

# A Modularized Design for Multi-Drivers Off-Road Vehicle Driving-Line and its Performance Assessment

Yi Jianjun, Sun Yingce, Hu Diqing, and Li Chenggang

**Abstract**—Modularized design approach can facilitate the modeling of complex systems and support behavior analysis and simulation in an iterative and thus complex engineering process, by using encapsulated submodels of components and of their interfaces. Therefore it can improve the design efficiency and simplify the solving complicated problem. Multi-drivers off-road vehicle is comparatively complicated. Driving-line is an important core part to a vehicle; it has a significant contribution to the performance of a vehicle. Multi-driver off-road vehicles have complex driving-line, so its performance is heavily dependent on the driving-line. A typical off-road vehicle's driving-line system consists of torque converter, transmission, transfer case and driving-axles, which transfer the power, generated by the engine and distribute it effectively to the driving wheels according to the road condition. According to its main function, this paper puts forward a modularized approach for designing and evaluation of vehicle's driving-line. It can be used to effectively estimate the performance of driving-line during concept design stage. Through appropriate analysis and assessment method, an optimal design can be reached. This method has been applied to the practical vehicle design, it can improve the design efficiency and is convenient to assess and validate the performance of a vehicle, especially of multi-drivers off-road vehicle.

**Keywords**—Heavy-loaded Off-road Vehicle, Power Driving-line, Modularized Design, Performance Assessment.

## I. INTRODUCTION

THE target of a modularized method to modeling of complex systems, which is based on commercial CAD/CAE technology, is to support behavior analysis and simulation in an iterative and thus complex engineering process, by using encapsulated sub-models of components and

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of their interfaces. Through this way, we can check and evaluate the assembly/disassembly, matching performance and the potential deficiencies among the components of a product at the designing stage. It is very beneficial especially for complex products.

Driving system is very important for a vehicle. A well designed driving-line will greatly improve the performances of the vehicle. As for those multi-drivers off-road vehicles, it is even more important, because the off-road performance of the vehicle is heavily dependent on it.

A typical off-road vehicle's driving-line system consists of torque converter, transmission, transfer case and driving-axles, which transfer the power, generated by the engine and distribute it effectively to the driving wheels according to the road condition. A main task during the concept design of a driving-line is to optimally match these parameters to fulfill the performance requirement by the vehicle.

Generally, it is very difficult to design a good driving-line for a certain vehicle by using present design methods; the work needs to be done by experienced engineers. After the driving-line has been designed, a series of tests are required to verify the design, and redesigns are necessary. Although some optimal methods and software tools are developed to aid the design process, they are not very effective universally to all vehicles' driving-line system, especially those multi-drivers off-road vehicles.

As an aid to conceptual developments of the driving-line system for multi-drivers off-road vehicle, efficient and validated physically based system models are necessary. The model should capture the basic features of the system, and represent the most essential system behavior.

The purpose of this paper was to develop a modular based computer model for driving-line system. This model should be complete enough to predict correct trend of behavior of a driving-line in vehicle and be simple enough to study the influence of the main design factors so as to optimally match these parameters in the driving-line system through performance assessment.

## II. RELATED WORKS

There are a lot of researchers who have researched dynamic simulation and modularized design methodology about Off-road vehicle. Andrew W. Phillips, etc. developed a personal computer-based vehicle powertrain simulation (VPS)

to predict fuel economy and performance [1]. VPS code can provide good predictions of vehicle fuel economy, and thus is a useful tool in designing and evaluating vehicle powertrains. H-S Jo, etc., presented the main algorithms of POTAS-MSM (Power Transmission Analysis Software for MultiSlipping Mechanism), which are based on the concept of subsystem assembly and the self-determination technique for the system degrees of freedom [2]. Gumlnter H. Hohl and Alexander Corrieri put forward basic considerations for the Concepts of Wheeled off-Road Vehicles [3]. Lim, W.S., Jo, H.S., Jang, W.J. etc. developed a general purpose program based on the concept of subsystem assembly for the analysis of dynamic characteristics of a power-train system [4]. This paper puts forward the modularized methodology according to the components structural features of driving-line of multi-drivers off-road vehicle. Based on these, the tools to evaluate Driving-line Performance have been developed with Visual C++ and are flexible to integrate into other CAD/CAE systems.

### III. DRIVING-LINE MODELING

The main purpose of driving-line system in a vehicle is to convert, transfer and distribute torque and rotation generated by the engine correctly and efficiently driving wheels to fulfill the kinematic requirements of the vehicle. From this point of view, the parts in the driving-line system can be abstracted as elements, which inputs from the engine, and finally outputs to the driving. Because the torque and rotation are originally input from engine, and finally output to driving wheels, the models should also include them.

#### A. Engine

The main design parameters of an engine and their relations are defined as follow [5]:

$$M = M_{e \max} - \frac{M_{e \max} - M_P}{(n_P - n_M)^2} (n_M - n)^2 \quad (1)$$

where:  $M$ --Output torque

$n$ --Output rotate speed

$M_{e \max}$ --Maximal output torque

$n_M$ --Rotate speed at maximal torque

$M_P$ --Output torque at maximal power

$n_P$ --Rotate speed at maximal power

#### B. Transmission (Reducer)

Similarly, the model of transmission or reducer is defined as:  
 Torque equation:

$$M_O = \eta i_0 M_I \quad (2)$$

Rotate speed equation:

$$n_O = n_I / i_0 \quad (3)$$

where:  $M_O$ --Output torque

$M_I$ --Input torque

$i_0$ --Ratio

$n_O$ --Output rotate speed

$n_I$ --Input rotate speed

$\eta$ --Efficiency

#### C. Torque Converter

The difficult point of the torque converter modeling lies in the fact that the torque coefficient and speed ratio are non-linear, otherwise it can be simply regarded as a reducer. The input and output relationships are [6]:

Torque equation:

$$M_O = k_C M_I \quad (4)$$

Rotating speed equation:

$$n_O = i_C n_I \quad (5)$$

Constraint equation:

$$k_C = \sum_{k=0}^{N_C} B_k i_C^k \quad (6)$$

where:  $M_O$ --Output torque

$M_I$ --Input torque

$k_C$ --Torque coefficient

$n_O$ --Output rotate speed

$n_I$ --Input rotate speed

$i_C$ --Speed ratio

$B_k$ --Polynomial coefficient

$N_C$ --Degree of the polynomial

#### D. Transfer Case

The equations for the transfer case are defined as:

Torque equation:

$$M_{O1} = w M_{O2} \quad (7)$$

Rotating speed equation:

$$n_{O1} = n_{O2} = n_I / i \quad (8)$$

Constraint equation:

$$M_I \geq \frac{M_{O1} + M_{O2}}{\eta i} \quad (9)$$

where:  $M_{O1}$ --Front output torque

$M_{O2}$ --Rear output torque

$w$ --Torque bias ratio

$M_I$ --Input torque

$n_{O1}$ --Front rotating speed

$n_{O2}$ --Rear rotating speed

$n_I$ --Input rotating speed

$i$ --Ratio

$\eta$ --Efficiency

#### E. Differential

The equations for the differential are defined as:

Torque equation:

$$\max(M_{O1}, M_{O2}) = k_D \min(M_{O1}, M_{O2}) \quad (10)$$

Rotating speed equation:

$$n_I = \frac{n_{O1} + n_{O2}}{2} \quad (11)$$

Constraint equations:

$$M_I \geq \frac{M_{O1} + M_{O2}}{\eta} \quad (12)$$

$$k_D = \sum_{k=0}^{N_K} D_k \Delta n_O^k \quad (13)$$

where:  $M_{O1}$ --Left output torque

$M_{O2}$ --Right output torque

$M_I$ --Input torque

- $k_D$ --Torque bias ratio
- $n_{O1}$ --Left rotating speed
- $n_{O2}$ --Right rotating speed
- $n_I$ --Input rotating speed
- $D_k$ --Polynomial coefficient
- $N_k$ --Degree of the polynomial
- $\Delta n_0$ --Rotating speed difference
- $\eta$ --Efficiency

#### F. Wheel

The equations for the wheel are defined as:  
 Force equations:

$$F_f = \mu F_p \quad (14)$$

$$F_t = \frac{M_I \eta}{R} \quad (15)$$

Speed equation:

$$v = \frac{\pi n_I (1 - \delta)}{108} \quad (16)$$

Constraint equation:

$$F_f \leq F_t \leq \phi F_p \quad (17)$$

- where:  $F_f$ --Rolling resistance  
 $F_p$ --Normal force  
 $F_t$ --Circumferential force  
 $M_I$ --Input torque  
 $R$ --Rolling radius  
 $v$ --Speed  
 $n_I$ --Input rotating speed  
 $\mu$ --Coefficient of rolling resistance  
 $\delta$ --Slip rate  
 $\phi$ --Coefficient of adhesion  
 $\eta$ --Efficiency

#### G. The Whole Driving-Line Model

The whole driving-line is the combination of the above mentioned parts' models. According to the property of the model, it is convenient to represent it in a computer using the object-oriented programming method. So it is very flexible to model the driving-line system according to a certain design.

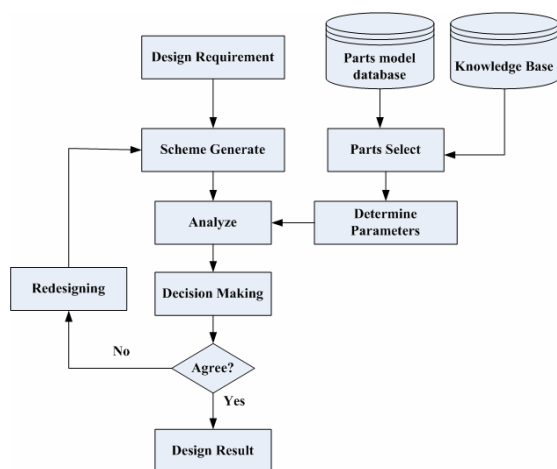


Fig. 1 Conceptual design procedure

#### IV. CONCEPT DESIGN OF DRIVING-LINE

How to select parts according to the design requirements of the driving-line is the first step that must be considered during the concept design stage. Then the design parameters of every these parts should be determined, and the design is finished after all these parameters are optimally matched.

Here modular design methodology is used based on the driving-line model, and some heuristic rules are provided. The design procedure is divided into four steps, which is shown in Fig. 1, they are driving-line scheme generation, performances analysis, decision making and redesign.

In this paper, only performances analysis is particularly described in the following section.

#### V. PERFORMANCE ASSESSMENT

In order to optimally match the design parameters of a driving-line system during the design, performance assessment is essential.

##### A. Assessment Indices

Usually validity, safety, reliability and economy are four basic indices to an off-road vehicle [7]. Among them, the validity index is the most important measured performance to a ground vehicle.

According to the particularity of off-road vehicle, the following indices are chosen for assessment. They are tractive and velocity performances, turnability and ride stability.

Tractive and velocity performances define the ability of vehicles to move under the influence of circumferential wheel forces. Ususally tractive and velocity performances are considered together with fuel economy. It is described by the transport efficiency  $\eta_x$  [8]:

$$\eta_x = \frac{\sum_{i=1}^m (F_{fi}' + F_{fi}'')}{\sum_{i=1}^n [F_{pi}' / (1 - \delta_i') + F_{pi}'' / (1 - \delta_i'')] } \quad (18)$$

Circumferential force distribution influences the turnability of vehicle. Turnability is an ability to change the trajectory according to changes in construction and working conditions. The resistant torque ( $M_R$ ) is used to estimate the turnability:

$$M_R = \sum_{i=1}^n (F_{ti}' - F_{ti}'') \times \frac{B}{2} \quad (19)$$

where B is the wheel-base of the vehicle.

##### B. Calculation of the Indices

According to the equation (18) and (19), the circumferential force on each driving wheel should be calculated out first before any of the indexes can be gained. It is a very complex task for a multi-drivers off-road vehicle's driving-line. The following algorithm is used to automatically calculate circumferential force on each driving wheel under any running condition of the vehicle imposed on the driving-line model.

Step 1: Give working condition; calculate rolling resistance and adhesive force on each driving wheel;

Step 2: Let the circumferential force be equal to the

adhesive force, from wheels to engine inversely calculate every parts' necessary output torque and rotating speed in the system which will provide the needed circumferential force;

Step 3: To each part, know the output, seek the input; input torque is gained by torque equation, input for rotating speed is gained by rotating speed equation, and the constraint equations should be satisfied as well;

Step 4: Repeat step 2 and step 3 until the engine is reached;

Step 5: If the part is engine, judge whether the output torque or rotating speed is out of the range; if not, go to step 9; otherwise continue;

Step 6: Start with the engine, from engine to wheels, output torque and rotating speed of each part in the driving-line can be calculated;

Step 7: To each part, know the input, calculate the output; output torque is get by torque equation, the output for rotating speed is gained by rotating speed equation, all constraint equations should be satisfied;

Step 8: Compare the output torque with the torque obtained during the inverse seeking; choose a small one, repeat step 7 and 8;

Step 9: The algorithm is ended after all parts have been calculated, the circumferential forces are gained.

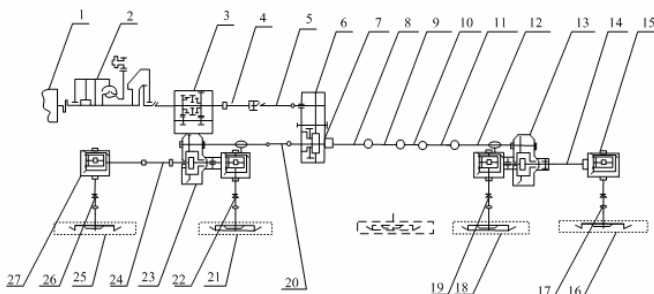


Fig. 2 An example of a 10×8 heavy off-road vehicle's driving-line  
 1 Engine & Torque Converter 3 Transmission 6 Transfer Case 7 Grip  
 Brake 13,18,19 4th Driving-axle 15,16,17 5th Driving-axle 21,22,23  
 2nd Driving-axle 25,26,27 1st Driving-axle  
 2,4,5,8,9,10,11,12,14,20,24 Shafts & Bearings

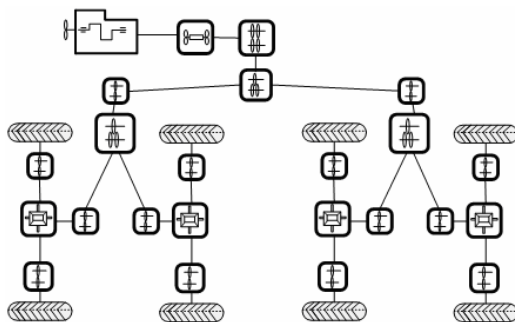


Fig. 3 Model of the driving-line

## VI. AN EXAMPLE

A 10×8 heavy off-road vehicle's driving-line system is chosen for an example (see Fig. 2). Fig. 3 is its model after the conceptual generation.

### A. Calculate Rolling Resistance and Adhesive Force

As you can see from the previous section, the first step for performances assessment during the design stage is to calculate rolling resistance and adhesive force by the given driving conditions and design requirements. These are gained from the dynamic modeling of the vehicle.

Refer to the typical Ackerman steering geometry, the 10×8 vehicle's dynamic model is shown in Fig. 4. According to the figure, the following equations are defined.

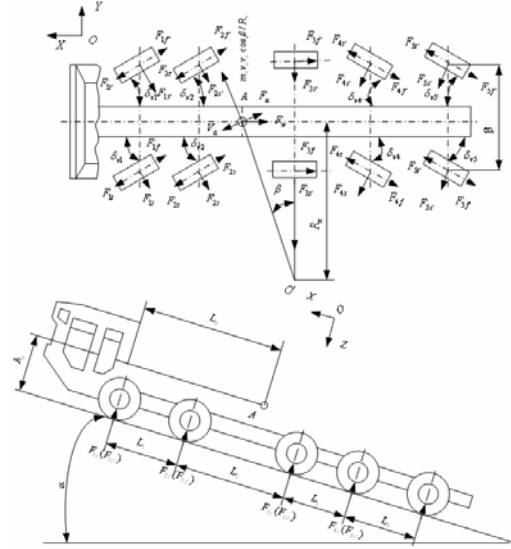


Fig. 4 10×8 vehicle's dynamic model diagram

CG (Center of Gravity) kinematic equation along longitudinal direction:

$$\begin{aligned}
 & m_a(1 + \delta_\varphi)\dot{v}_a \cos(\beta) - \frac{m_a v_a^2 \sin(\beta)}{R_n} \\
 & - \sum_{i=1}^4 [F_{it}' \cos(\delta_{iv}') + F_{it}'' \cos(\delta_{iv}'')] \\
 & + \frac{C_D A v_a^2 \cos(\beta)}{21.15} + m_a g \sin(\alpha) \\
 & + \sum_{j=1}^5 [F_{jf}' \cos(\delta_{jv}') + F_{jf}'' \cos(\delta_{jv}'')] \\
 & + \sum_{j=1}^5 [F_{js}' \sin(\delta_{jv}') + F_{js}'' \sin(\delta_{jv}'')] = 0
 \end{aligned} \tag{20}$$

CG kinematic equation along lateral direction:

$$\begin{aligned}
 & m_a(1 + \delta_\varphi)\dot{v}_a \sin(\beta) + \frac{m_a v_a^2 \cos(\beta)}{R_n} \\
 & - \sum_{i=1}^4 [F_{it}' \sin(\delta_{iv}') + F_{it}'' \sin(\delta_{iv}'')] \\
 & + \frac{C_D A v_a^2 \sin(\beta)}{21.15} \\
 & + \sum_{j=1}^5 [F_{jf}' \sin(\delta_{jv}') + F_{jf}'' \sin(\delta_{jv}'')] \\
 & - \sum_{j=1}^5 [F_{js}' \cos(\delta_{jv}') + F_{js}'' \cos(\delta_{jv}'')] = 0
 \end{aligned} \tag{21}$$

CG kinematic equation along vertical direction:

$$m_a \ddot{z}_A - m_a g \cos(\alpha) + \sum_{i=1}^5 (F_{iP}' + F_{iP}'') = 0 \quad (22)$$

Yaw moment equation:

$$\begin{aligned} \Theta_Z \ddot{\psi}_Z - \sum_{i=1}^4 [F_{iI}' \cos(\delta'_{iv}) + F_{iI}'' \cos(\delta''_{iv})] \frac{B}{2} \\ - \sum_{i=1}^4 [F_{iI}' \sin(\delta'_{iv}) + F_{iI}'' \sin(\delta''_{iv})] d_i \\ - \sum_{j=1}^5 [F_{jS}' \sin(\delta'_{jv}) + F_{jS}'' \sin(\delta''_{jv})] \frac{B}{2} \\ - \sum_{j=1}^5 [F_{jF}' \cos(\delta'_{jv}) + F_{jF}'' \cos(\delta''_{jv})] \frac{B}{2} \\ - \sum_{j=1}^5 [F_{jS}' \cos(\delta'_{jv}) + F_{jS}'' \cos(\delta''_{jv})] d_i \\ - \sum_{j=1}^5 [F_{jF}' \sin(\delta'_{jv}) + F_{jF}'' \sin(\delta''_{jv})] d_i = 0 \end{aligned} \quad (23)$$

Pitch moment equation:

$$\begin{aligned} \Theta_Y \ddot{\psi}_Y - \sum_{j=1}^5 [F_{jP}' + F_{jP}'] d_i \\ + \sum_{j=1}^5 [F_{jS}' \sin(\delta'_{jv}) + F_{jS}'' \sin(\delta''_{jv})] h_A \\ + \sum_{j=1}^5 [F_{jF}' \cos(\delta'_{jv}) + F_{jF}'' \cos(\delta''_{jv})] h_A \\ + \sum_{i=1}^4 \{ [F_{iI}' \cos(\delta'_{iv}) + F_{iI}'' \cos(\delta''_{iv})] h_A \} = 0 \end{aligned} \quad (24)$$

Roll moment equation:

$$\begin{aligned} \Theta_X \ddot{\psi}_X - \sum_{j=1}^5 [F_{jP}' - F_{jP}'] \frac{B}{2} \\ - \sum_{j=1}^5 [F_{jS}' \cos(\delta'_{jv}) + F_{jS}'' \cos(\delta''_{jv})] h_A \\ + \sum_{j=1}^5 [F_{jF}' \sin(\delta'_{jv}) + F_{jF}'' \sin(\delta''_{jv})] h_A \\ + \sum_{i=1}^4 \{ [F_{iI}' \sin(\delta'_{iv}) + F_{iI}'' \sin(\delta''_{iv})] h_A \} = 0 \end{aligned} \quad (25)$$

### B. Performances Assessment

By using the above mentioned method, three scenarios of the driving-line's design concept had been compared within five typical working conditions of the vehicle. The Parameters of these three scenarios are given as Table I.

TABLE I  
PARAMETERS OF THREE SCENARIOS

Parameters	No.1	No.2	No.3
$i_0$	4.55	3.36	3.97
$K_c$	2.16	2.37	2.21
$i_{g1}$	1.53	1.69	1.57
$i_w$	5.30	5.61	5.48
$t_e$ (N.m)	1599	1599	1599
$n_e$ (rpm)	2259	2259	2259

**Note:**  $i_0$  is the main ratio of transmission;  $K_c$  is torque efficient of torque converter;  $i_{g1}$  is ratio of the first transfer case;  $i_w$  is ratio of wheel reducer;  $t_e$  is output torque of engine;  $n_e$  is output for engine rotating speed; and the other parameters are the same.

TABLE II  
WORKING CONDITIONS OF VEHICLE

Case	Conditions
1	Front two driving axles' coefficient of adhesion is smaller than the rears'
2	Front two driving axles' coefficient of adhesion is bigger than the rears'
3	On split road surface
4	Cornering on normal road surface
5	Cornering on split road surface

TABLE III  
PERFORMANCE ASSESSMENT RESULT

Condition	Indices	No. 1	No. 2	No. 3
Case 1	Transport efficiency	0.148	0.148	0.148
	Resistant torque(Nm)	0	0	0
Case 2	Transport efficiency	0.148	0.148	0.148
	Resistant torque(Nm)	0	0	0
Case 3	Transport efficiency	0.565	0.406	0.235
	Resistant torque(Nm)	2400	22200	73200
Case 4	Transport efficiency	0.059	0.059	0.092
	Resistant torque(Nm)	0	0	70611
Case 5	Transport efficiency	0.23	0.23	0.345
	Resistant torque(Nm)	3688	3688	804

The working conditions are listed in Table II. We use the tools which we developed for the analyzing on power flowchart of Multi-drivers Off-road Vehicle Driving-line based on Objected-oriented ideas [9] [10] and analysis result is shown in Table III. We assessed the three design schemes based on five working conditions. From the Table II, although case 1, case 2, the results are same. But case 3,4,5, their difference are evident. Considering the comprehensive performance is the best scheme is No.1.

### VII. SUMMARY

The models described in this paper have provided a convenient measure to aid the concept design and performance analysis for multi-drivers off-road vehicle's driving-line. This tool has been used in the practical products developing of Special Vehicle Technical Center of China Sanjiang Space Group [9] [10]. Some related papers [11] [12] have shown the experimental results in detailed based on this project.

In this paper, multi-drivers Off-road Vehicle Driving-line is designed based on the modularized methodology, Object-oriented method is very suitable for analyzing its performance of torque, power, speed flow. Therefore we use Visual C++ to develop these analyzing tools. Compared the

other similar tools [1] [2][4], the drive-line model can be built easily with the components library. The components of built model will be connected via input/output interface and their performance parameters can be displayed, checked or modified by users conveniently. Additionally, the tool can be integrated into an Intelligent CAD Expert System which is specially developed for multi-drivers Off-road Vehicle Driving-line design, including scheme design, conceptual design; structural and topological design and so on. From its application in Special Vehicle Technical Center of China Sanjiang Space Group for recent several years, it has been proved to a useful, economical tool to facilitate the vehicle's driving-line system design.

Furthermore, it can be developed as a computer simulation tool to save a lot of test needed during the design stage of the vehicle development. It also can be used for traction control system design and evaluation for multi-drivers off-road vehicle, which is the work we are doing now.

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