Development of a Multi-body Dynamics Simulation Tool for Tracked Vehicles*

(Part II, Application to Track Tension Controller Design)

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Multi-body Dynamics simulation of tracked vehicles is very useful not only for the analysis of dynamic behaviors, but also for the performance evaluation of the chassis controllers. The track tension is closely related to the maneuverability of tracked vehicles and the durability of tracks and suspension systems. In order to minimize the excessive load on the tracks and to prevent the peel-off of tracks from the sprocket, the track tension needs to be maintained at the optimum level throughout the maneuver. In this paper, a co-simulation tool is developed such that the performance of the track tension control system can be investigated for various maneuvering tasks. The MBD (multi-body dynamics) vehicle model for tracked vehicles is very complicated (189 bodies, 36 revolute joints, 152 bushing elements and 954 degrees of freedom) and cannot be easily implemented in commercial software. Besides, the track tension controller based on fuzzy logic can be easily constructed in the commercial control software. Therefore, co-simulation methodology is proposed so that the designed tension controller is interfaced into the MBD simulation software. The performance of the tension control system is verified through the proposed co-simulation tool.

Key Words: Tracked Vehicle, Track Tension Estimator, Tension Control

1. Introduction

Multi-body-dynamics (MBD) modeling of tracked vehicles is a very challenging task to describe the highly nonlinear characteristics of dynamic behaviors when the vehicles run. Computer simulation of the MBD models has been reported as a virtual prototyping tool^{(1)-(5),etc} and mainly utilized for the component design and dynamic analysis of tracked vehicles. Recently, it is suggested that the MBD models of tracked vehicles be considered as virtual proving grounds for the sophisticated chassis modules and that the controllers of the chassis modules be evaluated with the MBD models regarding the corresponding dynamic characteristics.

The track tension of tracked vehicles plays significant roles in various dynamic characteristics such as track peel-off from the sprocket, power efficiency, vehicle maneuverability and pressure distribution in chassis components. If the track tension is increased to prevent the peel-off of tracks, excessive load is generated on chassis components. At paved roads, the track tension needs to be adjusted to the minimum value so that the power efficiency is maximized by reducing the friction loss. But at the cross-country roads, the track tension should be increased to a higher value such that the maneuverability is maximized by increasing the tractive force between tracks and soil.

Conventionally, the maintenance of the track tension is achieved by the hydraulic passive tensioner which is connected to the idler arm with the fixed displacement. The tensioner is given a pre-tension of about 10% of the vehicle weight and has e.g. three modes: paved-road, cross-country and pivot turning. The driver of the tracked vehicle is supposed to choose the mode based on his/her experience, but this

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choice could result in serious problems such as peel-off and breakage of tracks.

The track tension needs to be controlled actively because dynamic variation in the track tension cannot be compensated by the passive tensioner. In the related literature only two approaches of indirect active tensioners can be found. The first approach⁽¹⁾ is constructed using a mechanical link between the idler arm and the closest road-wheel, where the position of the idler is determined solely by the location of the road-wheel without considering the track tension. The second approach⁽⁶⁾ compares the calculated track path length (TPL) with the desired value and adjusts the position of the idler by the hydraulic actuator. Both approaches should be categorized as indirect active tensioners in the sense that they actually do not use the track tension information for the track tension control.

In this paper, co-simulation methodology is developed such that the performance of the track tension control system can be investigated for various maneuvering tasks of tracked vehicles. The control system is composed of three sub-systems: track tension estimator, a hydraulic servo unit and track tension controller. The track tension around the idler is estimated in real-time based on the geometric model of the idler assembly. The position of the idler is the key variable determining the track tension and is activated by the hydraulic servo unit. Considering the fact that the track tension in maneuvering vehicles varies in a highly nonlinear manner with respect to the servo valve position, a fuzzy logic controller is designed to maintain the desired level in the track tension.

The designed controller is constructed using a control software⁽⁷⁾ and is interfaced into the multi-body dynamics (MBD) simulation tool⁽⁸⁾ for tracked vehicles. The MBD vehicle model used in this tool consists of 189 bodies, 36 revolute joints and 152 bushing elements and has 954 degrees of freedom. A recursive formalism and explicit numerical integration scheme is employed to solve the equations of motion. While simultaneously running both the control software and the MBD simulation tool, they exchange the dynamics and control output data at

Fig. 1 Geometry of the idler and the tension actuator

every control time-step. The performance of the tension control system is verified for the various maneuvering tasks and road conditions.

2. Track Tension Control System

2.1 Track tension estimator

The track tension can change abruptly depending on the location along the track path. Because the tension control actuator is connected to the idler arm, the track tension information around the idler is the most important and should be obtained both in real-time and on-line during driving. The real-time issue requires the major constraint on the computational capability. The on-line issue requires that the track tension be obtained based on the information available inside the maneuvering tracked vehicles. That is, soil condition and road shape cannot be used for determining the track tension.

The track tension around the idler can be estimated from a geometric model of the idler assembly and the measured tensioner pressure. The geometry of the idler, the idler arm and the tension actuator is shown in Fig. 1. The idler is supported with a tension actuator and guides tracks to the sprocket and road-wheels with maintaining the track tension. Huh et al.⁽⁹⁾ derives dynamic models for the idler assembly and estimates the track tension around the idler. The detailed derivation of the following equation is given in Huh et al.⁽⁹⁾

$$\hat{T}_{i1} \approx \hat{T}_{i2} \approx \frac{F_r l_3 \cos(\theta_r + \theta_{ia} + \theta_e) - m_{ia} g l_c \sin \theta_{ia} + F_{txrt} - m_i g \sin \theta_{ia}}{\cos(\theta_{ia} - \theta_1) + \sin(\theta_{ia} + \theta_2)} \quad (1)$$

where

l_1 : length between the road-arm hinge (p_1) to the idler pivot (p_2)

l_2 : length between the road-arm hinge (p_1) to the actuator pivot (p_3)

l_3 : length between the road-arm hinge to the mass center of the idler arm

F_r : tension actuator force

θ_r : rotational angle of the tension actuator

m_i : idler mass

m_{ia} : idler arm mass

θ_{ia} , θ_1 and θ_2 can be calculated from the geometry of the idler assembly⁽⁹⁾. F_{tnt} is the centrifugal force of a track segment wrapping around the idler. The inertial effect of the tension actuator is ignored because the hydraulic actuator is slow compared to other moving parts. The inertial effect of the idler is also negligible (less than 5%) compared to the normal values of the track tension. Equation (1) can be used for any maneuvering task of the tracked vehicles because the relation is derived without any assumption about the road shape or the motion resistance of the tracked vehicles.

2.2 Hydraulic unit

The hydraulic unit with servo valves is selected as the tension control actuator for the proposed tension control system. For the tension controller design, the dynamic characteristics of the hydraulic unit as well as tracked vehicles should be considered, in particular with respect to the track tension. The dynamic behaviors of tracked vehicles are described by the Multi-body Dynamics modeling⁽⁸⁾. The dynamic behaviors of the hydraulic unit are modeled in this section, whose configuration is illustrated in Fig. 2. The hydraulic unit consists of main hydraulic pump, electric motor, accumulator, 3-step servo valve, solenoid valve, relief valve, pressure reducing valve, hydraulic reservoir, and cooling system. The pressure of the hydraulic pump can be as large as 31.4 MPa assuming that the maximum tension is about 230 kN and the maximum operating displacement is 0.25 m.

The main hydraulic pump is a pressure-compensating type pump. The pump discharges full flow when the pump pressure is below the kneeling pressure (e.g. 31.4 MPa), otherwise it adjusts the inner

Fig. 2 Configuration of the hydraulic unit

swash plate to zero-flow state to prevent the energy loss through the relief valve. The dynamic characteristics of the swash plate around the kneeling pressure can be represented into a first-order equation⁽¹⁰⁾.

$$\dot{P}_{\text{pump}} = \frac{\beta}{V_{\text{pump}}} (Q_{\text{th}} - Q_{\text{leak}} - Q_{\text{load}} - Q_{\text{accum}}) \quad (2)$$

where

Q_{th} : theoretical flow

Q_{leak} : leakage flow

Q_{load} : load flow

Q_{accum} : accumulator flow

V_{pump} : pump volume

P_{pump} : pump pressure

β : bulk modulus

In Eq. (2), the theoretical flow can be calculated from the pump motor characteristics such as motor speed, ω_{motor} ,

$$Q_{\text{th}} = \frac{63}{6 \times 10^7} \cdot \omega_{\text{motor}} \cdot \phi \cdot I_{\text{yoke}} \quad (3)$$

$$\phi = -30\phi + 30$$

$$I_{\text{yoke}} = \begin{cases} \phi_{\min} = 0 & \text{if } P_{\text{pump}} \geq 31.4 \times 10^6 \\ \phi_{\max} = 1 & \text{if } P_{\text{pump}} < 31.4 \times 10^6 \end{cases}$$

where $\phi_{\min}$ and $\phi_{\max}$ represent that the swash plate is closed and open, respectively. The leakage flow can be expressed from the pump pressure as follows:

$$Q_{\text{leak}} = (2.5 \times 10^{-12}) \cdot P_{\text{pump}} \quad (4)$$

The accumulator is filled with the nitrogen gas and increases the instantaneous power in order to move the cylinder position to the maximum within 0.5 sec. The gas pressure is related to the pre-pressure, P_{pre} , and the pre-volume, V_{pre} :

$$P_{\text{gas}} = (P_{\text{pre}} + 1.0 \times 10^5) \left[\frac{V_{\text{pre}}}{V_{\text{pre}} - \int Q_{\text{accum}} dt} - 1.0 \times 10^5 \right] \quad (5)$$

Then, the accumulator flow in Eq. (2) is mainly determined by the gas pressure and expressed as:

$$Q_{\text{accum}} = K_{\text{accum}} \sqrt{\frac{2}{\rho} |P_{\text{pump}} - P_{\text{gas}}|} \times \text{sign}(P_{\text{pump}} - P_{\text{gas}}) \quad (6)$$

otherwise

$$Q_{\text{accum}} = 0$$

The four-way valve is adopted for the servo valve component where the area gradient ratio is consistent with the pressure gradient ratio. The load flow in Eq. (2) is determined depending on the servo valve direction and is expressed as:

if $u_n \geq 0$

$$Q_{\text{load}} = K_q \cdot u_n \cdot \frac{1+n}{2\sqrt{1+4n^3}} \sqrt{P_{\text{pump}} - \text{sign}(u_n) \cdot P_L}$$

if $|P_L| > P_{\text{pump}}$, $P_L = \text{sign}(P_L) \cdot P_{\text{pump}}$

if $u_n < 0$

$$Q_{\text{load}} = K_q \cdot u_n \cdot \frac{1+n}{2\sqrt{1+4n^3}} \sqrt{n \cdot P_{\text{pump}} - \text{sign}(u_n) \cdot P_L}$$

$$\text{if } |P_L| > n \cdot P_{\text{pump}}, P_L = \text{sign}(P_L) \cdot n \cdot P_{\text{pump}} \quad (7)$$

where u_n is the control input to the servo valve and other parameter notations are summarized in Table 1. In case the load pressure, P_L , does not satisfy the above relations, it can be expressed into the following dynamic equation from the cylinder model.

$$\dot{P}_L = (Q_{\text{load}} - A_e \cdot \dot{X}) \frac{2\beta \cdot (1 + 4n^3)}{A_b \cdot X + 4n^3 \cdot A_b \cdot (L_{\text{cyl}} - X)} \quad (8)$$

where X is the spool position and A_e is expressed as $(A_b + A_r)/2$. The relation between the spool position and the load pressure is expressed into second-order dynamic equations⁽¹¹⁾.

$$A_b \cdot P_L = M \cdot \ddot{X} + B_v \cdot \dot{X} + F_{\text{coul}} + F_{\text{ext}} \quad (9)$$

where F_{ext} is the external force and parameters values for Eq.(2) through Eq.(9) are summarized in Table 1. Simulation results with the above equations show that the hydraulic unit has about 0.5 sec time delay for the maximum displacement of the actuator.

2.3 Track tension controller

An active tension controller is designed so that the track tension follows the reference tension level during maneuvering. The reference tension level is selected as 4×10^4 N based on the concept that the track tension should be small to minimize the power consumption and mechanical wear. However, in case of rough road maneuvering, the reference tension level is increased to 5×10^4 N to prevent the peel-off of tracks. The track tension around the idler is considered as the controlled variable because the tension actuator is connected to the idler arm. The actuator pushes the idler arm forward if the estimated track tension is smaller than the reference value at the specified speed and driving mode.

Figure 3 shows the block diagram of the proposed active tension control system whose structure actually

Table 1 Parameter values for the hydraulic unit

Parameters	Description	Value
K_{accum}	Flow gain of the accumulator	$3.05 \times 10^{-4} [m^3]$
ρ	Density	$890 [kg/m^3]$
n	Area ratio of the cylinder	0.5
β	Bulk Modulus	$1.03 \times 10^9 [Pa]$
V_{pump}	Volume of the pump	$1.5 \times 10^{-3} [m^3]$
A_r	Rod area	$4 \times 10^{-3} [m^2]$
A_b	Bore area	$7.854 \times 10^{-3} [m^2]$
L_{cyl}	Stroke of the cylinder	0.27 [m]
K_q	Flow gain	$2.0 \times 10^{-6} [m^3]$
P_{pre}	Pre-pressure	$10 \times 10^6 [Pa]$
V_{pre}	Pre-volume	$20.0 \times 10^{-3} [m^3]$
M	Spool mass	300 [kg]
B_v	Damping coefficient	$3.2 \times 10^3 [N]$
F_{coul}	Coulomb friction	$1.18 \times 10^3 [N]$

represents the indirect adaptive control scheme. The tension controller is composed of two control algorithms. The first algorithm adjusts the servo actuator to maintain the reference tension value around the right and left idlers, respectively. The fuzzy logic⁽¹²⁾ is applied in designing the first algorithm not only because the track tension in moving vehicles changes in a nonlinear manner, but also because the characteristics of the track tension versus the hydraulic spool position are nonlinear. The triangular membership functions are used for the three linguistic variables of e , Δe and Δl_{rr} . These variables represent the error between the estimated and the reference track tension, the error rate and the displacement change in the tension actuator, respectively. The correlation between the input variables and the output variable is constructed into a rule-base as shown in Table 2. The rule-base represents the control law in the sense that it determines the control size of the actuator displacement based on the track tension error and the error rate. Three control gains are determined through a series of simulations with the Multi-body dynamics simulation tool⁽⁸⁾. Mamdani's minimum fuzzy implication rule⁽¹²⁾ is used for the fuzzy inference. The widely used COA (Center Of Area) strategy is used in defuzzification for the possibility distribution of the control action.

The second algorithm in the tension controller compensates the tensioner length in the proportional manner if the tensioner displacement is near the minimal position.

$$U_t = (L_{\text{bound}} - L_r + 0.0025) \cdot G_t \quad (10)$$

Table 2 The fuzzy rule base of the controller

Δe	e	NB	N	Z	P	PB
NB		PB	PB	PB	P	Z
N		PB	PB	P	Z	N
Z		PB	P	Z	N	NB
P		P	Z	N	NB	NB
PB		Z	N	NB	NB	NB

NB : negative big, N : negative, Z : zero, P : positive, PB : positive big

Fig. 3 The proposed track tension control system

where

U_i : servo valve command

L_r : current tensioner length

L_{bound} : lower limit of the tensioner length

G_i : length control gain

3. Co-Simulation of the MBD Tool and the Controller

The designed controller is constructed using a control software⁽⁷⁾ and is interfaced into the multi-body dynamics (MBD) simulation tool⁽⁸⁾ for tracked vehicles. The control results from the designed track tension controller are obtained by running both the control software and the MBD simulation tool simultaneously.

3.1 MBD simulation tool

The tracked vehicle consists of a chassis and two track subsystems. The chassis subsystem includes a chassis, sprockets, support rollers, idlers, road arms, road wheels and the suspension units. The sprockets, support rollers, and road arms are connected to the chassis by revolute joints. The track link subsystem includes a shoe body, a pin, rubber bushings, and a rubber pad. Rubber bushings and pin are inserted into the hole of shoe body with certain radial pressure and a rubber pad is mounted on the ground interaction side of shoe body.

In the MBD vehicle model the track chains interact with the chassis components through contact forces. Since adjacent track links are connected by compliant force elements, each link in the track chain has six degrees-of-freedom that is represented by three translational coordinates and three Euler angles⁽¹³⁾. The MBD model⁽⁸⁾ employs the relative generalized coordinates as shown in Fig. 4 in order to reduce the number of equations of motion. The efficient contact search kinematics and algorithms are developed in the context of the compliance contact model between track links, rollers, sprockets, and

ground. A recursive formalism and explicit numerical integration scheme is employed to solve the equations of motion⁽⁶⁾. In order to validate the developed MBD model against the experimental measurements, several empirical techniques are applied to the real proving ground tests of the high mobility tracked vehicle⁽⁸⁾.

3.2 Interface between the MBD model and the tension controller

Based on the tension estimator (section 2.1), the hydraulic unit (section 2.2) and the tension controller (section 2.3), the designed control system is constructed in the control software⁽⁷⁾. In order to verify the performance of the designed control system for the various maneuvering tasks and road conditions, the control system is interfaced to the MBD simulation tool as shown in Fig. 5. The estimator block receives the motion data from the MBD simulation tool and sends the estimated tension value to the controller. The hydraulic block receives the control output from the controller and sends the actuator force data to the MBD simulation tool. Every interfacing data is stored in the separate files and used for the later analysis.

3.3 Co-simulation for the performance evaluation

Co-simulation of the MBD simulation tool and the control software requires the data exchange at every control time-step. The MBD tool utilizes the generalized-alpha method⁽⁶⁾ for integrating stiff mechanical characteristics and employs the variable integration time-step as small as 10^{-6} sec. Because the control time-step is selected as 10^{-2} sec, the MBD tool has to stop calculations at every 10^{-2} sec and sends the dynamics data to the control system. After the control system updates the calculations of the tension estimator and controller, it has to run the hydraulic unit at the smaller time step (10^{-4} sec) because of the non-linearity and integration difficulties. The actuator force data calculated from the hydraulic unit is transmitted back to the MBD tool

Fig. 4 MBD simulation tool

Fig. 5 MBD model and controller interface

at every control time-step. The schematic structure of the developed co-simulation strategy is illustrated in Fig. 6.

4. Simulation Results

Utilizing the co-simulation strategy developed in section 3, the feasibility of the proposed tension control system is demonstrated for the various maneuvering tasks and road conditions. The active control performance is compared with those obtained from the passive-type (fixed displacement) tensioner. The reference tension level is set to different values depending on driving conditions.

(a) Longitudinal acceleration and traveling

The tracked vehicle is accelerated for 10 seconds and subsequently traveling with the constant speed at 22 km/h. Figure 7 compares the results that the track tension with the passive tensioner changes depending on the vehicle velocity, but with the active tension controller, the track tension is maintained to the reference level (40 kN) regardless of vehicle speed.

(b) Hard deceleration and stop

The tracked vehicle is decelerated from 36 km/h to zero in 8 seconds and stops for 2 seconds. The track tension with the passive tensioner can have large values depending on the decelerating rate as

Fig. 6 Co-simulation of MBD tool and controller

Fig. 7 Tension control results for test (a)

shown in Fig. 8. The track tension with the active tension controller is maintained to the reference level (40 kN) except at the transient period. The tension difference between the active and passive tensioners is closely related to the power consumption and is important from the power efficiency perspective.

(c) Longitudinal traveling on the rough road (PROFILE IV)

The track tension is compared when the tracked vehicle is traveling over the rough and rigid road. The rough road (called Profile IV road) consists of bumps with various size and slope as illustrated in Fig. 9. The tracked vehicle is accelerated to 22 km/h in 10 seconds and traveling with the constant velocity for another 10 seconds. With the active controller the track tension is maintained around the reference level (50 kN) as shown in Fig. 10, whereas the track tension with the passive tensioner fluctuates with much higher peaks. The size of peaks is closely related to the fatigue life of chassis components.

(d) Traveling across a bump

The track tension is compared when the tracked vehicle is traveling across a bump with the constant velocity of 22 km/h. The bump is a semicircular shape with the radius of 0.2 m as illustrated in Fig. 11. Compared to the passive case, the active tension controller settles down the track tension more quickly after passing the bump and maintains the reference level of 50 kN as shown in Fig. 12.

Fig. 8 Tension control results for test (b)

Fig. 9 Rough road profile (Profile IV)

Fig. 10 Tension control results for test (c)

Fig. 11 Single bump for test (d)

Fig. 12 Tension control results for test (d)

5. Conclusions

The co-simulation tool is developed such that the performance of the track tension control system can be investigated for various maneuvering tasks. A fuzzy logic controller is designed in order to compensate for the non-linear characteristics of tracked vehicles and the hydraulic unit. The developed control system is constructed in the control software and

interfaced to the Multi-body dynamics (MBD) simulation software. Co-simulation strategy runs both the control and the MBD softwares simultaneously. The control performance is evaluated for the various maneuvering tasks and it demonstrates the feasibility of maintaining the track tension at the desired level. The proposed co-simulation method is believed very useful from the perspectives of controller gain tuning, control validation for risky maneuvering, power consumption evaluation, etc.

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