Modeling and Identification of Longitudinal Responses of an Electric Vehicle for Drivability Improvement

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***Abstract*—This paper develops a control-oriented drivability model for an electric vehicle. First, we establish a discrete-time nonlinear input-output mechanism model through investigating the longitudinal dynamic characteristics of the vehicle. Second, we use the least squares method to identify the model parameters based on the data obtained from a real electric vehicle. The model is built in Simulink and its accuracy is validated by CRUISE. The calibration results demonstrate that the identified model is capable of predicting longitudinal vehicle responses that affect drivability and useful in control algorithm design.**

***Index Terms*—electric vehicle, longitudinal response, modeling, identification**

I. INTRODUCTION

The legislation on reduction of fuel consumption and CO2 emissions has created a large interest in electric vehicle (EV) technologies. Electric vehicles have developed by world's major car manufacturers in recent years [1]. The EVs have several advantages over vehicles with internal combustion engines, such as energy efficiency and environmental friendliness, and is seen to the right way to solve the energy and environment problems.

During development and calibration phases of an EV control system, it is of crucial importance to assess and optimize vehicle drivability. Accurate modeling of the vehicle propulsion system is therefore necessary to evaluate performance in early stages of the vehicle development process. In this paper, we focus on vehicle longitudinal responses which greatly influence vehicle drivability. The objective of this paper is to present a control-oriented EV drivability model that allows for the prediction of vehicle longitudinal responses [2].

A mechanism model is made up in Simulink which describes the details of the dynamic EV drivability. As the mechanism model is too complex for real-time computing, a simple input-output model is developed. As the mechanism model shows, the input-output model is nonlinear. The available nonlinear identification techniques have been subdivided into three basic classes[3]:

(1) Cascade or block-oriented structured approaches;

(2) Kernel or nonparametric approaches;

(3) Parametric approaches.

The parametric approaches is used because the output of the model depends on its parameters in linear way, though the relationship between the input and output of the model is nonlinear.

This paper is organized as follows. Section II presents the architecture of the experimental vehicle. Section III describes the details of the dynamic EV drivability model. Section IV provides comparisons of the developed model and the result of CRUISE. A simplified discrete-time nonlinear input-output model is deduced from the mechanism model, then identification and verification of the simplified model is made in Section V. Conclusions are made in Section VI.

II. EXPERIMENTAL VEHICLE

The experimental vehicle is a pure electric bus of Anhui Ankai Automobile Co., Ltd. Its driving system is mainly constituted by a traction electric machine (motor), a propeller shaft, a universal joint, a final drive and drive shafts. Fig. 1 shows the driving system configuration.

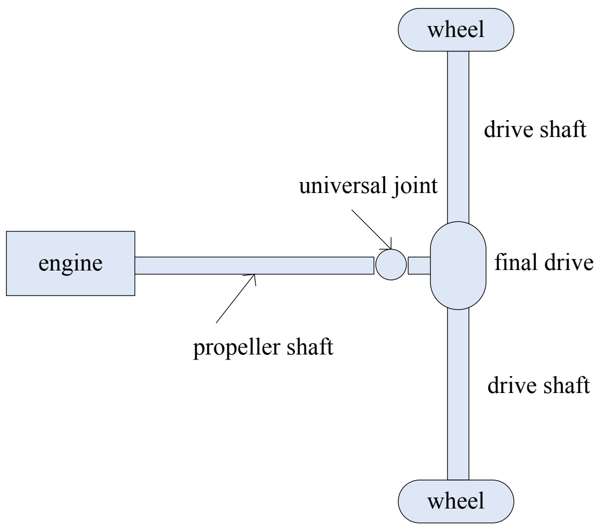

Figure 1. Driving system configuration

The traction electric machine (EM) is an asynchronous AC induction machine. The main parameters are shown in Table I:

TABLE I.
MAIN PARAMETERS

Parameters	***Value***
frontal area	6m^2
drag coefficient	0.6
load on front axle	5096Kg
load on rear axle	9403Kg
vehicle weight	14500Kg
radius of the wheel	0.465m
wheel inertia	9 kg*m^2
efficiency of the transmission system	0.97
transmission inertia	0.03 kg*m^2
transmission ratio	5.63
max motor torque	2400N
max motor speed	2520rpm
max motor power	240Kw
motor inertia	4 kg*m^2
motor efficiency	0.88

III. SYSTEM MODEL

A dynamic model of the experimental vehicle is developed to facilitate the evaluation of vehicle drivability in terms of longitudinal vehicle responses. The model is implemented in the MATLAB/Simulink environment using a variable-step solver that is suited for stiff dynamic systems [4].

In developing the model, the propeller shaft is assumed to be rigid, while the rear and front half-shafts are assumed to be elastic. The impacts of environmental factors such as temperature, pressure and humidity are not taken into consideration. Drivetrain losses are represented by lumped efficiency. The backlash impact is ignored.

Mathematical models of the vehicle subsystems and components are described as following.

A. Electric motor and propeller shaft

The dynamics of the electric motor and propeller shaft can be described using a lumped model with inertias reflected to the electric motor (EM) output shaft [5] as:

$$\dot{w}_{em} = \frac{1}{J_{em} + J_{mech}}\left(T_{em} - T_{em,fr} - \frac{2}{\varsigma_{rd}}T_{shaft}\right) \quad (1)$$

where w_{em} is the angular velocity of the motor rotor shaft, J_{em} is the EM rotor inertia, J_{mech} is the mechanical inertia at the EM rotor shaft, T_{em} is the EM output torque, $T_{em,fr}$ is the friction torque, T_{shaft} is the reaction torque of a single half shaft on the rear axle, ς_{rd} is the rear main reducer speed ratio. $T_{em,fr}$ is modeled as a constant value.

As the power supplied by the motor is limited by the motor max power P_{em}, T_{em} is limited by the max power. When the current power exceeds the max power, T_{em} reduces to:

$$T_{em} = 9550\frac{P_{em}}{n_{em}} \quad (2)$$

where n_{em} is the speed (with the unit as rpm) of the motor.

B. Universal joint

Universal joint connects the propeller shaft and the drive shaft. When the angle between the input shaft(the propeller shaft) and output shaft(the drive shaft) is not zero, the rotating speed of the output shaft is not the same as the rotating speed of the input shaft, even when the input drive shaft axle rotates at a constant speed. When the angle between the input and output shafts is α, and the input drive shaft axle rotates a round at a constant speed w_1, the rotating speed of the output shaft changes from $w_1 \dfrac{1}{\cos\alpha}$ to $w_1 \cos\alpha$. Therefore, the rotating speed of the output shaft changes instantaneously, but the average speeds of the input and output shafts are the same. Because the vehicle model is assumed driven on a straight road, α =0 and the speeds of input and output shafts are the same.

The transmission efficiency of the universal shaft is a function of α, shaft supporting structure and materials, processing and assembling accuracy, and lubrication conditions.

When $\alpha \le 25^o$ the efficiency $\eta_u = 1 - f\dfrac{d_1}{r}\dfrac{2\tan\alpha}{\pi}$, where f is coefficient of friction, d_1 is the diameter of the cross shaft, r is the radius of the cross shaft. Under normal circumstances, the transmission efficiency of the universal joint is 97% ~ 99% [6].

C. Drive shaft

The half shafts are modeled as elastic rods with a damping coefficient, as Fig. 2 shows. The torque applied to the shaft is proportional to the speed difference and twist angle between the shaft's terminals.

The torque of the driveshaft can be obtain as [7]:

$$T_{shaft} = k_{shaft}\int\left(w_{diff} - w_{wheel}\right)dt + c_{shaft}(w_{diff} - w_{wheel}) \quad (3)$$

where the k_{shaft} [N • m/rad] is the torsional stiffness, c_{shaft} [N • m/(rad/s)] is the shaft damping coefficient.

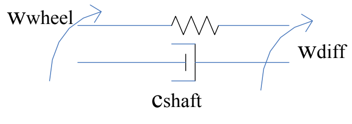

Figure 2. Drive shaft model

D. Brakes

The vehicle meets the driver's braking commands using a combination of conventional pneumatic braking and regenerative braking. Regenerative braking torques is realized by the electrical motor, pneumatic braking torques is modeled using an empirically determined relationship between the brake pedal position and the pneumatic brake pressure. So the brake torques on the front and rear wheels can obtain as [8]:

$$T_{br} = R_{br} p(\alpha_{br,pedal}) A_{br} \mu_{br} \tag{4}$$

where T_{br} is the brake torque, R_{br} is the effective friction radius, A_{br} is the brake piston surface, μ_{br} is the friction coefficient, $p(\alpha_{br,pedal})$ is the function that maps the brake position $\alpha_{br,pedal}$ to the brake air pressure.

E. Tire Model

The frictional properties of the road surface are assumed to be uniformly acting on all tires.

The tire model proposed by Pacejka [9] is used to represent the longitudinal tire dynamics. This model is widely used in professional vehicle dynamics simulations; it uses the semi-empirical "Magic Formula" to compute the tractive forces generated by the tires:

$$F_x = D \cdot \sin(\mathrm{C} \cdot \mathrm{atan}(\mathrm{B} \cdot \kappa \text{-} \mathrm{E}(\mathrm{B} \cdot \kappa \text{-} \mathrm{atan}(\mathrm{B} \cdot \kappa)))) \tag{5}$$

where B is stiffness factor, C is shape factor, D is peak factor, E is curvature factor, κ is transient tire slip, F_x is tractive forces of the tire. B, C, D are obtained from the manufacturer's tire data. $D = \mu F_z$, μ is the coefficient of friction between the tires and the road surface, F_z is vertical forces acting on the front and rear wheels.

The vertical forces acting on the front and rear wheels ($F_{z,f}$ and $F_{z,r}$) change with the vehicle acceleration. We assume that the vertical forces acting on all front (rear) wheels the same. Then they can be calculated from the following [10]:

$$\begin{aligned} 2F_{z,f} L &= M_{veh} g l_r - M_{veh} \dot{v}_{veh} H \\ 2F_{z,r} L &= M_{veh} g l_f + M_{veh} \dot{v}_{veh} H \end{aligned} \tag{6}$$

where L is the distance between the front axle and rear axle, l_r is the distance between the rear axle to the center of mass, l_f is the distance between the front axle to the center of mass, H is the distance between the center of mass to the grand, and M_{veh} is the mass of the vehicle.

The wheel slip is $\kappa = \dfrac{r_{wheel} w_{wheel} - v_{veh}}{\max(v_{veh}, r_{wheel} w_{wheel})}$, where r_{wheel} is the radius of the wheel, w_{wheel} is the wheel's angular velocity, v_{veh} is the vehicle's longitudinal velocity.

The wheel's angular velocity is calculated from a torque balance at the wheels:

$$\dot{w}_{wheel} = \frac{1}{J_{wheel}} (T_{shaft} - T_{br} - r_{wheel} F_x - r_{wheel} F_{roll,fr}) \tag{7}$$

where J_{wheel} is the wheel inertia and $F_{roll,fr}$ is the tire rolling force. The tire model is shown in Fig. 3.

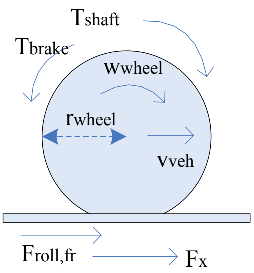

Figure 3. Tire model

$F_{roll,fr}$ is modeled as a predominantly static function that also has a slight linear dependence on vehicle speed at constant tire pressure:

$$F_{roll,fr} = F_z \cos\gamma (c_1 + c_2 v_{veh}^2) (8) \tag{8}$$

where c_1 and c_2 are rolling coefficients and γ is the road inclination.

F. Vehicle

Using Newton's second law on the longitudinal direction, the following equation is obtained

$$\dot{v}_{veh} = \frac{1}{M_{veh}} (2F_{x,f} + 2F_{x,r} - \frac{1}{2} \rho_{air} C_d A v_{veh}^2 - M_{veh} g \sin(\gamma)) \tag{9}$$

where ρ_{air} is the air density, C_d is the vehicle's drag coefficient, A is the vehicle's frontal area, γ is the climbing angle.

In summary, the total model developed in Simulink is shown in Fig. 4.

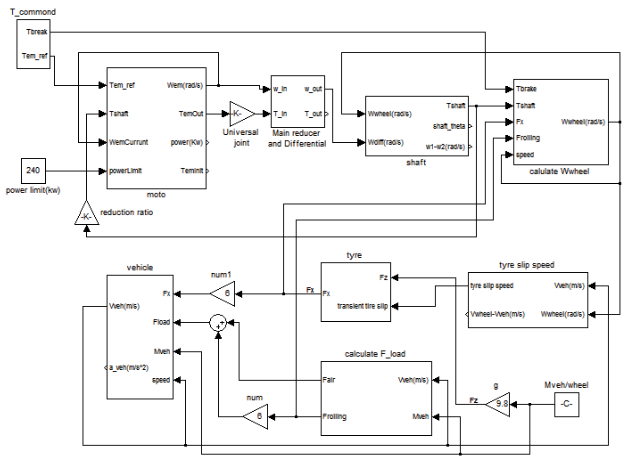

Figure 4. Model block diagram

IV. VALIDATION

The developed model is validated by AVL CRUISE, which is a simulation package that supports common tasks in a vehicle system and driveline analysis throughout all development phases, from concept planning to launch and beyond [11].

A precise pure electric bus model is created in CRUISE, and is used for validating the developed model. AVL CRUISE is a industry's most powerful, robust and adaptable tool for vehicle system and driveline analysis. AVL CRUISE is a simulation software which is widely used in industry; it can simulate vehicle fuel consumption and emissions for any driving cycle or profile, driving performance for acceleration, hill climbing, traction forces, braking and so on. The driving performance for acceleration is used to compare the CRUISE model and developed model.

In the acceleration performance test, the acceleration pedal is set as its max value, so the torque command sent to the motor is always the max state, and the vehicle speed will always increase. The test lasts 28 seconds, and the vehicle speed and motor torque acquired from the two models are depicted in Fig. 5 and Fig. 6:

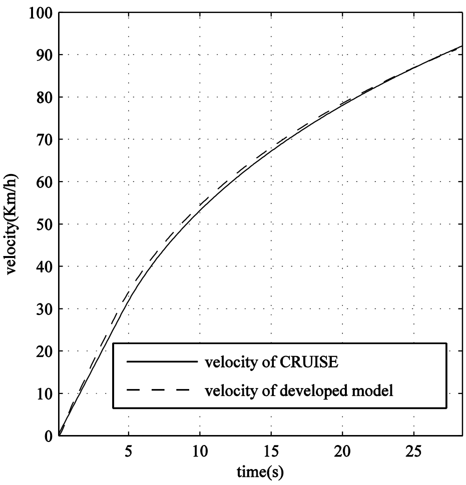

Figure 5. Vehicle speed comparison

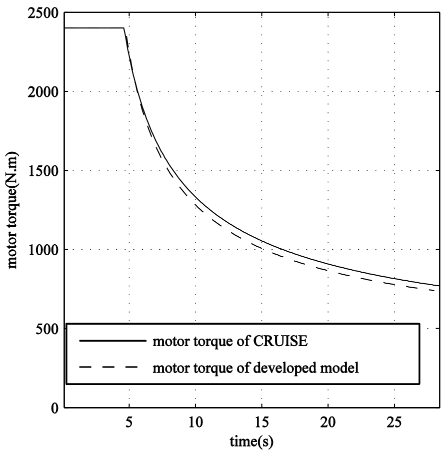

Figure 6. Motor torque comparison

V. NONLINEAR INPUT-OUTPUT MODEL

From the mechanism model of the longitudinal dynamics of the vehicle, an input-output model can be deduced.

As the torque applied to the shaft T_{shaft} is generated by the torque on the motor shaft T_{em}, taking into account the main reduction gear ratio ς_{rd} and the transmission efficiency η, the relationship between T_{shaft} and T_{em} can be express as:

$$T_{shaft} = T_{em}\varsigma_{rd}\eta \qquad (10) \qquad (10)$$

From (7), ignored the slip ratio of the tire and assume $v_{veh} = w_{wheel}r_{wheel}$ the driving force F_x can be express as:

$$F_x = \frac{T_{em}\varsigma_{rd}\eta - T_{br}}{r_{wheel}} - \frac{J_{wheel}\dot{v}_{veh}}{r_{wheel}^2} - F_{roll,fr} \quad (11) \quad (11)$$

Take (11) into (9), ignore the difference of the vertical forces acting on the front and rear wheels, the longitudinal dynamics of the vehicle can be express as:

$$M_{veh}\delta\dot{v}_{veh} = \frac{T_{em}\varsigma_{rd}\eta - T_{br}}{r_{wheel}} - F_{roll,fr} - \frac{1}{2}\rho_{air}C_d A v_{veh}^2 - M_{veh} g \sin(\gamma) \quad (12)$$

where the $\delta = 1 + \frac{\sum J_{wheel}}{r^2{}_{wheel}}$ is rotating mass conversion factor.

The vehicle is assumed to driven on straight roadway, so the road inclination γ is zero, and the rolling friction $F_{roll,fr}$ is assumed to be constant, the relationship between T_{em} and v_{veh} can be express as an input-output system:

$$a_0 + a_1 v_{veh}^2 + a_2\dot{v}_{veh} = b_1 T_{em} \qquad (13)$$

The input variable of the system is T_{em}, and the output variable is v_{veh}, and b_1, a_0, a_1, a_2 are parameters.

According to (13), the longitudinal dynamics of the vehicle is nonlinear, i.e. the relationship between the motor torque T_{em} of pure electric vehicle and the vehicle speed v_{veh} is nonlinear.

The discrete model can be obtained by discretization of (13) using forward difference scheme:

$$y(k+1) = -\theta_1 y^2(k) - \theta_2 y(k) + \theta_3 u(k) + \theta_4 \quad (14)$$

where y is v_{veh}, u is T_{em}, $\theta = [\theta_1, \theta_2, \theta_3, \theta_4]$ is the parameter vector.

The (14) shows the non-linear model is a Non-linear Auto Regressive with eXogenous inputs (NARX)model [12], and as the model depends on its parameters in linear way, it can be treated as a linear-in-the-parameters model.

The linear-in-the-parameters model takes the form:

$$y(k+1) = \theta x(k) + \varepsilon(k+1) \qquad (15)$$

where $x(k) = [-y^2(k), -y(k), u(k), 1(k)]'$.

Suppose N data samples $\{x(k), y(k)\}_{k=1}^{N}$ are used for model identification, equation (15) can be formulated as:

$$Y = \theta X + \Xi \qquad (16)$$

(where $Y = [y(1),..., y(k)]'$, $X = [x(1),..., x(k)]'$, $\Xi = [\theta_1,...,\theta_2]$.Then use least-squares (LS) method to determine the parameters.[13]

In order to identification, a pure electric bus of Anhui Ankai Automobile Co., Ltd is made to run on a straight roadway for about 125 seconds, two groups of data is recorded down as Fig.7 shows. In Figure.7, the dotted line is T_{em} and the solid line is v_{veh}, both lines are normalized against their maximum value, and the interval between data points is 50ms.

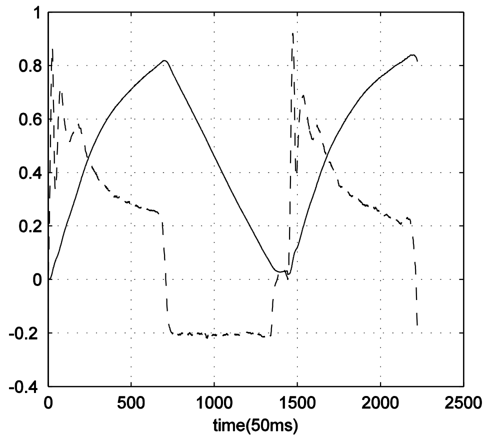

Figure 7. data collected from a pure electric bus

The previous part of the recorded data points are used for identification, and the other data points are used for calibration. Least squares method yields the following result as shown in Table II.

The calibration result is show in Fig. 8, the solid line is the data obtain from the real bus, and the dotted line is the calculated value derived by the model.

TABLE II.
ESTIMATED PARAMETERS

parameters	estimates
θ_1	0.0004
θ_2	-1
θ_3	0.004
θ_4	-0.0003

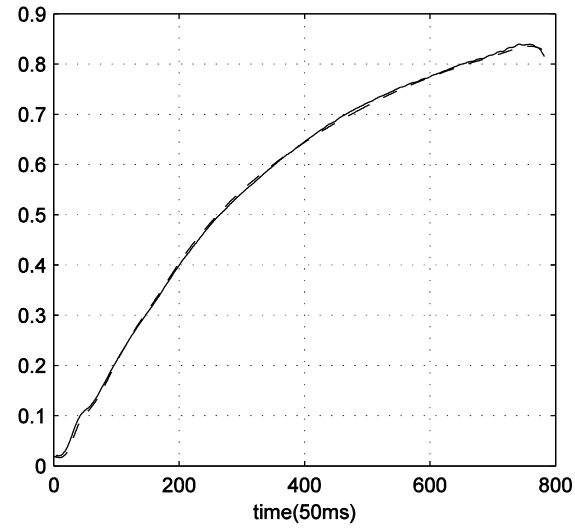

Figure 8. calibration

VI. CONCLUSION

This paper demonstrates the development and validation of a pure electrical vehicle model which is suitable for the evaluation of vehicle drivability. The model aims at describing longitudinal modes that affect drivability.

In this article, components of pure electric vehicle power system are analyzed, and a mechanistic model is built up in Simulink. The model is verified by CRUISE, and the simulation results shows good fitting effect.

According to the structure of the mechanism model, an input - output model is developed. As the input - output model is nonlinear; it can be described as Non-linear Auto Regressive with eXogenous inputs (NARX) model. Furthermore, although the relationship between the input and output of the model is nonlinear, the output depends on its parameters in linear way, so it can be treated as a linear-in-the-parameters model. Least squares method is used to estimated parameters from the data collected from a pure electric bus, and the calibration results show the high accuracy of the identification.

The primary application of the model is to assist in developing algorithms that can actively control drivability. Future work will focus on the control algorithms and their implementation in pure electrical vehicle.

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REFERENCES

[1] Tom Hazeldine, Sujith Kollamthodi, Charlotte Brannigan,Matt Morris and Laura Deller,"Market outlook to 2022 for battery electric vehicles and plug-in hybrid electric vehicles", Issue Number 4, June 2009

[2] Kerem Koprubasi, "Modeling and control of a hybrid-electric vehicle for drivability and fuel economy improvements"[D], The Ohio State University, 2008.

[3] F.Guo,"A New Identification Method for Wiener and Hammerstein System"[D], Institut für Angewandte Informatik,2004

[4] Kerem Koprubasi and Giorgio Rizzoni, "Development and experimental validation of a low-frequency dynamic model for a Hybrid Electric Vehicle"[J], Int. J. Powertrains, Vol. 1, No. 3, 2012

[5] V. T. Minh and A. A. Rashid, "International Journal of Automotive Technology"[J], Vol. 13, No. 3, pp. 477−485 ,2012.

[6] Allan Mills, Robert Hooke's "'universal joint' and its application to sundials and the sundial-clock"[J], Notes and records- royal society of London, Vol 61; Numb 2, pages 219-236,2007

[7] Kerem Bayar, "Development of a Vehicle Stability Control Strategy for a Hybrid Electric Vehicle Equipped With Axle Motors"[D], The Ohio State University, 2011.

[8] Yimin Gao, Mehrdad Ehsani, "Electronic Braking System of EV And HEV---Integration of Regenerative Braking, Automatic Braking Force Control and ABS"[J], SAE Technical Paper 2001-01-2478, 2001. http://dx.doi.org/10.4271/2001-01-2478

[9] Pacejka, Hans B. , "Tyre Modeling for Use in Vehicle Dynamics Studies"[J], Society of Automotive Engineers. 1987.

[10] Xu Zhisheng. "Automobile Theories". China Machine Press, China,2011.

[11] Farhad Sangtarash, " Effect of Different Regenerative Braking Strategies on Braking Performance and Fuel Economy in a Hybrid Electric Bus Employing CRUISE Vehicle Simulation"[J], SAE International Journal of Fuels and Lubricants April 2009 vol. 1 no. 1 828-837, http://dx.doi.org/10.4271/2008-01-1561

[12] S. Chen, S. A. Billings, "Representations of non-linear system: the NARMAX model" ,Int. J. Control, vol. 50, pp. 1873–1896, 1989. http://dx.doi.org/10.1080/00207178908953472

[13] Kang Li, Jian-Xun Peng, and George W. Irwin, "A Fast Nonlinear Model Identification Method", IEEE TRANSACTIONS ON AUTOMATIC CONTROL, VOL. 50, NO. 8, AUGUST 2005

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