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Dynamic Correction Analysis and Optimization Model of Gear Pair Based on the Normalization of Gear Modification Parameters

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Abstract: In this paper, a method of condition recognition under multi-gear excitation and a method of structural dynamics model modification based on random combination stiffness of large heavy transmission equipment are studied and proposed, and the modified structural dynamics model is applied to the dynamic reliability analysis of gearbox. Firstly, a variable working condition correction method based on working condition characteristics is proposed. The working condition and other commonly used eigenvalue parameters are normalized to the eigenvalue for training. The relationship model between working condition, characteristic value and correction value is established to identify correction value. The effectiveness of the method is verified by the vibration signal of the gearbox. Secondly, the comprehensive meshing stiffness of gear pair is normalized by Fourier series, and the dynamic correction model of gear pair is established. Finally, the method of structural dynamic model modification is applied to gearbox dynamic reliability analysis, and a research method of gearbox dynamic reliability based on structural dynamic model modification is proposed. An improved dynamic model of gearbox structure is used to predict the vibration response of the actual unmeasured points. The dynamic reliability sensitivity analysis and the dynamic reliability analysis based on the first overtaking failure mechanism of the dangerous parts of the transmission are carried out. The three factors which have the greatest influence on the dynamic reliability of the gearbox are calculated.

Keywords: gear shaping; normalization; power reliability; revise the model; side dynamics

1 INTRODUCTION

At present, most of the gear correction for variable working conditions is aimed at vibration signals with continuous variable speeds. Order analysis and time-frequency analysis methods are used to determine whether the equipment has undergone correction. Due to the lack of integration with machine learning, intelligent diagnosis and judgment of the severity of correction cannot be achieved. To achieve intelligent correction diagnosis, it is necessary to use feature extraction and pattern recognition. For continuous variable speed signals, due to the uncertainty of speed, it is impossible to effectively identify corrections by directly extracting feature values and inputting them into the pattern recognition model. In practical work, collecting a small segment of mechanical equipment vibration signals working under variable speed conditions, whose working conditions generally vary very little, can be regarded as signals under constant working conditions. Therefore, this article combines multiple vibration signals under constant operating conditions to achieve gearbox correction diagnosis under variable operating conditions. At the same time, the comprehensive meshing stiffness of gear teeth also changes periodically with time. In the dynamic analysis model of the system, it reflects the periodic time-varying parameters of elastic force, and the time-varying meshing stiffness is the core factor affecting the dynamic characteristics of the gear transmission system [2-3]. There are currently three main methods for obtaining the time-varying mesh stiffness of gears: finite element method, semi analytical method, and simplified square wave method. In order to solve the problem of matrix based correction methods lacking physical meaning, parameter based correction methods have been proposed as a correction method that has clear physical meaning and is convenient for engineering applications [4]. The basic idea of parametric correction method is similar to structural optimization theory, which constructs the error of dynamic characteristics between the theoretical model and the actual model under the same conditions, and then selects specific correction parameters for correction [5] to minimize the error and achieve the goal of correction. The parameter based correction method mainly includes sensitivity based optimization algorithms, statistical inference based correction algorithms, as well as computer intelligence based optimization algorithms such as neural network methods, genetic algorithms, particle swarm optimization, simulated annealing, etc. [6]. These methods mainly focus on analyzing the results obtained from software simulation, without establishing an efficient and rigorous dynamic theoretical model [7]. The analysis process and results are discrete, and therefore cannot accurately reflect the dynamic characteristics of gear meshing at any time. In response to the difference between the single tooth meshing section and the double tooth meshing section in gear transmission, it is proposed to normalize the meshing stiffness of the gear, so that the corresponding dynamic model can achieve continuity in time series, and thus achieve the goal of analyzing the dynamic characteristics at any meshing moment. For large heavy-duty transmission equipment, there is a lot of uncertainty in its modeling, and it is crucial to determine the most reasonable correction parameters for sensitive parameters from the numerous modeling parameters. Therefore, how to reasonably select the correction parameters and accurately correct the solution is a key issue in model correction. Introduce the theory of stochastic dynamics into the sensitivity analysis model correction method, study the modal frequency dynamic model correction method based on stochastic combined stiffness, and conduct practical verification. The application research of modifying the structural dynamic model in the dynamic reliability analysis of gearboxes, establishing a gear dynamic reliability model based on stress intensity interference theory, using the modified gear structural dynamic model to predict the vibration response of actual unmeasurable points, conducting dynamic reliability sensitivity analysis and reliability analysis based on the first overtaking failure mechanism for dangerous parts of gears, and studying their variation patterns.

2 RELATED WORK

Traditional spectral analysis methods are not suitable for gearbox vibration signals at variable speeds, and for angle domain stationary signals, traditional methods can play their role, resulting in methods such as order spectrum, envelope order spectrum, and inverse order spectrum. By combining adaptive sparse time-frequency analysis with order analysis [8], the corrected order information of the gearbox variable speed signal was successfully obtained; By using generalized demodulation time-frequency analysis to decompose multi-component signals into single component signals [9], the envelope order spectrum of the single component signal is calculated to achieve corrected diagnosis of variable speed gearboxes. The combination of Hilbert demodulation and inverse order spectrum analysis for angle domain stationary signals can effectively remove interference components in the order spectrum [10], and has been successfully applied to the analysis of gearbox variable speed correction signals. By decomposing the variable speed signal through LMD (Local Mean Decomposition), the envelope order spectrum of the modified PF component in the decomposition result is generated, and the amplitude of the modified order is input as the characteristic value into the variable prediction model to achieve the recognition of variable speed rolling bearing correction [11]. Perform singular value decomposition on the PF (Physical Function) component obtained from LMD decomposition, and input the obtained singular values as eigenvalues into the limit learning machine to achieve variable speed rolling bearing correction recognition [12]. By decomposing the equal angle sampled signal through continuous wavelet transform, the wavelet coefficients of the decomposition results are extracted as feature values and input into the neural network to achieve the diagnosis of wear correction for variable speed gears. The vibration and impact of gears can cause fatigue fracture, and gear shaping is the most important means to reduce vibration and improve the reliability of high-speed and heavy-duty gear transmission. Gear shaping technology is a key technology for high-precision gear transmission design and manufacturing, and gear companies treat it as a core technology. Gear modification mainly includes tooth profile modification and tooth direction modification. During the meshing process of a pair of teeth, the change in the number of teeth involved in meshing causes a change in meshing stiffness, and a sharp change in meshing stiffness in a very short period of time will cause serious vibration. In order to make the changes in meshing stiffness more gentle and reduce the meshing and meshing impacts caused by base joint errors and load deformation; Alternatively, in order to improve the lubrication of the tooth surface and prevent adhesion, a portion of the original involute tooth profile may be removed at the tooth tip or near the root fillet. This is the so-called tooth profile modification [13]. Gear modification technology is listed as a Class I tackling problem among the 16 key technologies of gears [14]. The discussion on the modification of involute gear tooth profile proposes a method of keeping the single tooth meshing section unchanged [15], and using the length of the double tooth meshing section on the meshing line as the tooth top modification height of small and large gears. Later gear modification methods such as Sigg tooth profile modification [16] are usually performed on small gears, while matching large gears are not modified. This method requires leaving a length equal to the base tooth pitch on the meshing line without any adjustment. Compared to the Walker method, its trimming length is shorter. In order to reduce the dynamic load coefficient of the transmission system and improve the bearing capacity of the gear pair, people consciously use curves such as parabolas, arcs, and arc envelopes to modify the tooth profile of the gear, so that the gear can have a good meshing state in the actual working process. Combining theory and experiment, the drum shape of involute gear modification was studied, and the calculation formula was derived [17]. A study was conducted on the tooth profile modification height, modification area, and modification curve of spur and helical gears [18], and an estimation formula was provided. The probability method and function approximation method were used to quantify the gear tooth error, and the functional relationship between load distribution rate, driven wheel lag angle, gear tooth error and gear modification parameters was established [19]. An experimental study was conducted on the thermal deformation tooth profile modification of high-speed gears [20], and a calculation method for determining the amount of thermal deformation tooth profile modification of high-speed gears was provided. The finite element method was used to analyze and calculate the deformation of the gear shaft and the volume temperature of the gear, and a "target modification" design method was proposed to determine the modification amount and modification curve specifically and reasonably for different objects [21], requirements, and purposes. As the analysis of gear meshing dynamic load develops from impact theory to vibration theory, the calculation of gear modification amount is also related to how to reduce the fluctuation of vibration and gear transmission error. The optimization objective is to minimize the root mean square value of gear vibration acceleration [22], and a gear dynamic performance optimization design method is proposed. This method is used to obtain the optimal tooth profile modification amount and modification length for dynamic performance. Tooth profile modification as an effective means to reduce tooth meshing excitation [23] is used to eliminate the impact of tooth meshing and tooth meshing, and to minimize the fluctuation of gear transmission errors. Subsequently, research on gear modification based on vibration theory rapidly developed. In the study of vibration and noise in automotive transmissions, it was found that when the modification amount is equal to the deformation of the teeth at the upper boundary point of single tooth meshing [24], the dynamic load coefficient and root mean square value of acceleration of the gear pair are both small. The root mean square value of the noise in cylindrical gear transmission was used as the optimization objective function, and the optimal modification curve that minimizes the noise was obtained. Using the root mean square value of noise as the optimization objective function [25], the differences between straight line and curve modifications were studied, and the conclusion was drawn that the curve modification effect is better under high-speed and heavy loads. The three maintenance profiles of gear teeth are carried out by the method of tooth elevation modification, drum modification, spiral modification and tooth end modification [26]. The generation of involute gear profile modification was studied using a hypothetical rack and pinion rigid tool. Based on the relationship between gear transmission load and tooth deformation, the expression of the relationship between the transmission error of modified spur gears and the comprehensive modification parameters of tooth pairs under fixed load conditions is derived [27]. The above research involves the determination of the modification curve, the size of the modification amount, and the selection of the modification location. However, due to the imprecision of the calculation model for gear mesh stiffness and elastic deformation, and the modification parameters are generally accurate to micrometers, the current method of determining the modification parameters cannot accurately guide the design of gear modification. The methods for calculating tooth profile modification can be roughly classified into three categories based on their theories: material mechanics methods, elastic mechanics methods, and numerical solutions. The results obtained by the three types of calculation methods are not significantly different, each with its own advantages and disadvantages. Depending on the specific situation, different methods can be adopted. The key is to grasp the fundamental purpose of shape modification and apply it flexibly in practice. Although some progress has been made in the theoretical research of gear modification, the complex calculation process and large workload have brought difficulties to its application in practical production. At present, most gear design adopts empirical formulas for gear modification, which is difficult to meet the requirements of heavy-duty, high-speed, and high-precision mechanical transmission. With the continuous development of computer technology, the calculation speed of large-scale finite element software is already acceptable. Therefore, optimization algorithms can be combined with 2D or 3D gear finite element models to determine the dynamic optimal parameters for profile modification. We have conducted in-depth research on this [28] and obtained good theoretical research results, opening up new ideas and methods for the theoretical research of gear modification.

3 RESEARCH ON NORMALIZATION CORRECTION ANALYSIS AND OPTIMIZATION OF GEAR MODIFICATION PARAMETERS

3.1 A Normalized Extraction Framework for Quantitative Correction Parameters of Variable Condition Gears Based on Working Condition Characteristics

By treating the operating conditions as eigenvalues and inputting them together with other eigenvalues into the neural network model for training, a model was established between the operating conditions, eigenvalues, and corrections, effectively improving the predictive performance of the model. The effectiveness of this method was verified through gearbox vibration signals. Applying transfer learning to corrective diagnosis solves the problem of not being able to train machine learning models with good performance in variable condition corrective diagnosis due to the small number of target condition samples. The framework diagram is shown in Fig. 1.

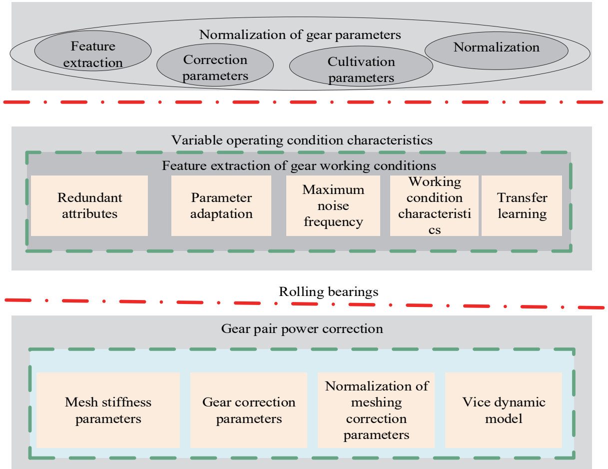

Figure 1 Normalized extraction framework for gear correction parameters

Gear shaping refers to fine-tuning the shape of a gear through a certain smooth curve, in order to change the meshing state of the gear, eliminate geometric interference between teeth, and improve the load impact and vibration noise generated by gear operation. Gear modification can be divided into tooth alignment modification and tooth profile modification. Among them, tooth profile modification refers to optimizing the top and root of the gear to reduce the deviation between the actual and theoretical modification of the gear base section.

3.2 Normalization of Time-varying Comprehensive Meshing Stiffness

The symbols in the text are shown in Tab. 1.

Table 1 Symbols

Symbols	Implication
k_{mi}	the meshing stiffness
k_1, k_2	the meshing stiffness of the driving and driven gears
r	damping coefficient
$y(t)$	projection matrix
R_p, R_g	base circle radii of the active and driven gears
x	relative displacement
w	corresponding natural frequency

When a pair of gears mesh, due to the intervention of external loads, the teeth of both the active and driven gears will undergo elastic deformation. If k_1 and k_2 are the meshing stiffness of the driving and driven gears in the normal direction at the meshing point, then the meshing stiffness k_i of a pair of teeth can be regarded as the series result of k_1 and k_2, that is:

$$k_{mi} = \frac{k_1 k_2}{k_1 + k_2} \tag{1}$$

In general, the overlap degree of spur gears is between 1 and 2, so there is an alternation of one pair of teeth meshing and two pairs of teeth meshing during the meshing process. When two pairs of teeth mesh, their respective meshing stiffness k_{m1} and k_{m2} can be regarded as parallel, and the comprehensive meshing stiffness of the entire gear pair can be expressed as:

$$k_{mi} = k_{m1} + k_{m2} \tag{2}$$

Due to the different deformations of each gear tooth at different meshing positions on the tooth profile, the meshing stiffness k_{mi} of the gear teeth is a function of the meshing point position and has time-varying and abrupt characteristics. Considering the alternating changes between single tooth meshing and double tooth meshing during the meshing process, the meshing stiffness exhibits significant periodicity. To normalize the meshing section, the meshing stiffness is uniformly represented as a Fourier series with the meshing frequency as the fundamental frequency:

$$k_{mi} = k_0 + \sum_{i=1,n} k_i \cos(iwt) \tag{3}$$

w: Gear pair meshing frequency.

The comprehensive meshing stiffness variation curves corresponding to Eq. (3) for single tooth and multi tooth meshing are shown in Fig. 2. After normalization, the curve achieved continuity in time series. When conducting actual analysis, the curve can be approximated as a rectangular wave function.

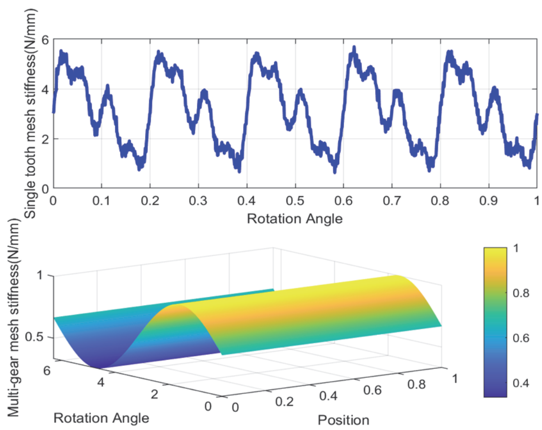

Figure 2 Normalized comprehensive meshing stiffness curve of tooth pairs

Table 2 Time domain characteristic parameters

$p_1 = \max\|x\|$	$p_2 = \max\|x\| - \min\|x\|$
$p_3 = \dfrac{1}{N}\sum_i x_i$	$p_4 = \sqrt{\dfrac{1}{N}\sum_i x_i^2}$
$p_5 = \sqrt{\dfrac{1}{N-1}\sum_i (x_i - p_3)^2}$	$p_6 = \dfrac{1}{N}\sum_i \left(\dfrac{x_i - p_3}{p_5}\right)^4$
$p_7 = \sum_i \left(\dfrac{x_i - p_3}{p_5}\right)^3$	$p_8 = \dfrac{p_1}{\frac{1}{N}\sum_i x_i}$
$p_9 = \dfrac{p_7}{\frac{1}{N}\sum_i x_i}$	$p_{10} = \dfrac{p_1}{\frac{1}{N}\sum_i x_i^2}$

When constructing the eigenvector of each sample, the improved method is used to decompose the signal, and the 13 time domain characteristic parameters of the first component of the decomposition result are calculated. The amplitude spectral entropy, envelope spectral entropy, and energy of the first two components, as well as the energy entropy of the decomposition result, are extracted, resulting in a total of 20 feature values, then normalize all feature values and transform them into intervals of [0, 1]. The time-domain characteristic parameters are shown in Tab. 2.

Firstly, the simulation signal is decomposed and the above feature values are extracted. Then, the projection matrix is calculated to project the feature values and determine whether the goal of removing working conditions can be achieved.

$$\begin{aligned} y(t) &= A\sin\left(w_n\sqrt{1-\lambda^2 t}\right) \\ w_n &= 2\pi f_n \\ s(t) &= y(t) + n(t) \end{aligned} \tag{4}$$

In the formula, A is the vibration amplitude caused by the correction defect, r is the damping coefficient, f is the natural frequency, and $n(t)$ is the random noise signal. By simulation, vibration signals with frequencies of 20, 40, 60, and 80 Hz were obtained. 50 sets of data were taken for each frequency, with 1024 points per set. The time-domain diagrams of simulation signals under different frequencies are shown in Fig. 3.

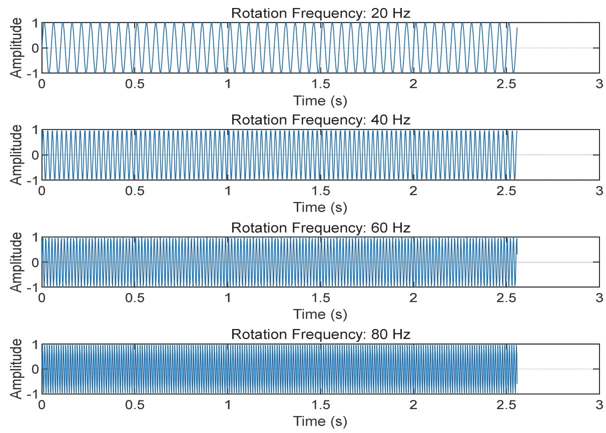

Figure 3 Simulation signals of bearing correction under different rotational frequencies

3.3 Establishment of Gear Pair Dynamics Model

Considering that the flexibility of the transmission shaft in general gear transmission is smaller than that of the teeth, and it is not in the same order of magnitude. Therefore, a dynamic model of the gear pair can be established when only considering the elastic deformation at the gear pair, as shown in Fig. 4. Among them, the base circle radii of the active and driven gears are R_p, R_g, and k_m respectively, which are the comprehensive meshing stiffness of the gear pair. There is also an interference of tooth profile error $e(t)$ between the two teeth.

If the overlap degree during the gear meshing process is not greater than 2, the comprehensive deformation of the logarithm i of the meshing teeth on the meshing line can be expressed as:

$$\delta_i = \theta_p R_p - \theta_g R_g - e_i \tag{5}$$

In the formula: the tooth profile error of e_i meshing teeth to l. The dynamic normal meshing force of gears, including meshing damping, can be expressed as:

$$F = \sum_{I} k_{mi}\delta_i + c_m\delta_i \tag{6}$$

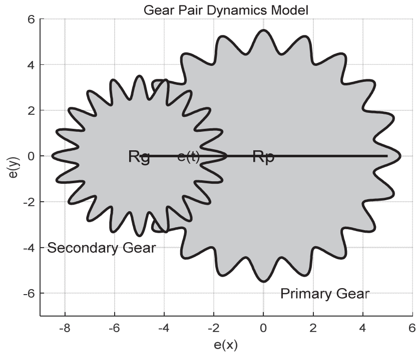

Figure 4 Dynamic model of gear pair

The torque balance equation for each gear is:

$$T_p = R_p \sum\sum_{I} k_{mi}\delta_i + c_m\delta_i \tag{7}$$

As can be seen from the above formula, even if the speed of the driving gear and transmission load are constant, the change of the comprehensive meshing stiffness k_m will cause the fluctuation of the driven wheel speed c_m, resulting in the circumferential vibration of the gear and the dynamic meshing force in the transmission, thus forming vibration incentive for the whole system. For the convenience of discussing the time-varying comprehensive meshing stiffness k further simplify the model based on its impact on the dynamic equations. The relative displacement x between two gears on the meshing line can be defined as:

$$x = \theta_p R_p - \theta_g R_g \tag{8}$$

The gear vibration during normal operation is characterized by internal excitation vibration, and the change in the stiffness of the meshing teeth forms an excitation force between the teeth of the meshing pair. The corresponding natural frequency of the system can be expressed as:

$$w_n = \sqrt{\frac{\sum k_{mi}}{m_e}} \tag{9}$$

Obviously, the natural frequency of the system is also a function of the comprehensive meshing stiffness, that is, there are significantly different natural frequencies during the meshing of one pair of gears and two pairs of gears, which is significantly different from the natural frequency characteristics of general mechanical transmissions. From Fig. 5 and Fig. 6, it can be seen that within one meshing cycle of the gear teeth, it is precisely due to the continuous variation of the comprehensive meshing stiffness that the vibration excitation response effects of the single tooth meshing section and the double tooth meshing section are different. In single tooth meshing, the average value and abrupt change amplitude of tangential acceleration are both large. When double tooth meshing, due to the smaller load on each pair of meshing teeth compared to single tooth meshing, the tangential acceleration response value is smaller. Although the combined strain of two pairs of teeth is larger than that of single tooth meshing, its average value is still smaller than the strain obtained from single tooth meshing. Therefore, gear transmission is relatively stable when entering the double tooth meshing section, and the fundamental reason is that its vibration is relatively small, which is consistent with the conclusion of conventional theoretical analysis.

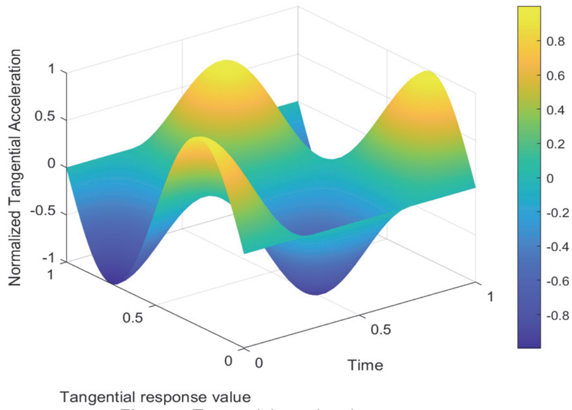

Figure 5 Tangential acceleration response curve

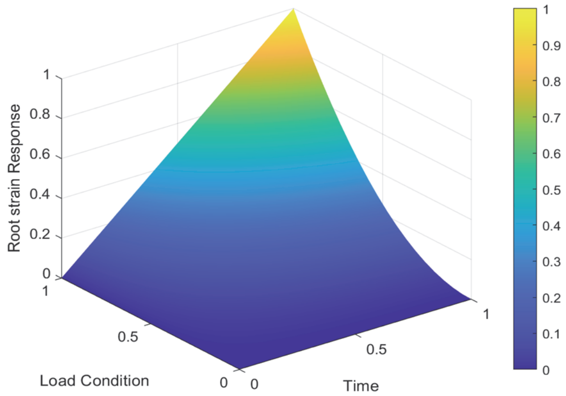

Figure 6 Root strain response curve

The meshing cycle is:

$$T = \frac{60}{nz_1} = \frac{60}{3.75 \times 47}\text{s} = 0.34\text{s} \tag{10}$$

The calculation results correspond to the total period of single tooth meshing and double tooth meshing in the acceleration curve in Fig. 5. Therefore, the above analysis results are in good agreement with the actual situation, and the model is reliable and suitable for analyzing the dynamic characteristics of gear pairs.

4 ESTABLISHMENT AND ANALYSIS OF A SUB DYNAMIC MODEL FOR GEAR MODIFICATION PARAMETER NORMALIZATION

In the process of modeling the mechanical joint surfaces of the gearbox, the main analysis is to model the mechanical joint surfaces of the gear transmission system.

The mechanical joint surfaces of the meshing of each gear, the rolling elements of each bearing, and the connection between the inner and outer rings are replaced by equivalent springs and dampers. In addition, in the process of modal analysis of the gearbox, it is assumed that the gearbox is a weak damping or proportional damping structure. Therefore, in the modeling process, the mechanical joint surfaces are simplified into a series of equivalent springs. Based on the different bonding conditions and states of the mechanical bonding surface, different numbers of bonding points, degrees of freedom for each bonding point, and equivalent spring stiffness for each degree of freedom are selected to simulate. As for the gear part, due to the ability of the herringbone gear to counteract axial forces, gear pair springs are only set up in the normal direction of a few tooth surfaces in the meshing state, and not in other directions. The internal mechanical joint surfaces of each bearing use rolling pair springs in the x, y, and z directions, and other fixed joint surfaces are constrained by different numbers of springs in the normal direction of the joint surface, as shown in the upper and lower boxes. Overall, the main steps of finite element model modification include establishing the finite element model, determining optimization objectives, optimizing parameters, as shown in Fig. 7.

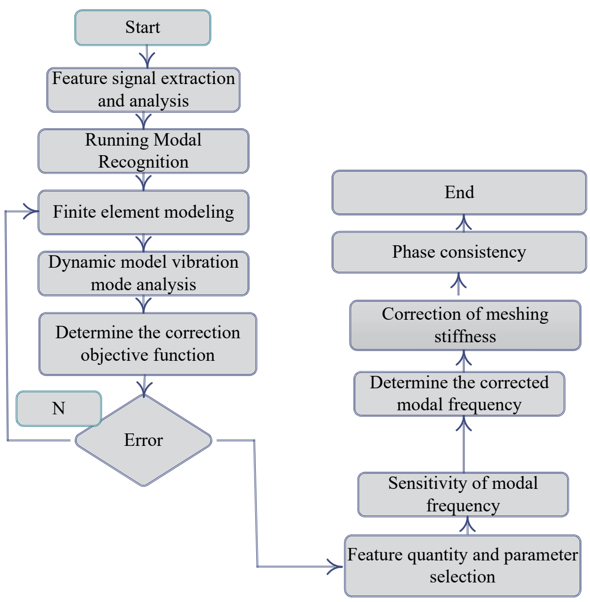

Figure 7 Basic process of finite element model revision

For modal analysis of mechanical structures, the sensitivity of the modal parameter T of the system to the J-th quantity in the design variable b is defined as:

$$S_j = \frac{\partial T}{\partial b_j} \tag{11}$$

After some normalization process (such as mass normalization), the orthogonality test formula for finite element calculation mode shapes and experimental mode shapes is:

$$k_t = \varphi_i^T S_j \varphi_i \tag{12}$$

The value of inertia factor determines the search performance. When the inertia factor is large, the algorithm has strong global optimization ability. When it is relatively small, it can ensure the precision of particle local search. Therefore, by adding a linearly decreasing inertia factor K to balance the performance of the algorithm search, during the iteration process, the current value of the decreasing inertia factor is:

$$k(t) = \frac{t_{\max} - t}{t_{\max}} \left(k_{\max} - k_{\min}\right) + k_{\max} \tag{13}$$

The basic parameters and force state of the gears are shown in Tab. 3.

Table 3 Gear Parameters and Meshing States

Modulus / mm	Number of teeth 1	Number of teeth 2	Tooth width / mm	D_n / mm	R_c / mm	T / N·m
2.5	47	61	21	31	58.6156	7.62

Finally, the sensitivity of the first 5 natural frequencies to the design variables was calculated, as shown in Tab. 4. At the same time, when taking the initial value, modal analysis is performed on the model to obtain the natural modes of each order of the system.

Table 4 Sensitivity of the first 5 natural frequencies to design variables

Natural frequency	1-st	2-nd	3-rd	4-th	5-th
Sensitivity	4.94 E–08	1.74 E–08	2.66 E–08	2.465 E–07	8.42 E-08

For gear pairs, a finite element model of tooth profile parabolic modification was used for simulation calculation and genetic algorithm was used for search. The optimal chromosome was decoded to obtain the gear modification parameters as shown in Tab. 5.

Table 5 Shape modification optimization results

Driving wheel		Driven wheel	
S	r	S	r
0.06435	64.218	0.06365	64.473

According to the modification data, simulation calculations were conducted to obtain the transmission error curve of the gear during parabolic modification, as shown in Fig. 8.

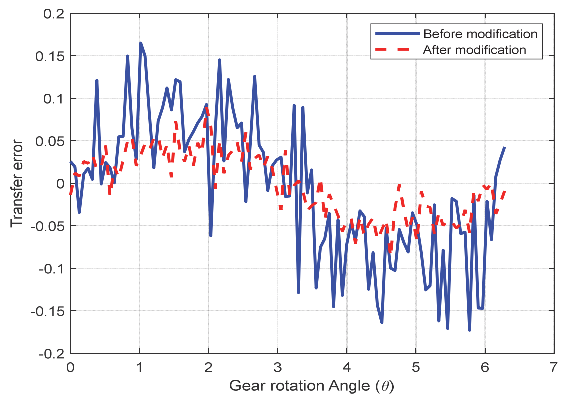

Figure 8 Dynamic curve of gear transmission error for parabolic modification

The dashed line in Fig. 8 represents the transmission error curve before the modification, indicating that the gear transmission error fluctuates greatly during the alternation of single and double teeth, indicating the presence of meshing in and out impacts on the gear, with a maximum fluctuation of 10 um. The solid line represents the transmission error curve after parabolic modification, and at this point, the curve no longer shows obvious stepped characteristics, indicating that meshing in and out impacts no longer exist. The peak in the double tooth meshing area indicates that the alternation of single and double teeth is the main cause of fluctuations. Its maximum fluctuation decreased to 8.023 um, a decrease of 20%, and the reduction in transmission error was not significant, indicating good straight line shaping. It should be pointed out that although straight line modification is better than parabolic modification in this example, this does not apply to all ranges, and in some cases, parabolic modification may be better than straight line modification. When designing a modification curve, the type and parameters of the modification curve must be determined based on the actual working conditions. The method proposed in this article is suitable for this requirement. Modal analysis was conducted on the beam shell element model using adhesive method without considering the connection relationship of various components. The existing finite element model of the beam shell element using adhesive method was applied, as shown in Tab. 6.

Table 6 Natural frequencies of gearboxes with bonding methods

Order	Natural frequency / Hz	Order	Natural frequency / Hz
1	76.458	7	250.33
2	126.8	8	253.38
3	137.49	9	255.85
4	139.67	10	257.46
5	145.66	11	272.89
6	158.38	12	274.97

The first four vibration modes are shown in Fig. 9. It can be seen from the Fig. 9 that 3 of the first 4 vibration modes of the model are reflected as the vibration of the thin shell part of the box, and only the component modes of the moving parts appear in the first vibration mode, that is, in the third vibration mode, the tooth rings of the big gear swing up and down around the x axis.

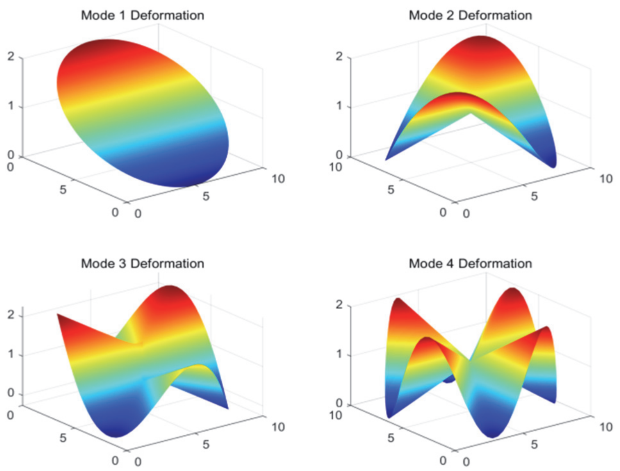

Figure. 9 The first 4 vibration modes of the structure based on the combined stiffness

5 SIMULATION VERIFICATION

To verify the rationality and effectiveness of the method in the article, the optimal value of gear modification amount obtained will be imported into the software for simulation experiments, and the noise, transmission error, and unit distributed load of the gears before and after modification will be analyzed. Noise data can intuitively reflect the dynamic performance of the gearbox transmission system. Through analysis, it can be seen that the transmission noise value before and after gear modification increases with the increase of experimental load, but the noise value after gear modification is significantly lower than before. Especially under a high load of 145 Nm, the noise value decreases by about 12.8 dB. This can indicate that reasonable gear modification can improve the meshing characteristics of gear transmission, effectively reduce the noise generated by meshing impact and tooth profile error. The load distribution per unit length of the sun gear and planetary gear after modification is shown in Fig. 10, and the comparison results of simulation results before and after gear modification under excavation conditions are shown in Tab. 7.

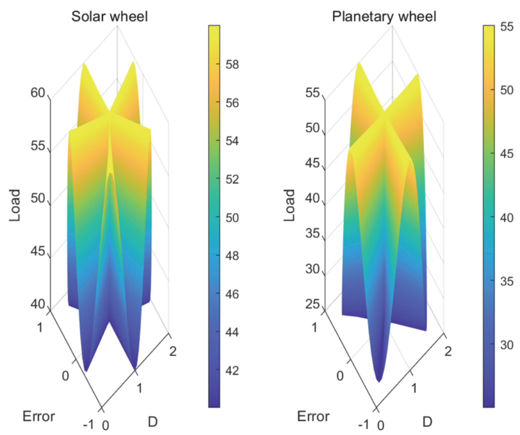

Figure 10 Load distribution per unit length after gear modification

Table 7 Comparison of simulation results before and after shape modification

Comparison item	Before shaping		After shaping		Changes	
	Solar wheel	Planetary gear	Solar wheel	Planetar y gear	Solar wheel	Planetary gear
Transmission error	0.216	0.387	0.1364	0.1754	–0.086	–0.302
Maximum load per unit length	215.3	238.7	243．5	267.8	28.2	29.1
vibration acceleration	67.1		38.1		29	

From Fig. 10 and Tab. 6, it can be seen that the load on the modified gear is concentrated at the tooth center, avoiding the occurrence of load deviation, making the gear transmission more reasonable and effective, and further improving the service life of the gear. According to the experimental results, it can be seen that the dynamic performance of gears can be significantly improved after micro modification. Gear transmission error, as one of the main sources of high-frequency noise, has reduced the transmission error amplitudes of the sun gear and planetary gear by 0.0876 and 0.302 respectively compared to before modification. The maximum unit length load of the modified gear has also been increased, but it does not exceed a reasonable range, and the impact is relatively small. Vibration acceleration is one of the key indicators to measure the size of noise. After modification, the maximum vibration acceleration of the gear decreased by 29 m/s^2, which can effectively suppress the generation of high-frequency transmission noise. The meshing stiffness of gears has a significant impact on the dynamic characteristics of the system, which can be demonstrated by drawing the maximum exponential and bifurcation diagrams with was the parameter. If the meshing stiffness is set as a constant, the bifurcation diagram and maximum exponent diagram of gear clearance vibration are obtained, as shown in Fig. 11.

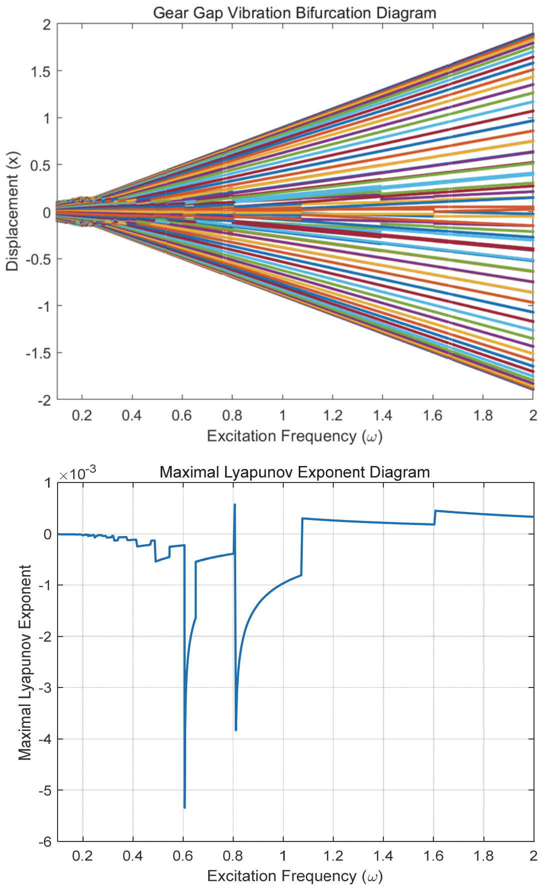

Figure 11 Bifurcation diagram and maximum exponential diagram of gear gap vibration

From Fig. 11, it can be seen that the system undergoes a period 1 motion from 0.54 to 0.76. At 0.77, the system becomes a period 2 motion, and then enters the chaotic region. In this chaotic region, the system has several period windows, and finally, at 1.67, the system leaves the chaotic region and enters a period 3 motion state. Under natural changes, the dynamic behavior of the system is quite diverse, including periodic motion and chaotic motion. From the maximum exponent diagram of the system, it can be seen that there are several periodic windows in the chaotic region, among which there is a wider period 3 window, which is consistent with the bifurcation diagram. The bifurcation diagram of the system with the variation of parameter w is shown in Fig. 12, the system undergoes quasi periodic motion when the parameters range from 1.3 to 1.8, and then enters the chaotic region. In this chaotic region, the system has four periodic windows, and finally leaves the chaotic region at 1.71. In the case of changes in w, the dynamic behavior of the system is quite rich, including periodic motion, quasi periodic motion, and chaotic motion.

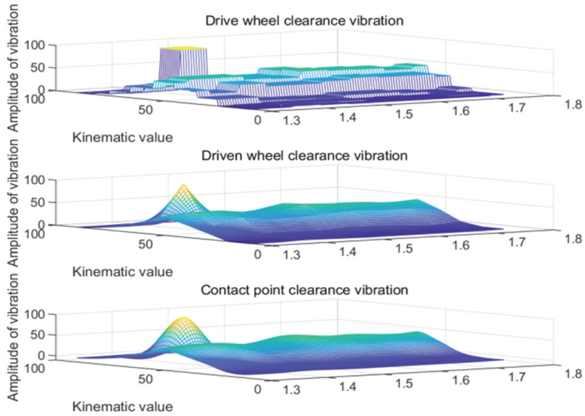

Figure 12 System bifurcation diagram with parameter *w* as the bifurcation parameter

Due to the fast calculation speed of gear shaft modeling and modal analysis, Monte Carlo probability design method is selected for reliability and reliability sensitivity calculation when the calculation time is acceptable. This can obtain more accurate calculation results. In the probability design process, the natural frequency of the gear is specified as the random output variable and the number of random simulations $N = 2300$ is specified to perform probability finite element analysis of the gear mode. Due to the large span of natural frequencies, in order to make the specified value Y better applicable to situations with multiple excitation frequencies, this example takes the ratio of the absolute value of the difference between natural frequencies and excitation frequencies to natural frequencies as the sampling result. According to the vibration failure criterion, conduct a vibration reliability analysis on the gear. Perform 36 cyclic sampling on the gear vibration modal analysis module to obtain its reliability and sensitivity values for various random parameters. Fig. 13 shows the historical curves of each parameter sample, reproducing the values of each parameter in 36 samples.

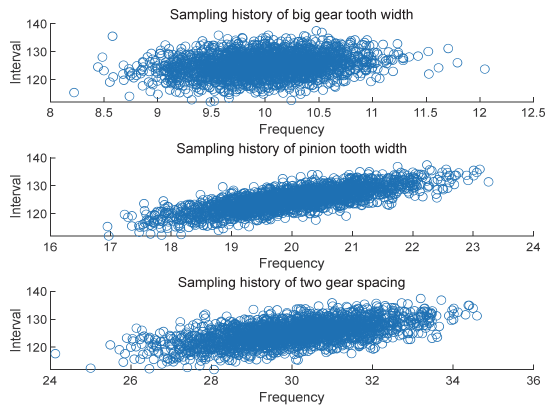

Figure 13 Sample historical curves of gear meshing parameters

It can be seen that the box modes of the two models are relatively similar, with similar natural frequency values; In the frequency response function curves of the two models, the peak value of the frequency response function curve based on the bonding method model mostly appears at higher natural frequencies, with all peaks being box mode. However, the frequency response function curve based on the bonding stiffness model has a wider peak distribution, with the larger peak value mostly at the frequency of the moving component mode. At the same time, its phase change is more frequent than that of the bonding method model.

6 CONCLUSION

In this paper, the modified method of gear structure dynamic parameter normalization model is applied to the dynamic reliability analysis of gear box, and a new method of gearbox dynamic reliability research based on the modified parameter normalization model is proposed. Based on the theory of stress intensity interference, an improved dynamic model of gearbox structure is established to predict the vibration response of the actual unmeasured points. Based on the failure mechanism of the first overtaking, the dynamic reliability sensitivity analysis and dynamic reliability analysis of the dangerous parts of the transmission are carried out. Through calculation, the three factors that have the greatest influence on the dynamic reliability of the gearbox are obtained, namely, the random stiffness of the gear transmission pair, the random stiffness of the input bearing in the horizontal direction and the load on the output shaft. In this paper, only the normalized model modification method of dynamic parameters of gear structure is used to optimize the modification parameters, and the common optimization methods such as genetic algorithm, neural network and simulated annealing are not used for lateral comparison selection, so as to further simplify the constrained nonlinear and multi-objective optimization problems, so as to strengthen the feasibility and effectiveness of the modification optimization design. The next step is to normalize the time sequence of the integrated meshing stiffness, calculate the resonance peak value from the corresponding dynamic model, and comprehensively consider the various structural parameters and motion parameters of the gear pair when carrying out gear design and maintenance.

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7 REFERENCES

[1] Wei, L., Liu, H., & Chen, H. (2020). Research on Parameter Optimization of Tracked Vehicle Transmission System Based on Genetic Algorithm. *Cyber Security Intelligence and Analytics, CSIA 2019, Advances in Intelligent Systems and Computing,* 928-936. https://doi.org/10.1007/978-3-030-15235-2_71

[2] Yan, Y., Tao, Q., & Wang, H. (2021). Noncyclic Scheduling of Multi-cluster Tools with Multi-type Wafers Based on Pareto Optimization. *13th International Conference on Computer and Automation Engineering (ICCAE)*, 95-99. https://doi.org/10.1109/ICCAE51876.2021.9426127

[3] Li, Z., You, K., & Song, S. (2021). Optimization-Based Control for Bearing-Only Target Search With a Mobile Vehicle. *IEEE Transactions on Systems, Man, and Cybernetics: Systems, 41*(7), 4149-4159. https://doi.org/10.1109/TSMC.2019.2932449

[4] Huang, Y., Xu, Z., & Tang, X. (2023). Sampled Value Optimization Based Bearing-only Localization. *Journal of Physics: Conference Series, 2656*, 12011-12023. https://doi.org/10.1088/1742-6596/2656/1/012011

[5] Han, S. T. (2021). A Total Lp-Norm Optimization for Bearing-Only Source Localization in Impulsive Noise with SαS Distribution. *Sensors, 21*(19), 6471-6484. https://doi.org/10.3390/s21196471

[6] Ding, N., Pan J., & Liu, J. (2019). An optimization method for energy structures based on life cycle assessment and its application to the power grid in China. *Journal of Environmental Management, 238*(15), 18-24. https://doi.org/10.1016/j.jenvman.2019.02.072

[7] Elplacy, F., Samuel, M., & Mostafa, R. (2022). Modelling and simulation of hot direct extrusion process for optimal product characteristics: Single and multi-response optimization approach. *Advances in Production Engineering & Management, 17*(1), 33-44, https://doi.org/10.14743/apem2022.1.419

[8] Liu, K., Wu, Y., & Ge, Z. (2018). Adaptive Multi-Objective Optimization of Bionic Shoulder Joint Based on Particle Swarm Optimization. *Journal of Shanghai Jiaotong University(Science), 23*(4), 550-561. https://doi.org/10.1007/s12204-018-1958-1

[9] Alamir, N. (2024). Optimizing Two-stage Energy Management in renewable-based Multi-Microgrid using a Modified Student Psychology-Based Optimization with Demand Response and Hydrogen Storage. *International Journal of Hydrogen Energy, 63*, 696-719. https://doi.org/10.1016/j.ijhydene.2024.03.169

[10] Singh, D. & Singh, B. (2019). Investigating the impact of data normalization on classification performance. *Applied Soft Computing, 97*, 105524-105536. https://doi.org/10.1016/j.asoc.2019.105524

[11] Chen, W., Shao, Z., & Wakil, K. (2020). An efficient day-ahead cost-based generation scheduling of a multi-supply microgrid using a modified krill herd algorithm. *Journal of Cleaner Production, 272,* 122364-122647. https://doi.org/10.1016/j.jclepro.2020.122364

[12] Tan, B., Chen, S., & Liang, Z. (2024). An iteration-free hierarchical method for the energy management of multiple-microgrid systems with renewable energy sources and electric vehicles. *Applied energy, 356*(C), 440-456. https://doi.org/10.1016/j.apenergy.2023.122380

[13] Mohy, U. D. G., Vu, D. H., & Muttaqi, K. M. (2020). An Integrated Energy Management Approach for the Economic Operation of Industrial Microgrids Under Uncertainty of Renewable Energy. *IEEE Transactions on Industry Applications, 9*, 635-647.

[14] Duan, Q., Sheng, W., & Wang, H. (2021). A Two-Stage Robust Optimization Method Based on the Expected Scenario for Islanded Microgrid Energy Management. *Discrete dynamics in nature and society, 2021*(Pt.8), 296-312. https://doi.org/10.1155/2021/7079296

[15] Restrepo, M., Caizares, C. A., & Simpson-Porco, J. W. (2021). Optimization- and Rule-based Energy Management Systems at the Canadian Renewable Energy Laboratory microgrid facility. *Applied Energy, 290*(6), 116760-116776. https://doi.org/10.1016/j.apenergy.2021.116760

[16] Chadly, A., Aldayyani, H. Y., & Hamasha, M. M. (2024). Selection of optimal strategy for managing decentralized solar PV systems considering uncertain weather conditions. *Scientific Reports, 14*(1), 62891-62897.

https://doi.org/10.1038/s41598-024-62891-6

[17] Bilal, A., Alzahrani, A., & Almuhaimeed, A. (2024). Advanced CKD detection through optimized metaheuristic modeling in healthcare informatics. *Scientific Reports, 14*(1), 63292-63297. https://doi.org/10.1038/s41598-024-63292-5

[18] Merjora, A. A. & Maran, P. S. (2024). Optimized shuffle attention based Lidar signal denoising and temperature retrievals in the middle atmosphere. *Optical and Quantum Electronics, 56*(7), 7022-7032. https://doi.org/10.1007/s11082-024-07022-1

[19] Al-Rahayfeh, A., Atiewi S., & Almiani, M. (2024). Utilizing 3D magnetic source imaging with landmark-based features and multi-classification for Alzheimer's Disease diagnosis. *Cluster Computing, 27*(3), 2635-2651. https://doi.org/10.1007/s10586-023-04103-w

[20] Kumar, S., Maity, S. R., & Patnaik, L. (2022). Wear Parameter Optimization of Ceramic Coating Using the Fuzzy Integrated PSI-CODAS Decision-Making Framework. *Arabian Journal for Science and Engineering, 2022*, 1-23. https://doi.org/10.1007/s13369-022-07212-7

[21] Degasperi, A., Nguyen, L., & Fey, D. (2022). Relationship Between Dimensionality and Convergence of Optimization Algorithms: A Comparison Between Data-Driven Normalization and Scaling Factor-Based Methods Using PEPSSBI. *Methods in molecular biology (Clifton, N.J.), 2385*, 91-115. https://doi.org/10.1007/978-1-0716-1767-0_5

[22] Zeng, W., Yi, J., & Lin, R. (2022). Statistical tolerance-cost-service life optimization of blade bearing of controllable pitch propeller considering the marine environment conditions through meta-heuristic algorithm. *J. Comput. Des. Eng., 9*, 689-705. https://doi.org/10.1093/jcde/qwac023

[23] Qiu, N., Jin, Z., & Liu, J. (2021). Hybrid multi-objective robust design optimization of a truck cab considering fatigue life. *Thin-Walled Structures, 162*(4), 107545-107564. https://doi.org/10.1016/j.tws.2021.107545

[24] Xia, Y., Yang, M., & Ji, Z. (2021).Design and dimension optimization of cutter disassembly mechanism for shield tunneling machine. *Journal of Mechanical Science and Technology, 35*(7), 3005-3018. https://doi.org/10.1007/s12206-021-0624-y

[25] Elmahdi, L. A., Xu, Y., & Khalil, E. (2021).Integrated Design Tool for Conceptual Design of Aerodynamic Surfaces Based on Multi-objective Genetic Algorithm. *2021 IEEE 7th International Conference on Control Science and Systems Engineering (ICCSSE)*, 262-267. https://doi.org/10.1109/ICCSSE52761.2021.9545200

[26] Cheng, Z., Ke-Li, W. U., & Xin, W. Y. (2018). Energy Consumption Optimization of Cognitive Radio Relay Network Based on Energy Normalization. *Acta Electronica Sinica, 46*(1), 230-235.

[27] Xuxiang, T. & Zhi, Y. U. (2018). Study of Longjing Tea Quality Detecting Method Based on Sensor Array and Non-Linear Signal Analysis Technique. *Sensors and Actuators, 31*(9), 1450-1455.

[28] Peyman, M., Sadegi, S., & Gorgani, H. H. (2018). Comparative study and multi-objective optimization of plate-fin recuperators applied in 200 kW microturbines based on non-dominated sorting and normalization method considering recuperator effectiveness, exergy efficiency and total cost. *International Journal of Thermal Sciences, 124*, 50-67. https://doi.org/10.1016/j.ijthermalsci.2017.10.001

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