Multidisciplinary Design Optimization Method for Steering System of Large Wheeled Harvester

Zhongxiang Zhu[1, a], Yuefeng Du[1], Hao Li[1], Zhenghe Song[1, b, *]

[1] Beijing Key Laboratory of Optimization Design for Modern Agricultural Equipment, China Agricultural University, Beijing, 100083, China

[a]email: zhuzhonxiang@cau.edu.cn, [b]email: songzhenghe@cau.edu.cn

*corresponding author

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Abstract. To improve the design quality of steering system of the large wheeled harvester in China, an optimization method based on multidisciplinary design optimization (MDO) is proposed. According to the requirement of how to design and optimize the large wheeled harvester, optimal variables consisting of steering wheel toe-in angle, kingpin caster angle, kingpin inclination angle, steering wheel camber angle, length of steering trapezoid arm, angle between steering trapezoid and forward direction, distance from the intersection point between vertical axis and guide wheel axis to the intersection point between vertical axis and guide wheel split, driving wheel cornering stiffness coefficient and driven wheel cornering stiffness coefficient are chosen. Take error between the inside and outside wheel angles, dynamic response error between driver input and vehicle response, rolling resistance coefficient as optimal objective, a MDO model is established based on the constraints of steering stability, returnability character, steering portability and design specifications. The hierarchical level two systems integrated algorithm is taken as the multidisciplinary solution strategy after the planning and decomposition by the non-hierarchical structure. Finally, he MDO method for steering system of large wheeled harvester is validated by a certain design example.

Introduction

As an important component of the large wheeled harvester (LWH) chassis, steering system has an important influence on the handling stability and efficiency [1]. Reasonable and accurate design of steering system plays a significance role on improving the overall performance of LWH. At present, the product design methods for mechanical steering system of LWH are relatively backward [2], and the product development is still stay in experiential design stage [3]. Only a small handful of optimization design method is applied to the practical design, which is due to the complex structure of steering system of the LWH, various knowledge and types involved, diverse and complex mathematical model, insufficient understanding of performance optimization design in product cycle life. These lead to overlong development cycle and low quality of the mechanical steering system of LWH in China, and the developed products are difficult to compete effectively with foreign ones.

Multidisciplinary design optimization (MDO) can integrate the whole knowledge of each subsystem in the process of complex system design. Applying effective design and optimization strategy, making full use of the synergetic effects of interaction among each subject, the overall optimal solution system (that is, the better product quality or performance) can be obtained [4]. By using MDO, the design cycle is shortened so that the developed product becomes competitive in the international market.

Combining with the optimization design characteristics of LWH steering system, a MDO model for mechanical steering system of LWH is established in this paper. Based on the optimizing strategy of the nonhierarchical secondary system integrated algorithm, the MDO method for LWH steering system is proposed.

MDO Modeling of LWH Steering System

a) Design process description model

In the design process description model of LWH steering system, the commonly used hierarchy tree structure is applied to describe the MDO process. The model includes a complete design process, design objects and the performance function tree. The LWH steering system consists of steering gear and the steering linkage mechanism. The established design object tree is shown in Fig. 1. Considering the structure characteristics of LWH steering system and the influence of multiple handling and stability perform, the performance function tree for steering system is set up, as shown in Fig. 2.

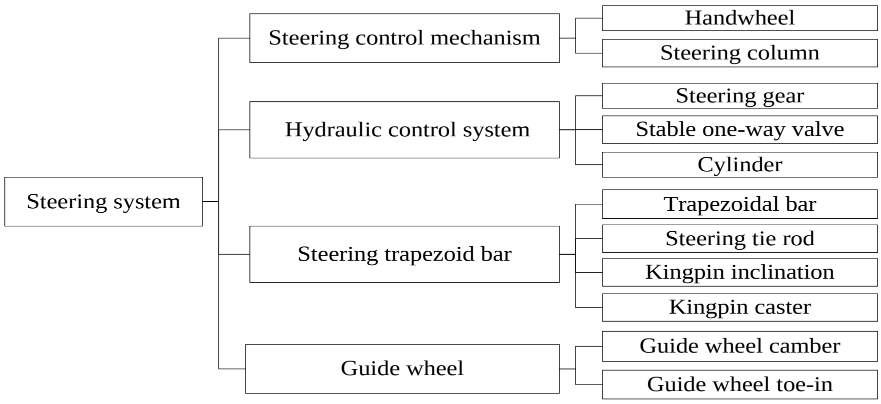

Fig.1. Design object tree of LWH steering system

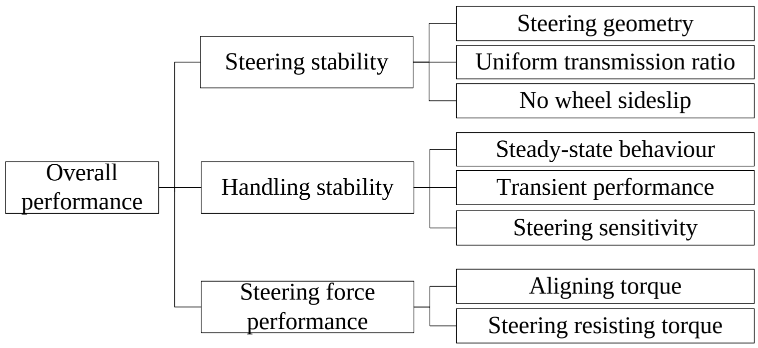

Fig.2. Performance function tree of LWH steering system

b) Design parameters and functions

The MDO parameters X_L for LWH steering system is selected as shown in Eq. 1, which includes toe-in ΔW, caster angle τ, kingpin inclination angle σ, guide wheel camber angle γ, the length of steering trapezoid arm l_3, the angle between steering trapezoid arm and forward direction φ_{st}, the distance from the intersection point between vertical axis and guide wheel axis to the intersection point between vertical axis and guide split e, the cornering stiffness coefficient of driving wheel $c_{f\alpha}$, the cornering stiffness coefficient of steering wheel $c_{r\alpha}$. All design parameters are obtained when the LWH run straight.

$$X_L = [x_1, x_2, \cdots, x_7] = [\Delta W, \tau, \sigma, \gamma, l_3, \varphi_{st}, e, \lambda, c_{r\alpha}, c_{f\alpha}] \quad (1)$$

Some main parameters are input by design interfaces of LWH or other subsystem. These parameters include the total weight m, wheelbase L, the dynamic radius of steering wheel r_e, height of center of gravity h_G, the distance between the two steering knuckles l_1, the camber stiffness coefficient of steering wheel c_γ, the distance H_{z1} from rocker shaft axis to the line whose two ends are the intersection points between the steering wheel axes and the corresponding side vertical axes.

The relationship among part intermediate input parameters, the design parameters and the system input parameters is shown in Eq. 2.

$$\begin{cases} B_r = l_1 + 2r_e \tan\sigma\cos\gamma \\ \varphi_0 + \varphi_D = \dfrac{\pi}{2} - \varphi_{st} \\ l_{\sigma 0} \approx e - r_e\gamma \\ h_{z10} = H_{z1} + r_e\cos\gamma \\ \delta_0 = \arcsin\dfrac{\Delta W}{4r_e} \end{cases} \tag{2}$$

Design function, namely performance function, consists of the constraints and objectives. The classification of the performance function produces the performance function tree [4]. Objective classification is similar to the constraint classification, including function, structure, manufacturing costs and social benefits and other various demands. In the design process, constraints and objectives can mutual transformation. That is to say in certain cases, a constraint can be used as an objective, and an objective can also be turned into a constraint [4]. For the purpose of unity, the constraint function and objective function are collectively referred to as the performance function. According to properties, engineering constraints in the design process are mainly from three aspects: function, structure and manufacturing. In the MDO of LWH steering system, the functional and structural constraints are considered, and the manufacturing constraints are not considered temporarily. Main function constraint on the requirements of product performance, which is often given in advance, is used to reflect the limit and relation among the functional characteristics. Performance constraint belongs to this type of constraint, such as the requirement of mechanical steering gear ratio of LWH, etc. Structure constraint is the mutual restriction and relation among shape, product structure, and product function. This type of constraint mainly includes the geometric constraints and structural static and dynamic constraints of product structure (such as strength, stiffness, frequency, vibration mode and stability, etc.).

(i) Optimization design objective

According to the design requirements of LWH steering system, combining with the characteristics of LWH, the optimization design objective of the LWH steering system is described with three aspects:

(1) In order to reduce the power consumption, tire wear and ground resistance, and improve the maneuverability, basic requirements for LWH is put forward to ensure that every wheel rolls without sliding (including sideslip, longitudinal slip and trackslip) [5].

(2) With good handling stability. The speed of LWH is lower than car, therefore, its steering stability problem is not so significant, it won't appear "high-speed drifting", completely "out of control" situation, etc. The LWH steering stability problems, is mainly reflected in the accurate response to pilot inputs. At the same time, the stability criterion should be satisfied, namely with understeer and reasonable damping coefficient.

(3) Due to the large mass of LWH and operation to maintain good linear driving ability, it is required that the LWH should has enough returning moments to overcome the steering resistance. At the same time rolling resistance moment should be as small as possible.

Based on the above three objectives, taking the error between the actual and theoretical wheel angles, the error between the driver input and dynamic response of LWH [6] and rolling resistance coefficient as the optimization goal, the expression of the objective function is determined, as shown in Eq. 3.

$$\begin{cases} f_1(x) = E_{AD} = \sqrt{\dfrac{1}{n}\sum\limits_{i=1}^{n}\left(\delta_{Do} - \delta_{Ao}\right)^2} \\ f_2(x) = E_{HS} = \dfrac{T_2 + \left(T_1 - \tau_1\right)^2}{2T_1} \\ f_3(x) = E_T = \dfrac{1}{Q_r n}\sum\limits_{i=1}^{n}\left(T_{rRy} + T_{rLy}\right) \end{cases} \tag{3}$$

(ii) Constraints of optimization design

In the design process of large mechanical steering system, the source of constraints knowledge is very complicated. Generally there are three types to acquire knowledge: factual knowledge, standard class knowledge and dynamic knowledge, the first two relative to the third can be referred as static knowledge.

The standard constraints:

(1) As the full hydraulic steering, one-side turn angle is commonly 2~2.5 loops.

(2) Steering trapezoid arm l_3, set the value as 0.12 to 0.18 times of the distance between the two steering knuckle vertical shaft according to the experience. When it comes to front-load trapezoidal structure, the intersection angle φ_{st} between l_3 and harvest machinery direction takes 15° ~ 20°, generally.

(3) For agricultural harvest machinery, the value of steering knuckle vertical shaft inside angle usually takes the range of 3° ~ 8°, the value of steering knuckle vertical shaft caster angle takes range of 0 ~ 5° and the value of steering wheel camber angle takes range of 2°~ 4°, the value of toe-in takes range of 3-10 mm, and the value of offset distance of steering wheel vertical shaft takes range of 0-100 mm [7].

(4) According to the relevant design experience, the front axle load distribution coefficient of LWH generally range from 80%/20% to 45% / 20%.

(5) The tire cornering stiffness coefficient is proportional to the charge pressure. By referring to the tire standards, under the same bearing capacity, wheeled harvest machinery tire inflation pressure value as follow: the front wheel range from 110 kPa to 280 kPa, the rear wheels range from 200 kPa to 400 kPa, respectively. The value range of the tire cornering stiffness coefficient can be estimated according to the empirical formula [8].

$$\begin{cases} g_1 = \left(x_7 \cos x_4 - r_e \sin x_4 - \left(r_e \cos x_4 + x_7 \sin x_4\right) \tan x_3\right) \le 0.1 \\ g_2 = \sqrt{\dfrac{1}{n}\sum_1^n \left| \left(\dfrac{d\varphi_{O'}}{d\varphi}\right)_i \dfrac{A_{p2}}{A_{p1}} - 1 \right|^2} \le \varepsilon_1 = 0.01 \\ g_3 = K \ge 0 \\ g_4 = \zeta_n \ge 0 \\ g_5 = \left(H_{z1} + r_e \cos x_4\right) \le r_f \\ g_6 = i_{\omega 0R} \ge 22.5 \\ g_7 = i_{\omega 0L} \ge 22.5 \\ g_8 = \left(\dfrac{x_8 + \Delta a}{fL}\right) \ge 1.428 \\ g_9 = \Delta T = T_{rLz} + T_{rRz} - T_{rLy} - T_{rRy} \ge 0 \\ 0.003 \le x_1 \le 0.01 \\ 0 \le x_2 \le \dfrac{\pi}{36} \\ \dfrac{\pi}{60} \le x_3 \le \dfrac{2\pi}{45} \\ \dfrac{\pi}{90} \le x_4 \le \dfrac{\pi}{60} \\ 0.12 l_1 \le x_5 \le 0.18 l_1 \\ \dfrac{\pi}{12} \le x_6 \le \dfrac{5\pi}{36} \\ 0 \le x_7 \le 0.3 \\ 0.2 \le x_8 \le 0.45 \\ 33.7 \le x_9 \le 85.8 \\ 29.6 \le x_{10} \le 59.3 \end{cases} \tag{4}$$

The dynamic constraints:

In this paper, the dynamic constraint relationship of LWH was established by mathematical derivation and numerical simulation.

(1) The steering ratio is different because the different area of piston rod of hydraulic cylinder when vehicle steering by simulation analysis. For purpose of eliminating this difference and reducing the inconvenience to driver, equal ratio of the left and right steering gear is selected in the process of design. In order to make the steering wheel turns satisfy the requirement of less than 2.5 loops, the maximum angle of steering wheel generally set at 40 °, and the transmission ratio should be greater than or equal to 22.5 [9].

(2) The vertical offset distance of steering is expressed by Eq. 5 when the harvest machinery runs in straight line. The Eq. 5 can be used as a dynamic constraint.

$$r_\sigma = e\cos\gamma - r_e\sin\gamma - \left(r_e\cos\gamma + e\sin\gamma\right)\tan\sigma \tag{5}$$

(3) According to the requirement of the steering stability analysis for harvest machinery, the stability response coefficient K should be greater than zero, ensuring that the harvest machinery is of understeer characteristics; the relative damping coefficient ξ_n should be greater than zero to satisfy the stability criterion in the process of transient response [10].

(4) In order to ensure that the wheel mechanical harvester has good trafficability, it should maintain a certain adhesion, namely, the adhesion should be greater than rolling resistance when the wheel mechanical harvester is full load.

MDO Rules

Decomposition planning of MDO can be divided into hierarchical and non-hierarchical rules. The MDO model of the LWH steering system established in this paper is typical of a non-hierarchical structure. The planning is carried out in accordance with the performance. Choosing three design targets for the optimal design of subsystem, the subsystems of D_1 to D_3 with non-hierarchical relationship are established as shown in Fig.3. The D_i denotes the i^{th} subtask, and each task represents one performance of large wheel mechanical steering, which is also called a subsystem or sub discipline. x^i is local variables of D_i and has no direct relationship to other subsystems. x is system variable or cross variable set, and is also a optimization variable to multiple systems. r^i is middle state variable for D_i. The middle state variables is set as derived parameters in design objects and is also a function of system input variables or design variables. y^{ij} stands for relevant variables of D_i, which affecting the relationship of multiple disciplines. X^i stands for parameter set of D_i including local variables, derivation and coupling variables. f^i stands for target function set of the subtasks D_i, and g^i stands for constraint functions of subtasks D_i.

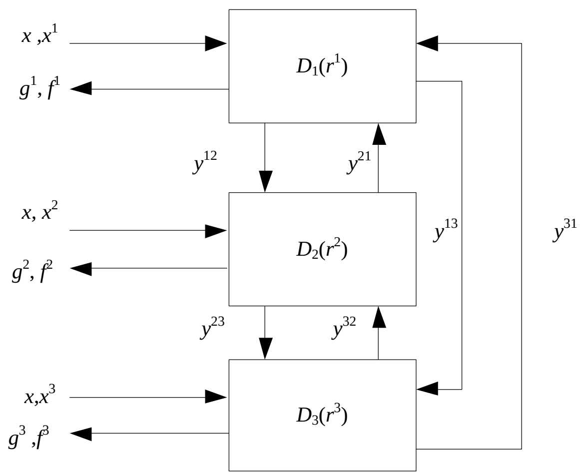

Fig.3. Non-hierarchical system with three subsystems

The hierarchical planning needs to clarify the independent variables, system variables, related variables and constraint function of different subtasks. The different level optimization models of LWH are expressed by Eq. 5, Eq. 6 and Eq. 7, respectively.

$$\begin{cases} \text{Find} \quad [x_2, x_3, x_5, x_6] \\ \min \quad f_1(x) = E_{AD} = \sqrt{\dfrac{1}{n}\sum_{i=1}^{n}\left(\delta_{Do} - \delta_{Ao}\right)^2} \\ s.t. \\ g_2, g_6, g_7 \end{cases} \tag{5}$$

$$\begin{cases} \text{Find} & [x_1, x_2, x_3, x_4, x_7, x_8, x_9, x_{10}] \\ \min & f_1(x) = E_{HS} = \dfrac{T_2 + (T_1 - \tau_1)^2}{2T_1} \\ s.t. & \\ g_1, g_3, g_4 & \end{cases} \tag{6}$$

$$\begin{cases} \text{Find} & [x_1, x_2, x_7, x_8, x_{10}] \\ \min & f_3(x) = E_T = \dfrac{1}{Q_r n} \sum_{i=1}^{n} (T_{rRy} + T_{rLy}) \\ s.t. & \\ g_5, g_8, g_9, g_{10} & \end{cases} \tag{7}$$

Decomposition Planning Modeling of MDO for LWH Steering System

There are six kinds of methods used in MDO, such as multidisciplinary feasible method (MDF, also known as the All - In - One, AIO), single subject feasible method (IDF), simultaneous analysis and design (SAD, also called All At One AAO), concurrent subspace optimization (CSSO), collaborative optimization (CO), and bi - level integrated system short (BLISS) [11]. Multidisciplinary optimization of the LWH steering system is a typical non-hierarchical, strong coupling design problem as shown in Fig. 4. At present, the advanced and mature methods to addressed hierarchy problem are CO and BLISS. CSSO is unfit for processing the multivariate problems, and CO is not applicable to process the problem of variable coupling more serious. Accordingly, the BLISS method was presented by Sobieski. Although the calculation efficiency can be improved with BLISS method, the subject is stripped of some processing ability and the subject layer only serves as analyzer of the system layer even in some stages. The BLISS is selected as the optimizing method for the LWH steering system according to the analysis on those architectures. During the solutions of BLISS, it is need to construct a system layer to get the optimal or satisfactory solution of the whole system by coordinating the coupling relationship among different disciplines. The principle of constructing the system layer is able to effectively deal with all the coupling relationship between disciplines inconsistencies.

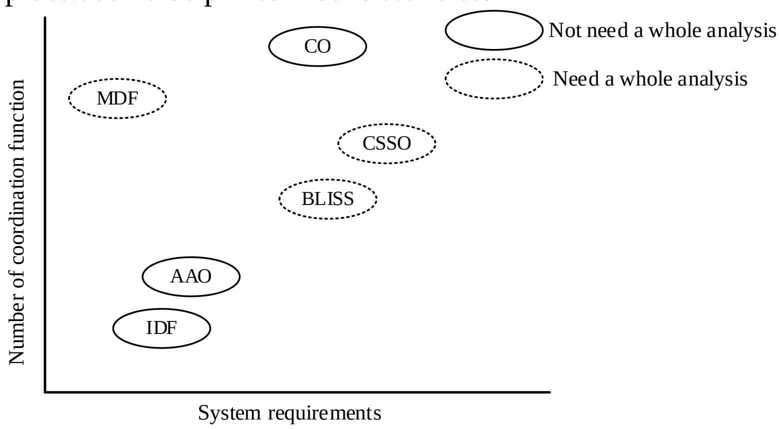

Fig.4. Comparison of each solving method

Taking a certain design as an example, the MDO system layer of LWH is built and expressed by Eq. 8. Target weights of the three disciplines are 1/3, x_2 stands for the system variables, x_3 is coupled variables of subject 1 and subject 2, and x_1, x_7, x_8, x_9 and x_{10} are coupled variables of subject 2 and subject 3. The variables g_1, g_3, g_4 and g_6 ~ g_8 are chosen as constraints, and the unused variables will be removed from the multi-disciplinary layer.

$$\begin{cases} \min & F = \dfrac{100}{3}\left(\dfrac{f_1}{14.238} + \dfrac{f_2}{3.5515} + \dfrac{f_3}{80.3}\right) \\ s.t. & G_i \le 0, i = 1,3,4,6,7,8 \end{cases} \tag{8}$$

Coupling factors are calculated according to solutions of system layer, which are delivered to the subject 1 and subject 2, and then the subsystems update values and the values return to the system layer. The solution repeats until the results become convergent. The iteration results are listed in Tab. 1.

Tab.1. Result of BLISS method

Parameters	Range	Initial value	Iteration results
x_1	[3,10]	9.2	5
x_2	[0,0.087]	0	0.083
x_3	[0.052,0.1396]	0.093	0.053
x_4	[0.0349,0.0523]	0.0349	0.052
x_5	[0.161,0.241]	0.183	0.240989
x_6	[0.261,0.436]	0.305	0.261
x_7	[0,0.3]	0.146	0.21
x_8	[0.2,0.45]	0.28	0.2
x_9	[33.7,85.8]	85.8	33.7
x_{10}	[29.6,59.3]	57.83	29.6
	System level		0.12210
	Subject 1		0.001411
	Subject 2		0.2332

Conclusion

In this paper, taking error between the inside and outside wheel angles, dynamic response error between driver input and vehicle response, rolling resistance coefficient as optimal objective, a MDO model is established based on the constraints of steering stability, returnability character, steering portability and design specifications. According to the requirement of how to design and optimize the large wheeled harvester, optimal variables consisting of steering wheel toe-in angle, kingpin caster angle, kingpin inclination angle, steering wheel camber angle, length of steering trapezoid arm, angle between steering trapezoid and forward direction, distance from the intersection point between vertical axis and guide wheel axis to the intersection point between vertical axis and guide wheel split, driving wheel cornering stiffness coefficient and driven wheel cornering stiffness coefficient are chosen. By comparing the existing multidisciplinary method, selecting BLESS for solving algorithm, convergent solutions are obtained.

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References

[1] Chinese Academy of Agricultural Mechanization Sciences: Handbook of Agricultural Machinery Design: Volume Two (Agricultural Science and Technology of China Press, China 2007) (in Chinese)

[2] Jie Li, Chuliang Yan, Fangfei Yang: Transactions of the Chinese Society for Agricultural Machinery, Vol. 37 (10) (2006), p. 74-76,135. (in Chinese)

[3] Hao Haiqing: Farm Machinery, Vol. 1 (2009), p. 54-56. (in Chinese)

[4] Yifang Zhong, Baihong Chen, Zhonghong Wang: Principle and Method of Multidisciplinary Integrated Optimization Design (Huazhong University of Science & Technology Press, China 2006) (in Chinese)

[5] Luoyang tractor research institute: Handbook of Tractor Design: Volume One (China Machine

Press, 1994) (in Chinese)

[6] Konghui Guo: Vehicle Handling Dynamics (Jilin Science &Technology Press, 1991) (in Chinese)

[7] B. A. Stout, B. Che. ze: CIGR Handbook of agricultural engineering: Volume III (American Society of Agricultural Engineers, 1999)

[8] Jude Zhuang: Advanced Technology of Tire (Beijing Institute of Technology Press, 1996) (in Chinese)

[9] Yiming Zhou: Theory of tractor (Xi'an Jiaotong University Press, 1993) (in Chinese)

[10] Zhisheng Yu: Automobile Theory (China Machine Press, 2006) (in Chinese)

[11] S I Yi, J K Shin, G J Park. Comparison of MDO methods with mathematical examples [J]. Struct Multidisc Optim, 2008 34 391-402.

[12] D Y Gao, R W Ogden: Nonlinear Dynamics of A wheeled Vehicle (Springer, USA 2005).