ISSN 1330-3651 (Print), ISSN 1848-6339 (Online)
https://doi.org/10.17559/TV-20230111000218
Preliminary communication

Fuzzy Multi-Objectives Topology Optimization of Slider Pallet in the Picking Machine of Camellia Fruit

Chengji MI, Yang HAI, Fazhan WU, Liang LUO, Zejun CHEN, Youhua FAN, Di KANG*

Abstract. In order to improve the dynamic characteristics of the slider pallet in the camellia fruit picking machine under the traditional empirical design and to lighten the weight, a fuzzy multi-objective topology optimization design method was proposed. In this paper, a static and dynamic topology optimization mathematical model was constructed by the compromise programming method, and the weight coefficients of each sub-objective were dynamically assigned by the fuzzy satisfaction variable weight coefficient method, and then the fuzzy multi-objective topology optimization design of the slider pallet for bending condition, bending-torsional complex condition, inertia condition and the first three orders of dynamic frequency was performed. The optimization results showed that the weight of the optimized slider pallet was reduced by 19.4%, and the first-order modal frequency was increased by 5.0%, second order modal frequency increased by 6.6%, third order modal frequency increased by 8.2%; the maximum deformation and maximum stress were increased, but still met the design requirements.

Keywords: camellia fruit picking machine; dynamic characteristics; fuzzy multi-objective topology optimization; lightweight

1 INTRODUCTION

The slider pallet is the main load-bearing part of the camellia fruit picking machine, which is used to connect the slider and the tightening rod, and suffer cyclic vibration and concentrated force of each connection point. In the previous paper [1, 2], the acceleration and stress tests were conducted on the prototype of picking machine, the experimental results showed that the slider pallet had stress concentration and weight redundancy. Therefore, it is necessary to perform multi-objective topology optimization design of slider pallet to improve the structural stiffness and dynamic characteristics, increase its reliability.

Topology optimization is to find the best material distribution in a given design space, and has become one of the most widely used optimization approaches in the engineering problems. The base structure method was proposed by Dorn et al. [3], which introduced numerical methods into the field of topology optimization research. The homogenization method was proposed by Bendsoe et al. [4], which considered the size of the single cell as design variable, and the change in the size of the single cell resulted in the removal of material to form the new topology structure. The homogenization method could solve the topology optimization problems under multi-constraints, such as displacement constraint, stress constraint, frequency constraint, and so on. Based on the homogeneous method, Mlejnek et al. [5] suggested the variable density method, which regarded the density of the element as the design variable. The functional relationship between the density and the elastic modulus of element was established, and could be solved through the mathematical programming method or the criterion approach. Evolutionary structural optimization was proposed by Xie and Steven [6], whose main idea is to optimize the structure by gradually removing inefficient or ineffective materials. The level set theory was applied to topological optimization problems by Sethian to explore the topological optimization of planar structures containing strain energy constraints [7].

In addition, many advanced optimization algorithms and mathematical planning methods have been applied to the field of topology optimization. The weight coefficients in multi-objective topology optimization were determined through hierarchical analysis method to achieve static and dynamic multi-objective topology optimization of the frame [8, 9]. The second generation non-dominated sorting genetic algorithm (NSGA-II) was used for topology optimization of the bus underframe [10]. The objective function of the artificial fish swarm algorithm was determined by using a penalty function form and topology optimization of transmission tower legs was performed based on it [11].

However, in consideration of multi-objectives topology optimization, each single objective satisfaction degree can have fuzziness when solving engineering problems. In recent years, the combination of topology optimization and fuzzy theory has become a hot topic in the research field. The three commonly used mathematical models were built for multi-objectives optimization and were utilized to do topology optimization after combining with fuzzy theory [12]. An ant colony optimization algorithm based on fuzzy objective programming was proposed to conduct on the topology optimization design of distributed local area networks [13]. The hybrid robust topology optimization (RTO) method was presented to solve the uncertainty problem in the topology optimization process, and an efficient iterative method based on the perturbation theory was suggested to accelerate the convergence speed of the proposed hybrid RTO model [14]. A novel topology optimization method was proposed by Li et al. [15] to optimize the overall stiffness and volume at the same time by minimizing the product of strain energy and volume, which avoids empirical decisions on design constraints and obtains lower structural volumes, and introduces fuzzy multi-attribute group decision theory to evaluate the weights of each objective to ensure the optimal design at an acceptable level.

In recent years, the research object has been extended as the research progresses, and the study of topological optimization of nonlinear continua has received more attention and made many advances [16]. A topological optimization method for the problem of maximizing energy absorption in elasto-plastic structures was investigated by Wallin et al. [17]. An extended topological optimization method considering the loads and geometric nonlinearities at the connection joints of the assembly structure was proposed and a super cell compression method was introduced to solve the numerical instability problem [18]. A topology optimization method for maximizing the stiffness of elastic structures with frictional contacts was proposed by Niu et al. [19].

In this paper, a multi-objective topology optimization mathematical model is established based on the compromise programming method, and the sub-objective weight coefficients are assigned by using the fuzzy satisfaction variable weight coefficient method , and the static stiffness topology optimization design under different working conditions and the dynamic frequency topology optimization design of the slider pallet are conducted respectively to achieve the improvement of dynamic characteristics and weight reduction of the structure.

2 MULTI-OBJECTIVE TOPOLOGY OPTIMIZATION MATHEMATICAL MODEL

2.1 SIMP Variable Density Method

The variable density method is the most widely used and mature topology optimization method, which is derived from the homogenization method. Compared with the homogenization method, the number of design variables required is greatly reduced. The variable density method uses the cell density as the design variable, and the cell density is related to the elastic modulus of the material, and its value varies from 0 to 1. Compared with other topology optimization methods, the variable density method is simple, efficient, and has good adaptability to complex design domains [20].

There are a large number of numerical instability problems in practical continuum topology optimization, such as: checkerboard grid phenomenon, local extrema and grid dependence problems. To solve these problems, penalty factor p is introduced to make the cell density approach 0 or 1 quickly, and finally obtain the optimal material distribution of the structure. The relationship between the elastic modulus and the cell density is [21].

$$\begin{aligned} & E_{ijkl}(x) = \rho(x)^p E_{ijkl}^0 \\ & \text{s.t.} \quad \int_\Omega \rho(x) \mathrm{d}\Omega \le V \end{aligned} \tag{1}$$

where, E is the elastic modulus, p is the penalty factor, $p > 1$; V is the allowable amount of material; the density function of the material $0 \le \rho \le 1$; and the design variables $x \in \Omega$.

The variable density method has been integrated in many commercial software, including HyperWorks, which is used in this paper.

2.2 Static Stiffness Optimization Mathematical Model

The strain energy can be considered as the reciprocal of the stiffness, and in the optimization process, is generally used to define the stiffness of the structure. Therefore, the static stiffness optimization mathematical model can be expressed as the following [22]:

$$\begin{cases} \text{Find } X = [x_1, x_2, \dots, x_n]^T \\ \min \ C(X) = U^T F / 2 = U^T K U / 2 \\ \text{s.t. } V(X) = \sum_{i=1}^{n} x_i v_i \le V^*; F = KU \\ \quad 0 < x_{\min} \le 1, i = 1, 2, \dots, n \end{cases} \tag{2}$$

where $C(X)$ is the structural strain energy; U is the displacement vector of the structure; K is the total stiffness matrix of the structure; F is the load vector; v_i is the volume of the structural unit; V^* is the upper volume limit; $x_{\min}$ is the minimum relative density.

2.3 Dynamic Characteristic Optimization Mathematical Model

In dynamic vibration frequency optimization process, it is generally expected to improve the low order frequency of the structure to avoid vibration in resonance. In the optimization process, this situation may be encountered: when the frequency of some order is raised to the maximum, the frequencies of other orders may be reduced, resulting in the confusion of the order between frequencies. This phenomenon can be avoided by using the average frequency method, so the average frequency formula of the first few orders frequency is used as the objective function of frequency optimization in this paper. The average frequency formula is [23].

$$\Lambda(X) = \lambda_0 + s\left(\sum_{j=1}^{f} \frac{\omega_j}{\lambda_j - \lambda_0}\right)^{-1} \quad j = 1, 2, \dots, f \tag{3}$$

where $\Lambda(X)$ is the average eigenvalue frequency; λ_0, s is the given parameter to adjust the objective function; λ_j is the j-th order eigenvalue; f is the frequency order to be optimized; ω_j is the weight coefficient of each lower order frequency.

The mathematical model of dynamic characteristic optimization is calculated as follows [22]:

$$\begin{cases} \text{Find } X = [x_1, x_2, \dots, x_n]^T \\ \max \Lambda(X) = \lambda_0 + s\left(\sum_{j=1}^{f} \frac{\omega_j}{\lambda_j - \lambda_0}\right)^{-1} \quad j = 1, 2, \dots, f \\ \text{s.t. } \ V(X) = \sum_{i=1}^{n} x_i v_i \le V^* \\ \quad \left[K - \lambda_j M\right] \frac{1}{n} \Phi_j = 0 \quad j = 1, 2, \dots, d_f \\ \quad 0 < x_{\min} \le x_i \le 1, \qquad i = 1, 2, \dots, n \end{cases} \tag{4}$$

where d_f is the total number of freedom degrees of the finite element model; K is the structural stiffness matrix; M is the structural mass matrix; Φ_j is the eigenvector.

2.4 Multi-Objective Topology Optimization Mathematical Model

The popular multi-objective optimization methods include linear weighting method, square sum weighting method, and compromise programming method. In this paper, the compromise programming method is used to describe the mathematical model as follows [23]:

$$\begin{cases} \text{Minimize:} F_3(\rho) = \sqrt{\begin{array}{l} \omega^2 \left[\sum_{i=1}^{\lambda} \omega_i \left(\dfrac{C_i(\rho) - C_{i\min}}{C_{i\max} - C_{i\min}} \right) \right]^2 + \\ (1-\omega)^2 \left[\left(\dfrac{\Lambda(\rho) - f_{\max}}{f_{\min} - f_{\max}} \right) \right]^2 \end{array}} \\ \text{s.t.} \begin{cases} V(\rho_i) - \bar{V} \le 0, & i = 1, 2, \ldots, n \\ 0 < \rho_{\min} \le \rho_i < 1, & i = 1, 2, \ldots, n \end{cases} \end{cases} \tag{5}$$

where $C_{i\min}$ and $C_{i\max}$ are the minimum and maximum strain energy of the single-objective optimization for the i-th working condition, respectively. $\Lambda(\rho)$ is the average frequency, while $f_{\min}$ and $f_{\max}$ are the minimum and maximum frequency of the average frequency optimization.

3 FUZZY SATISFACTION VARIABLE WEIGHTING COEFFICIENT METHOD

In the process of topology optimization, there is no clear criteria to determine the advantages and disadvantages of optimized results. Therefore, the result of iterative optimization is fuzzy properties, especially for the multi-objective topology optimization. When each sub-objective cannot reach the optimal value at the same time, it is better to make the value of each sub-objective fall within the fuzzy satisfaction set as much as possible.

3.1 Introduction of the Membership Function

The most important step of satisfaction theory is to construct the membership function associated with the objective function. Then, the specific objective function, could be fuzzified, and the process of finding optimal solution could be transformed into searching the best satisfactory solution. In this paper, the functional relationship between satisfaction q_i and iteration value $C_i(\rho)$ was built by an exponential-shaped membership function. The mathematical expression is listed as follows [12]:

$$q_i = \frac{1}{1 + \exp\left(a_1 - a_2 \dfrac{C_i - C_{i\max}}{C_{il} - C_{i\max}} \right)} \tag{6}$$

where a_1 and a_2 are set parameters, the purpose is to adjust the variation rate of satisfaction q_i so that it changes faster when the strain energy is taken near the worst value and slows down when the strain energy is taken near the optimal value, which are taken as 2 and 4 respectively. And $C_{i\max}$ is the worst value of the i-th sub-objective, which corresponds to the maximum value of the strain energy of the optimized single objective. C_{il} is the best value of the i-th sub-objective, which similiarly corresponds to the minimum value of the strain energy.

3.2 Dynamic Assignment of Weighting Factors

The weight coefficient ω_i indicated the importance of each condition in the multi-objective optimization, while the fuzzy satisfaction method is to combine the weight coefficient with the membership function, which resulted in the weight coefficient dynamically changing with the number of iteration step. The mathematical model of the weight coefficient was denoted as follows [12]:

$$\omega_i = \frac{1 - q_i}{\sum_{i=1}^{3} (1 - q_i)} \tag{7}$$

where ω_i is the weight coefficient of each working condition, and q_i is the satisfaction level of each working condition.

4 FUZZY MULTI-OBJECTIVE TOPOLOGY OPTIMIZATION OF SLIDER PALLET

Finite element analysis is an analytical method that uses mathematical approximation to simulate a real physical system by decomposing the solution domain into many small interconnected sub-domains, and linking the simple equations on these sub-domains to find the complex equations of the total solution domain. In this paper, HyperWorks is used to build a finite element model of the slider pallet of the oil tea fruit picker, and then finite element analysis and topology optimization are performed based on it.

The slider pallet was the main load-bearing part of the camellia fruit picking machine, its material was 45 steel, and the radius of external tangent circle was about 182 mm and the thickness of slider pallet was 5 mm. the geometric structure of the slider pallet was shown in Fig. 1. The strain rosette was pasted on the top surface of slider pallet, shown in Fig. 2. The experimental results showed that the maximum stress at this position reached 541 MPa. According to the initial analysis of tested and calculated results, there was not stress concentration point, so this high stress was probably induced by the vibraition in resonance [1].

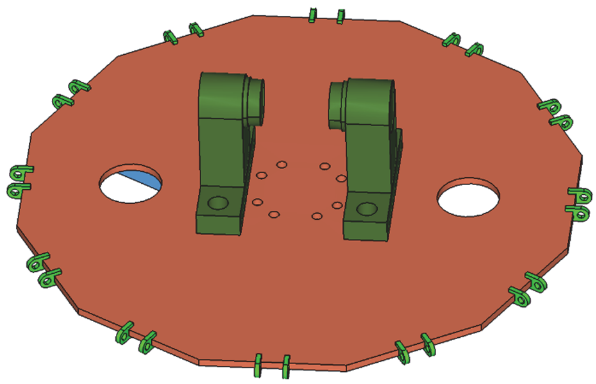

Figure 1 Geometry structure of slider pallet

Figure 2 Schematic diagram of strain rosette pasting on slider pallet

4.1 Finite Element Model

In this paper, pentahedral and hexahedral solid element were used for meshing the slider pallet. The number of nodes in the established finite element model was 27000, and the number of elements was 18514. The Young modulus was 210 GPa, and the Poisson's ratio was 0.3, and the density of material was 7850 kg/m^3.

The slider pallet was subjected to the following working conditions. The bending condition suffered the concentrated force 150 N in the vertical direction located at the twevel connection points between the slider pallet and the tightening rod, while all constrains were applied on the surface of the connection holes between the slider pallet and the left and right supports. In the bending-torsion case, the slider pallet suffered the same concentrated force and the tangential force 100 N appeared at the same position. The constraint condition was the same as the bending condition. In the inertia condition, the centripetal force 300 N was applied at the connection point between the slider pallet and the tightening rod, and the constraint condition was still the same as the first two cases. Since the slider pallet was moved under a kind of high-speed vibration state, the gravitational acceleration 9.43 g was applied in the model referred to the tested data [1]. The final finite element model was shown in Fig. 3.

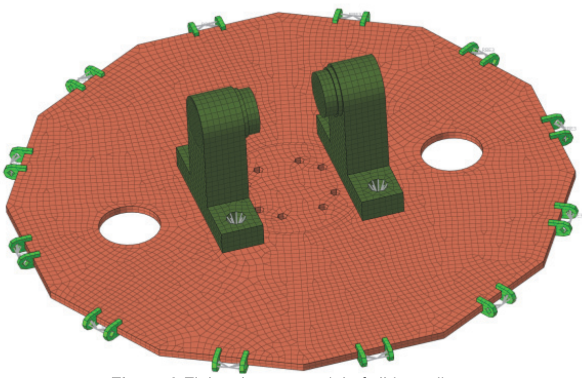

Figure 3 Finite element model of slider pallet

4.2 Single-Objective Topology Optimization

The purpose of the optimization in this paper is to increase the intrinsic frequency of the slider pallet and reduce its weight as much as possible while ensuring the strength and stiffness of the slider pallet. Before starting the topology optimization, the design variable, constraint function and objective function needed to be defined firstly. In this paper, the variable density method based on the SIMP interpolation model was used to find the optimal solution, the density of the element was considered as the design variable. The constraint function was selected as the volume percentage of the slider pallet, and its upper limit was assigned to be 0.4, which meant that the sixty percentage materials could be removed during the optimization process. The objective function was chosen as the maximum stiffness under each working condition and the maximum average frequency of the first three order modes, respectively. The demoulding was treated as the unidirectional type, and the direction was along the negative direction of Z-axis. Since the slider pallet was the cyclic symmetric structure, the constraint of the mode group was considered as the circumferential cyclic symmetry type, and the number of sector zones was 12.

Based on the HyperWorks software platform, the topology optimization design was implemented. Through multiple iterations, the topology optimization results of each single objective were obtained, and shown in Figs. 4 to 9, respectively. The simulated results showed that the topology structure of the optimized slider pallet in the second condition was similar with the first one. While the region near the two supports needed a lot of materials, the area around the middle line and two through holes only retained few materials. For the third inertia condition, compared with the resulats mentioned above, the biggest difference was that more materials were kept between the twevel connection points and two supports.

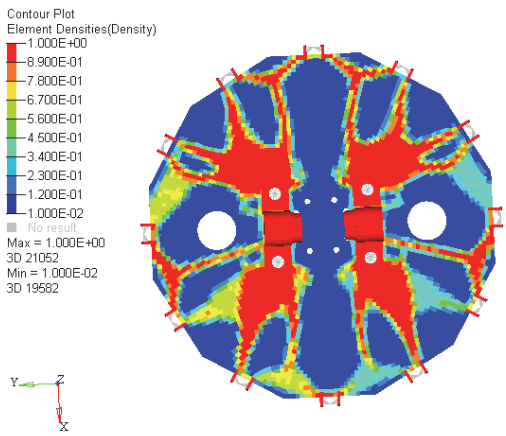

Figure 4 Topology optimization result of bending condition

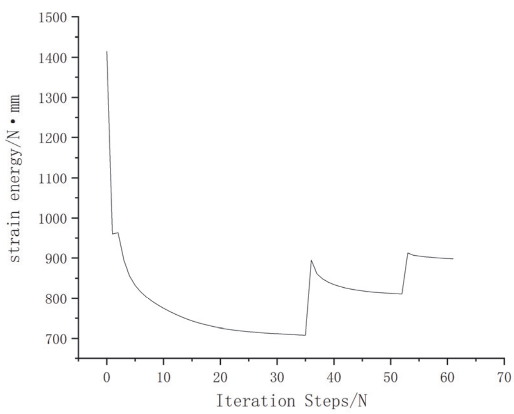

Figure 5 Optimization iteration process of bending condition

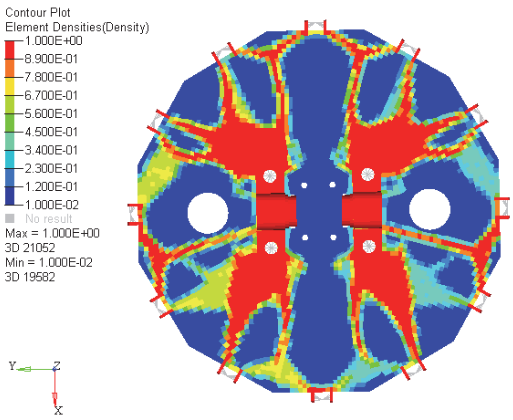

Figure 6 Topology optimization result of bending and torsion condition

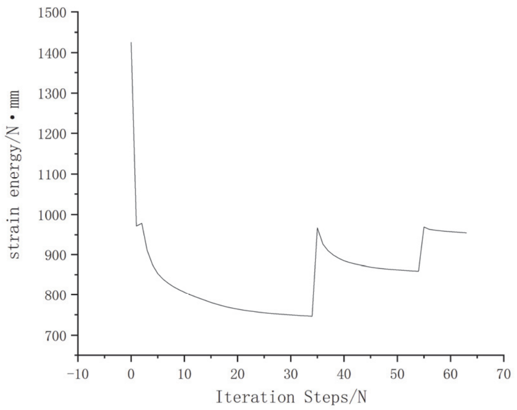

Figure 7 Optimization iteration process of bending and torsion condition

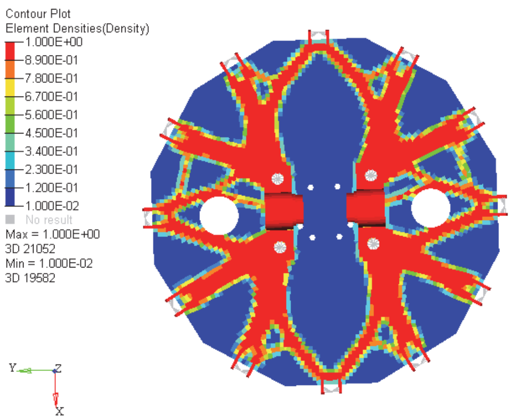

Figure 8 Topology optimization result of inertia condition

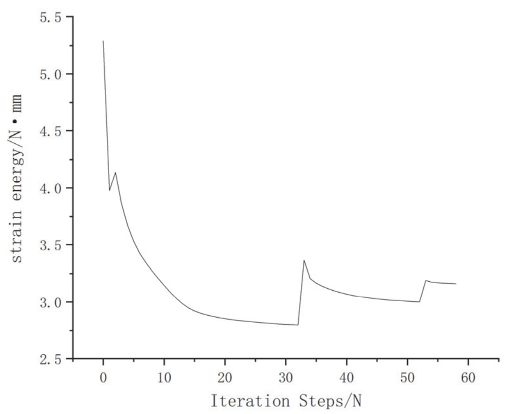

Figure 9 Optimization iteration process of inertia condition

The comparison of the structural strain energy of the initial and optimized slider pallet under different working conditions was shown in Tab. 1. From the calculated results, the optimized topology structure appeared in radial pattern. The structural strain energy in all conditions was reduced, which meant that the stiffness was enlarged. The optimized results were acceptable.

Table 1 Comparison of structural strain energy

	First case	Second case	Third case
Initial	1415 Nmm	1425.7 Nmm	5.29 Nmm
Optimized	897.793 Nmm	954.753 Nmm	3.16 Nmm
Reduction	517.207 Nmm	470.947 Nmm	2.13 Nmm

The modal frequency results of the initial and optimized slider pallet were listed in Tab. 2. Substituting the maximized frequency of the first three order modes into the Eq. (3), the quantification function of maximizing average frequency could be obtained when the parameter λ_0 and s was considered 0 and 1, respectively. The optimized results of dynamic charateristics were shown in Fig. 10 and Fig. 11. The final minimum and maximum average frequencies were 232.443 Hz and 476.359 Hz, respectively.

Table 2 Comparison of the first three order modes

	1st order mode	2nd order mode	3rd order mode
Initial	222.559 Hz	226.413 Hz	254.269 Hz
Optimized	415.176 Hz	538.84 Hz	518.124 Hz
Increase	192.617 Hz	312.427 Hz	263.855 Hz

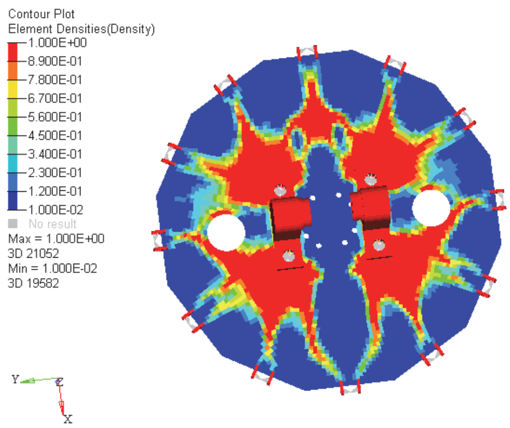

Figure 10 Topology optimization result of dynamic characteristics

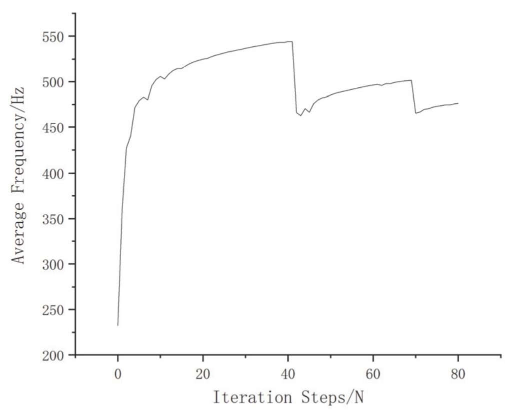

Figure 11 Optimization iteration process of dynamic characteristics

4.3 Fuzzy Multi-Objective Topology Optimization

The data of each single-objective optimization in Tabs. 1 and 2 are substituted into Eq. (5) to complete the multi-objective optimization function, and the weight of the single-objective optimization in the multi-objective optimization is dynamically processed by the variable weight coefficient method. The space and parameters of slider pallet for the multi-objectives topology optimization were consistent with the single-objective topology optimization. The multi-objectives optimization function established by the compromise programming method were written into the dequation module to find the corresponding relationship between the unknown variables and their responses. The final multi-objective topology optimization results of the slider pallet are shown in Fig. 12 and Fig. 13.

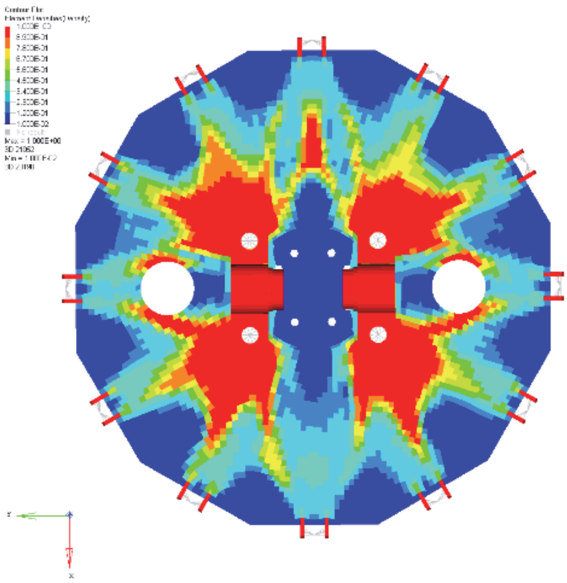

Figure 12 Multi-objective topology optimization results of slider pallet

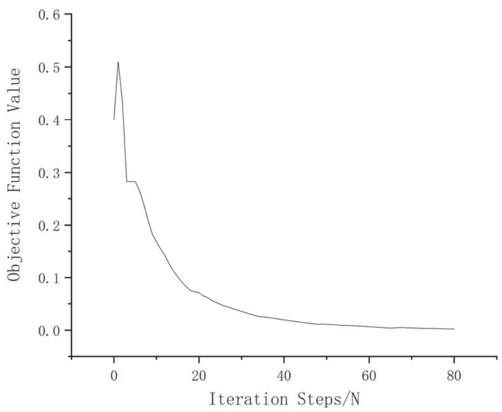

Figure 13 Multi-objective topology optimization iteration process of slider pallet

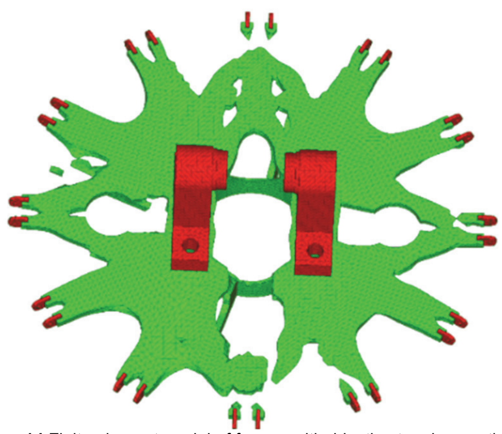

Figure 14 Finite element model of fuzzy multi-objective topology optimized slider pallet

The optimized finite element model of slider pallet was displayed by the OSSmooth module, as shown in Fig. 14. Based on the optimization results, the geometrical structure of the slider pallet was redesigned. U-shaped and V-shaped notches are created on the edges of the slider pallet and round holes are dug in the corresponding positions of the optimized structure, and rounded corners at locations prone to stress concentration in the new structure. The final rebuilt geometric model of the slider pallet was shown in Fig. 15.

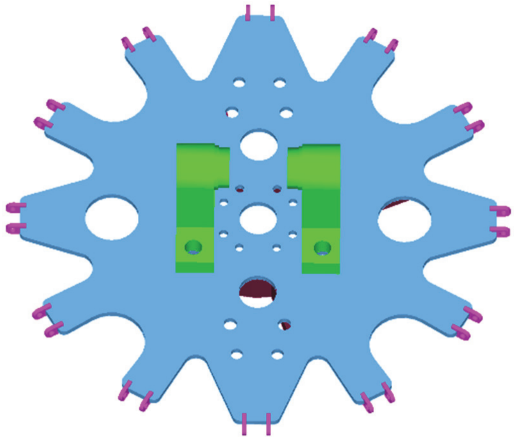

Figure 15 Geometry model of rebuilt slider pallet

The performance comparison of the initial and optimized slider pallet was shown in Tab. 3. According to the simulated results, the maximum displacement and the maximum VonMises stress of the initial structure occured in the bending condition, while the maximum displacement of the optimized slider pallet appeared in the bending condition and the maximum VonMises stress occured in the bending-torsion condition. The frequency of the first three order modes in the optimized model was all higher than the initial model, among which the frequency of the first order mode was increased by 5.0%. That value may effectively avoid the vibration in resonance. Moreover, the mass of the optimized slider pallet was reduced by 19.4%, compared with the inital model.

Table 3 Performance comparison of initial and optimized slider pallet

	Initial	Optimized
Max displacement	0.3917 mm	0.787 mm
Max VonMises stress	48.75 MPa	87.60 MPa
1st order mode	340.951 Hz	358.398 Hz
2nd order mode	345.418 Hz	368.376 Hz
3rd order mode	389.445 Hz	421.496 Hz
mass	6.04Kg	4.868Kg

As can be seen from Tab. 3, although the maximum displacement and maximum VonMises stress of the optimized model was a little higher than the initial one, they still meet the allowable requirements of stiffness and strength.

5 CONCLUSION

In this paper, the fuzzy multi-objectives topology optimization was presented by combining fuzzy theory. The static and dynamic single-objective topology optimization mathematical models and multi-objectives topology optimization mathematical model were firstly established based on SIMP variable density method, respectively. Then, the single-objective topology optimization of slider pallet with the objective of maximizing stiffness in static conditions and the objective of maximizing the first three orders modes was conducted. The fuzzy satisfaction variable weight coefficient method is proposed by combining the weight factors with the membership function to dynamically assign the weight factors to each sub-objective. Based on the compromise programming method, the fuzzy multi-objective topology optimization results of the slider pallet were obtained. Based on the optimization results to rebuild the slider pallet structure, and the new structure compared with the original structure, the results show that the weight of the optimized slider pallet was reduced by 19.4%, and the frequency of the first order mode was increased by 5.0%. The structural dynamic characteristics were improved with less weight. Even though the maximum displacement and VonMises stress had a slight increase, they still met the requirements.

Nomenclature

U - Displacement vector of the structure
K - Total stiffness matrix of the structure
F - Load vector
v_i - Volume of the structural unit
V^* - Upper volume limit
x - Relative density
λ - Eigenvalue
f - Order number of frequency to be optimized
d_f - Total number of freedom degrees
ω_j - Weight coefficient of each lower order frequency
M - Structural mass matrix
Φ_j - Eigenvector
q_i - Satisfaction level of each working condition

Acknowledgements

The authors gratefully thank for the support of Science and Technology Innovation Project of Hunan Province (Grant No.: 2018NK1030 and 2018NK2065) and Forestry Science and Technology Promotion Demonstration Project of Central Finance (Grant No.: 2018XT02) and Forestry Science and Technology Innovation Project of Hunan Province (Grant No.: XLK201910).

6 REFERENCES

[1] Lin, Y. P., Mi, C. J., Zhao, J. Y., Tang, Y. H., Li, M. H., Kang, D., Chen, Z. J., & Fan, Y. H. (2020). Development and dynamic characteristics of a new type of camellia fruit picking machine. *Journal of Northwest Forestry University*, *35*(05), 267-272.
[2] Kang, D., Chen, Z. J., Fan, Y. H., Li, C., Mi, C. J., & Tang, Y. H. (2021). Optimization on kinematic characteristics and lightweight of a camellia fruit picking machine based on the Kriging surrogate model. *Mechanics & Industry*. https://doi.org/10.1051/MECA/2021017
[3] Dorn, W. S., Gomory, R. E., & Greenberg, H. J. (1964). Automatic design of optimal structures. *Journal de Mécanique*, *3*(1), 25-52.
[4] Bendsøe, M. P. & Kikuchi, N. (1988).Generating optimal topologies in structural design using a homogenization method. *Computer Methods in Applied Mechanics and Engineering*, (2). https://doi.org/10.1016/0045-7825(88)90086-2
[5] Mlejnek H. P. & Schirrmacher, R. (1993). An engineer's approach to optimal material distribution and shape finding. *Computer Methods in Applied Mechanics and Engineering*, (1-2). https://doi.org/10.1016/0045-7825(93)90182-W
[6] Xie, Y. M. & Steven, G. P. (1993). A simple evolutionary procedure for structural optimization. *Computers & Structures*, (5). https://doi.org/1010.1016/0045-7949(93)90035-C
[7] Sethian, J. A. & Wiegmann, A. (2000). Structural Boundary Design via Level Set and Immersed Interface Methods. *Journal of Computational Physics*, (2). https://doi.org/10.1006/jcph.2000.6581
[8] Jiang, J. X. (2013). *Optimization of frame structure of SF35100 mining dump truck.* Hunan University.
[9] Jiang, J. X., Gu, Z. Q., Mi, C. J., Wang, Y. T., & Zang, X. L. (2013). Multi-objective topology optimization study of mining dump truck frame structure. *China Mechanical Engineering*, (08), 1028-1032.
[10] Gao, Y. K. & Duan, S. D. (2017). Discrete topology optimization of bus underframe based on NSGA-II algorithm. *Journal of Tongji University (Natural Science Edition)*, (11), 1664-1669.
[11] Guo, H. Y., Song, X. H., & Li, Z. L. (2017). Topology optimization of transmission tower legs based on improved artificial fish swarm algorithm. *Vibration and Shock*, (04), 52-58+87. https://doi.org/10.13465/j.cnki.jvs.2017.04.009
[12] Wang, H. G. (2016). *Static and dynamic characteristics analysis and optimization design of heavy truck frame.* Chongqing Jiaotong University.
[13] Salman, A. K. & Amjad M. (2019). Fuzzy goal programming-based ant colony optimization algorithm for multi-objective topology design of distributed local area networks. *Neural Computing and Applications*, (7). https://doi.org/10.1007/s00521-017-3191-5
[14] Meng, Z., Wu, Y., Wang, X., Ren, S., & Yu, B. (2021). Robust topology optimization methodology for continuum structures under probabilistic and fuzzy uncertainties. *International Journal for Numerical Methods in Engineering*, (8). https://doi.org/10.1002/NME.6616
[15] Hao, L., Liang, G., & Peigen, L. (2014). Topology optimization of structures under multiple loading cases with a new compliance - volume product. *Engineering Optimization*, (6). https://doi.org/10.1080/0305215X.2013.800054
[16] Wen, G. L., Liu, J., Chen, Z. J., Wei, P., Long, K., Wang, H. X., & Xie, Y. M. (2022). A review of topology optimization methods for nonlinear continua. *Journal of Mechanics*, (10), 2659-2675.
[17] Wallin, M., Jönsson, V., & Wingren, E. (2016). Topology optimization based on finite strain plasticity. *Structural and Multidisciplinary Optimization*, (4). https://doi.org/10.1007/s00158-016-1435-0
[18] Jie, H., Xiaojun, G., Jihong, Z., & Weihong, Z. (2020). Topology optimization of joint load control with geometrical nonlinearity. *Chinese Journal of Aeronautics*, (1). https://doi.org/10.1016/j.cja.2019.01.024
[19] Cao, N., Weihong, Z., & Tong, G. (2020). Topology optimization of elastic contact problems with friction using efficient adjoint sensitivity analysis with load increment reduction. *Computers and Structures.* https://doi.org/10.1016/j.compstruc.2020.106296
[20] Xiao, X. W. (2021). *Research on multi-objective optimization design of electric wheel dump truck frame based on interval method.* Hunan University of Technology.
[21] Mi, C. J. (2014). *Research on the fatigue reliability of mining dump truck frame based on strain energy method.* Hunan University.
[22] Fang, Z. L. Wang, L. J., Chen, Z. H., & Wang, Y. S. (2013). Satisfactory coordination method in multi-objective

topology optimization in OptiStruct. *Mechanical Science and Technology for Aerospace Engineering*, (08), 1190-1194. https://doi.org/10.13433/j.cnki.1003-8728.2013.08.028
[23] Fan, W. J., Fan, Z. J., & Su, R. Y. (2008). Research on multi-objective topology optimization method for automobile frame structure. *China Mechanical Engineering*, (12), 1505-1508.

Contact information

Chengji MI
Department of Mechanical Engineering,
Hunan University of Technology,
No. 88 West of Taishan Road, Zhuzhou City, China
E-mail: michengji@hut.edu.cn

Yang HAI
Department of Mechanical Engineering,
Hunan University of Technology,
No. 88 West of Taishan Road, Zhuzhou City, China
E-mail: happy2021hut@126.com

Fazhan WU
Zhuzhou Fengke Forestry Equipment Co., Ltd,
Hunan Zhuzhou 412000, China

Liang LUO
Department of Mechanical Engineering,
Hunan University of Technology,
No. 88 West of Taishan Road, Zhuzhou City, China

Zejun CHEN
Hunan Academy of Forestry Sciences,
Hunan Changsha 410004, China

Youhua FAN
Hunan Academy of Forestry Sciences,
Hunan Changsha 410004, China

Di KANG
(Corresponding Author)
Hunan Academy of Forestry Sciences,
No. 658 South of Shaoshan Road, Changsha City, China
E-mail: 13838@hut.edu.cn