the band force have a significant effect on the output shaft torque variations.

To investigate the effect of the dynamic friction coefficient of the band material, the 1–2 shift simulation is performed with a 2 per cent decrease in the friction coefficient. The results corresponding to the output torque profile are presented in Fig. 13. The torque phase

begins at $t = 0.25$ s and ends at $t = 0.55$ s, which is longer than the shift event presented in Fig. 10 (i.e. $t = 0.25$ s and $t = 0.49$ s respectively). Similarly, the inertia phase begins at $t = 0.55$ s and ends at 1.525 s. The inertia phase duration of 0.975 s is similar to the one shown in Fig. 10 (duration of 0.98 s). Results confirm that a small change in the dynamic friction coefficient has a significant effect on the time width and torque hole in the torque phase of the shift process. The static and dynamic friction coefficients vary during the service of automatic transmissions as a result of changes in oil temperature, oil viscosity, and components and oil ageing.

The developed simulation model was then rearranged to investigate what B1 band pressure and torque profiles were required to achieve a desired output torque profile. In the rearranged model, the desired shaft torque is used as one of the input parameters and the corresponding B1 torque is calculated as part of the simulation. As an example, a desired output torque profile, shown in Fig. 14, which is smoother than the torque profile presented in Fig. 10, is used as an input to the simulation model and then the 1–2 shift simulation is executed. The resulting B1 band torque required to produce the desired output torque profile is obtained and shown in Fig. 15. Here again the results show that a smooth B1 band torque profile is required to achieve smooth output torque variations during the 1–2 shift. This model can

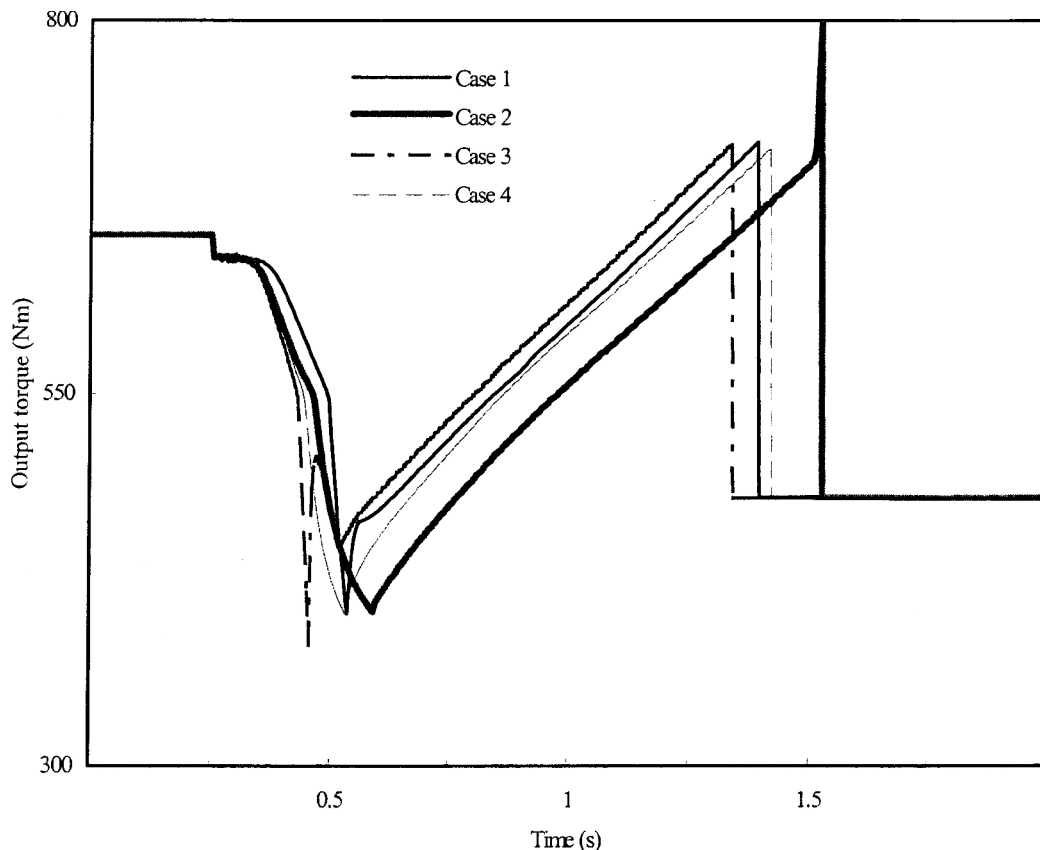


Fig. 12 Output torque profiles corresponding to the band forces shown in Fig. 10

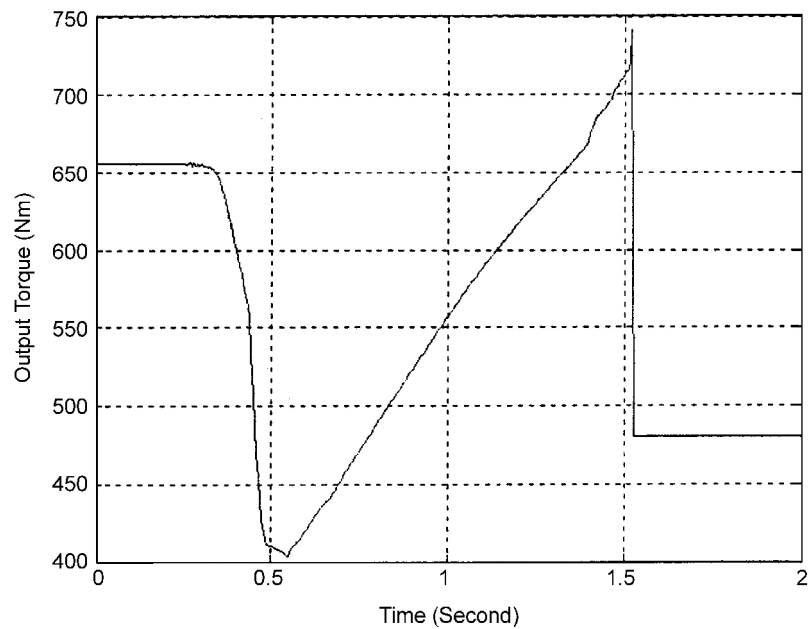


Fig. 13 Output torque profile with a 2 per cent decrease in the friction coefficient of Band B1

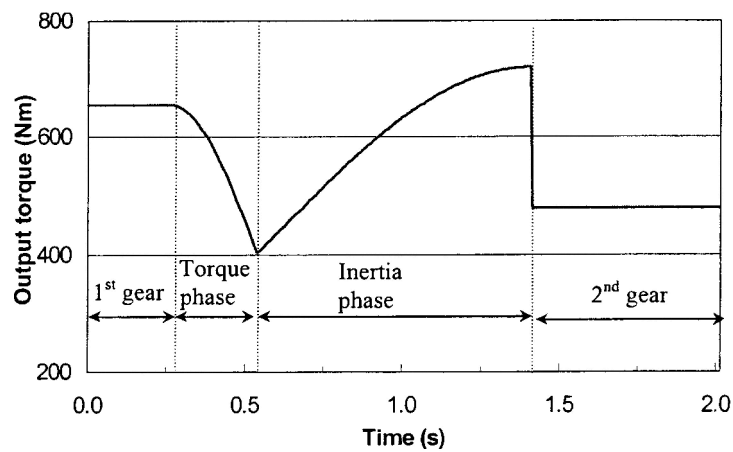


Fig. 14 Desirable output torque profile

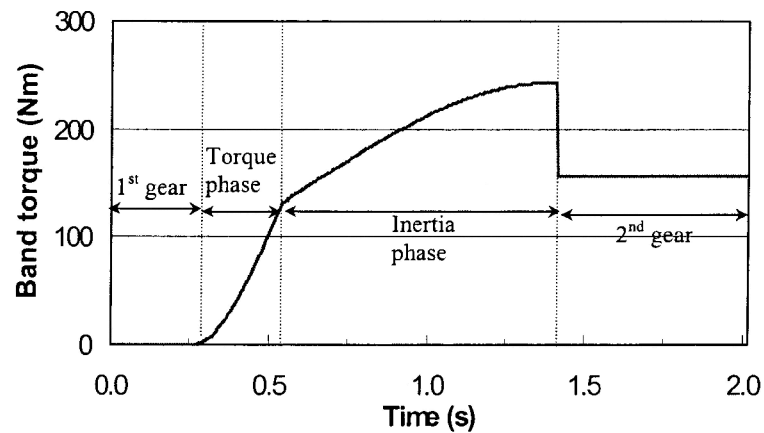


Fig. 15 Band torque required for the desirable output torque profile shown in Fig. 14

now be used to explore different torque profiles, which can drive an optimal shift strategy used in a vehicle.

7 CONCLUSIONS

A mathematical model and simulation system of the BTR four-speed automatic transmission have been developed in order to investigate the transient characteristics during gear changes. A mode description method of the shift process, a modular structure of the powertrain system and the flowchart of the simulation are presented. Simulation results show that the dynamic model accurately predicts dynamic transient performance of an automatic transmission during the 1–2 shift process. This is a very useful tool for determining optimum pressure profiles for the shifting elements.

ACKNOWLEDGEMENTS

The financial support for this work is provided jointly by BTR Automotive, Australia, and the Australian Research Council SPIRT Grant C00107787.

REFERENCES

- 1 Kotwicki, A. J. Dynamic models for torque converter equipped vehicles. SAE technical paper 820393, 1982.
- 2 Pan, C.-H. and Moskwa, J. J. Dynamic modelling and simulation of the Ford automobile transmission. SAE technical paper 950899, 1995.
- 3 Hojo, Y., Iwatsuki, K., Oba, H. and Ishikawa, K. Toyota five-speed automatic transmission with application of modern control theory. SAE technical paper 920610, 1992.
- 4 Hwang, S.-J., Chen, J.-S., Liu, L. and Ling, C.-C. Modelling and simulation of a powertrain-vehicle system with automatic transmission. *Int. J. Veh. Des.*, 2000, **23**(1/2), 145–160.
- 5 Megli, T. W., Haghighoie, M. and Colvin, D. S. Shift characteristics of a 4-speed automatic transmission. SAE technical paper 1999-01-1060, 1999.
- 6 Kim, Y. H., Yang, J. and Lee, J. M. A study on the transient characteristics of automatic transmission with detailed dynamic modelling. SAE technical paper 941014, 1994.
- 7 Jo, H.-S., Park, Y.-I., Lee, J.-M., Jang, W.-J., Park, J.-H. and Lim, W.-S. A study on the improvement of the shift characteristics for the passenger car automatic transmission. *Int. J. Veh. Des.*, 2000, **23**(3/4), 307–328.
- 8 Munns, S. A. Computer simulation of powertrain components with methodologies for generalized system modelling. MS thesis, University of Wisconsin-Madison, 1996.
- 9 *Design Practices: Passenger Car Automatic Transmissions*, 3rd edition, 1994, AE-18 (SAE Inc., Warrendale, Pennsylvania).