Artículo de investigación

Robotic systems: optimization of stiffness characteristics

Роботизированные системы: оптимизация жесткостных характеристик

Recibido: 12 de octubre del 2019 Aceptado: 25 de noviembre de 2019

Written by:
Kamil Z. Khairnasov[120]
https://www.scopus.com/authid/detail.uri?authorId=57201774042

Abstract

Optimization methods of structure are considered. Methods have been developed for modeling robotic structure made of a multilayer composite material, including a three-layer one, consisting of external bearing layers and lightweight aggregate between the bearing layers. It is necessary to arrange the basis of the composite material along the lines of maximum stresses to obtain the most stiffness composite structure. The structure of a homogeneous material under operating loads was calculated to determine the lines of maximum stresses at the first stage, and the trajectories of the maximum tensile and compressive stresses were determined. The structure was modeled by composite materials at the second stage, and the basis of the composite material was located along the found paths of maximum stresses. The trajectories arrangement of the basis of the composite material were adjusted according to the results of the composite structure calculation at the third stage. The process continued until the true location of the composite base along the maximum stress paths was achieved. A multi-stage dynamic stand was considered as a robotic system designed for semi-natural modeling. The technique is proposed for approximating parts of robotic systems containing ring gears, motors, reduction gear, and bearing supports. The essence of the method was to determine the stiffness of ring gears, motors, reduction gear, bearing supports based on analytical methods. The resulting stiffness were assigned to the rod systems in the future, replacing the elements under consideration. Methods and algorithms for the frequency analysis and the optimization of the semi-natural modeling stands designed to simulate the flight characteristics in laboratory

Аннотация

Рассмотрены методы оптимизации конструкций. Разработаны методики моделирования роботизированных конструкций, из многослойного композитного материала, в том числе трехслойного, состоящего из внешних несущих слоев и легкого заполнителя между несущих слоев. Для получения максимально жесткой композиционной конструкции необходимо располагать основу композитного материала по линиям максимальных напряжений. Для определения линий максимальных напряжений на первом этапе проводился расчет конструкции из однородного материала при действии эксплуатационных нагрузок, и определялись траектории действия максимальных растягивающих и сжимающих напряжений. На втором этапе конструкция моделировалась композиционными материалами, и основа композитного материала располагалась по найденным траекториям максимальных напряжений. На третьем этапе происходила корректировка траекторий расположения основы композиционного материала по результатам расчета композиционной конструкции. Процесс продолжался до достижения истинного расположения основы композита по траекториям максимальных напряжений. В качестве роботизированной системы рассматривался многостепенной динамический стенд, предназначенный для полунатурного моделирования. Предложена методика аппроксимации деталей роботизированных систем, содержащих зубчатые венцы, двигатели, редукторы, подшипниковые опоры. Суть метода

[120] PhD in Technical sciences, Associate Professor, Department of Instrument Engineering Technology, Moscow Aviation Institute (National Research University), 125993 Volokolamskoe highway 4, Moscow, Russia

conditions have been developed. The developed methodology and algorithms allow you the following: to determine the frequency characteristics of the stand; to optimize the stands' characteristics of various structures that have great technical, scientific and applied value. The analysis was carried out using finite element and analytical methods. The convergence of the calculation results was checked by increasing the number of finite elements, i.e. the thickening of the approximation grid. The last partition was considered sufficient to obtain reliable calculation results when the partition results of the previous and subsequent, the smaller ones do not differ by more than 3%. The data obtained as a result of the study for test problems were compared with the available experimental data. The good agreement was observed between theoretical and experimental results. The discrepancy was not more than 10%. The good agreement between theoretical and experimental results was observed. The discrepancy was not more than 10%.

Keywords: Composite materials, finite element method, robotic systems, stands, stiffness optimization, vibrations.

заключалась в определении жесткости зубчатых венцов, двигателей, редукторов, подшипниковых опор на основе аналитических методов. В дальнейшем полученные жесткости присваивались стержневым системам, заменяющим рассматриваемые элементы. Разработаны методики и алгоритмы частотного анализа и оптимизации стендов полунатурного моделирования, предназначенного для имитации полетных характеристик в лабораторных условиях. Разработанные методология и алгоритмы позволяют: определять частотные характеристики стенда; оптимизировать характеристики стендов различных конструкций, имеющих большое техническое, научное и прикладное значение. Анализ проводился с применением конечно-элементных и аналитических методов. Сходимость результатов расчетов проверялась путем увеличения числа конечных элементов: сгущения сетки аппроксимации. При расхождении результатов разбиения предыдущего и последующего, более мелкого, не более чем на 3% последнее разбиение считалось достаточным для получения достоверных результатов расчета. Полученные в результате исследования данные для тестовых задач сравнивались с имеющимися экспериментальными данными. Наблюдалось хорошее согласование теоретических и экспериментальных результатов. Расхождение составило не более 10%.

Ключевые слово: колебания, композитные материалы, метод конечных элементов, оптимизация жесткости, роботизированные системы, стенды.

Introduction

Optimization issues for complex structures containing ring gears, bearings, reduction gear, motors, etc. a complex task requiring a large amount of research, experimental research in the process of development and manufacture (Berbyuk, 1989; Mehlenhoff, Bloedorn, 2010). The optimization process consists, as you know, in the maximum functions that determine the operability of mechanisms, at acceptable costs. It is necessary to distinguish from the many characteristics of the device those that are critical to device performance efficiency to solve the optimization problem. The optimization methods are different in each case.

For example, a topology optimization is an optimization method that, within a given volume with constant loads, boundary conditions, allows you to get the maximum value of the parameter by arranging the material in a given space, which contributes to the greatest system performance. The topological optimization is different from form optimization and size optimization in the sense that the project can take any form within the structural design space instead of dealing with predefined configurations.

The topological optimization allows you using the finite element method to change the mass and improve the stiffness characteristics of structures. The topology optimization is the change in structure, including the creation of new body boundaries and the removal of existing ones.

The goal of the topological optimization is to increase or decrease the given structure property (for example, decrease the strain energy, increase the main eigenfrequency) while satisfying certain conditions (for example, reduce material consumption). Direct work begins on the topological optimization of the structure after solving the strength problem. The objective function is formulated, i.e. a decrease in the compliance of a structure experiencing one or several loading options, an increase in eigenfrequencies, a restriction on deformations, or another.

The structural design space for the correct operation of the algorithm is described, i.e. the structure area, the topology of which can change, and the region for which changes are prohibited are determined. Additional convenience is provided by tools that control the possibility of manufacturing the resulting topology. These tools allow you to require compliance with linear or cyclic symmetry of the topology, to check the possibility of manufacturing parts by molding or milling, the absence of internal cavities. It is also possible to control the minimum or maximum size of newly obtained structural topology elements.

It remains to indicate the desired percentage reduction in material consumption and start the calculation after setting all the necessary settings for the optimization algorithm.

Theoretical basis

The article discusses the methodology for optimizing the stiffness characteristics of robotic systems using the example of multi-stage dynamic stands for semi-natural modeling with various layout options of the stand, namely various structural and kinematic schemes of pitch and roll channels (Turcic, Midha, 1984; Kolovsky, Sloushch, 1988; Chernousko, Bolotnik, Gradetsky, 1989; Jonker, 1990).

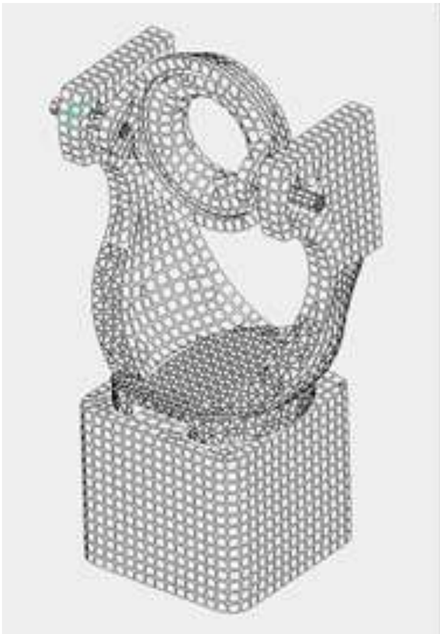

Figure 1. Finite element approximation of the stand model

We study a three-degree stand for definiteness, having a frame (pitch and roll channels have the form of a frame and rotate in mutually perpendicular planes) and a turret (pitch channel has the shape of a ring in the plane of which the roll ring rotates) component layouts (Khairnasov, 2001; Khairnasov, 2018; Khairnasov, 2019). The general view of the stand and its finite element approximation are shown in Figure 1.

The influence of the stiffness characteristics of the structure as optimization parameters and, the kinematic nodes of the stand in particular, containing bearing supports and reduction gears

ISSN 2322- 6307

on the frequency characteristics of the stand is considered. Tasks of this kind are of great practical importance, since the higher the eigenfrequency characteristics of the stand, the wider the range of operating frequencies not subject to interference from resonance. The tightening of the structure by increasing the thickness or volume (due to the spacing of the walls) leads to the increase in inertial characteristics or structural difficulties, which worsens and complicates the work of the stand. Increasing the stiffness of the structure by selecting bearings or reduction gear of increased accuracy helps to obtain the stand structurally identical to the original one with the wider range of operational characteristics. The product is more expensive, if the accuracy requirements are higher at the same time.

The considered dynamic stand of the semi-natural modeling is a three-degree slewing mechanism experiencing dynamic loads in the given ranges. The general view of the stand and its finite element approximation are shown in Figure 1.

The most suitable for modeling the structure of the stand is a shell of triangular or quadrangular finite element having six degrees of freedom in each node, taking into account both bending and membrane deformations.

The weakest link in discrediting the stand is the simulation of ball bearings. The challenge is to consider their interaction with the structure. Geometrically identical modeling of ball bearings leads to the unjustified increase in the number of equations, the inaccuracy of solution of which negates the efforts for accurate modeling; therefore, the assumptions were taken to account for ball bearings, allowing to take into account the mechanism of interaction of the ball bearing with the structure with sufficient reliability.

The assumptions are as follows:

- The ball was replaced by a rod element having a hinged bearing at the ends, which largely corresponds to the behavior of the ball in the bearing;
- The stiffness of the rod system simulating a ball bearing was taken equal to the stiffness of the ball bearing.

The reduction gears included in the stand were calculated separately for ease of calculation, using the developed program, and later the elastic ductility of the reduction gears was taken into account when calculating the structure (Sunada, Dubowsky, 1981; Hoopert, 2014).

The stand body was modeled by composite materials. Moreover, sections consisting of thin-walled elements were modeled by multilayer composite materials. The stand parts, consisting of spatial elements, were modeled by three-layer shells consisting of external bearing multilayer layers of composite and a filler layer between the bearing layers, consisting of light porous material, which perceives mainly shear stresses and prevents the approach of the bearing layers. The procedure for modeling the stand made of composite material was as follows. It is necessary to arrange the basis of the composite material along the lines of maximum stresses to obtain the most rigid composite structure. The structure of a homogeneous material under operating loads was calculated to determine the lines of maximum stresses at the first stage, and the trajectories of the maximum tensile and compressive stresses were determined. The structure was modeled by composite materials at the second stage, and the basis of the composite material was located along the found trajectories of maximum stresses. There was the trajectories adjustment of the location of the composite material base according to the calculation results of the composite structure in the third stage. The process continued until the true location of the composite base along the trajectories of maximum stresses was achieved (Savin, 2012; Grover, Singh, Maiti, 2013; Jones, 2014; Koutromanos, 2018).

Methodology

The slewing mechanism of the stand was modeled by the combination of the finite elements (FE) described above. The total number of FE was 310070, including rod elements taking into account bearing supports, ring gears and reduction gear.

The accuracy of the results was checked by converging the results depending on the number of partition elements. The results of the stress-strain state when divided into 502306 FEs and the adopted scheme differed by an acceptable error for the accepted type of calculations 2% ... 3%.

The program accuracy was verified by solving test problems and comparing the results with known analytical solutions. The discrepancy between the results was observed in the third decimal place.

The solution to the problems of determining the dynamic characteristics of the stand can be obtained from the Lagrange equation (Obraztsov, Volmir, Khairnasov, 1982; Khairnasov, 2001):

$$\frac{d}{dt}\frac{\partial T}{\partial \{\dot{q}\}} + \frac{\partial U}{\partial \{q\}} = \{Q\}, \qquad (1)$$

where $\{\dot{q}\}$ is the generalized speed, $\{q\}$ is the generalized displacement;

$T = [m] \times \{\dot{q}\} \times dA$ is the kinetic energy;
$U = \langle\sigma\rangle \times \{\varepsilon\} \times dA$ is the potential energy;
$Q = \langle p\rangle \times \{u\} \times dA$ is the vector of external forces;

$[m]$ is the matrix of mass characteristics of material, $\langle\sigma\rangle$ is the stress vector, $\{\varepsilon\}$ is the strain vector, $\langle p\rangle$ is the external forces, $\{u\}$ is the displacement vector, dA is the elementary volume, through $\langle\ \rangle$ and $\{\ \}$ the row vector and the column vector are respectively indicated; the dot above the letter means differentiation with respect to time *t*.

We take into account the dependencies:

$$[\sigma] = [E] \times \{\varepsilon\}, \rightleftarrows \quad \rightleftarrows \{\varepsilon\} = [S] \times \{u\},$$
$$\{u\} = [L] \times \{q\}, \quad \{\dot{u}\} = [L] \times \{\dot{q}\},$$

where $[L]$ is the matrix of mechanical characteristics of material, $[S]$ is the deformation and displacement coupling matrix, $[L]$ is the matrix of the transition from displacements $\{u\}$ to generalized displacements, and integrating the components of equation (1) over the generalized velocities and displacements, we reduce equation (1) to the following form:

$$[M] \times \{\ddot{q}\} + [K] \times \{q\} = \{Q\} \qquad (1)$$

Outlined here, [M] is the mass matrix, $[K]$ – is the stiffness matrix;

$\{Q\}$ – is the vector of external forces, $\{\ddot{q}\}$ – is the generalized acceleration, $\{q\}$ – is the generalized movement, through $\{\ \}$ vector column is indicated, a dot above the letter means differentiation with respect to time *t*.

We set the solution of equation (1) in the following form: $q = C_1 \times \sin(\omega \times t) + C_2 \times \cos(\omega \times t)$ and equating the vector of external forces to zero, we obtain the equation for determining the natural vibrations of the structure:

$$(\,[M] \times \{\omega\} + [K]\,) \times \{q\} = \{0\} \qquad (2)$$

The determination of the eigenfrequencies and vibrational forms of the structure is reduced to the determination of the eigenvalues and eigenvectors of the matrix $[K]^{-1}[M]$.

The condition for maximizing frequency characteristics was adopted depending on the stiffness characteristics of the bearing supports as an optimization criterion. This suggests that the movable support stiffness, at which the eigenfrequency of the stand did not depend on the support stiffness, but was determined by other structural elements, and was taken as a quantitative characteristic of optimization.

Lagrange multiplier method was used as a mathematical method for solving the optimization problem.

We write the Lagrange function as follows:

$$U = F(x, y, z) + \lambda \times \phi(x, y, z) \qquad (3)$$

where λ is the Lagrange multiplier, $F(x, y, z)$ is the minimized function, $\phi(x, y, z)$ are the restrictions imposed on $F(x, y, z)$.

and differentiate it by independent variables, we obtain a system of equations that allows minimizing the function $F(x, y, z)$, subject to restrictions $\phi(x, y, z) = 0$:

$$\begin{cases} \dfrac{\partial U}{\partial x} = \dfrac{\partial F}{\partial x} + \lambda \times \dfrac{\partial \varphi}{\partial x} \\ \dfrac{\partial U}{\partial y} = \dfrac{\partial F}{\partial y} + \lambda \times \dfrac{\partial \varphi}{\partial y} \\ \dfrac{\partial U}{\partial z} = \dfrac{\partial F}{\partial z} + \lambda \times \dfrac{\partial \varphi}{\partial z} \\ \varphi(x, y, z) = 0 \end{cases} \qquad (4)$$

The procedure for optimizing the stiffness characteristics of multi-stage dynamic stands for semi-natural modeling is reduced, in the notation given in formulas (3), in general form to the following:

ISSN 2322- 6307

- CAD-designed stand is approximated by finite elements;
- Frequency characteristics of the stand are determined for various rigidity values of the movable supports;
- Reveals the stiffness influence of each support on the frequency of the stand;
- Dependences of the stand frequency characteristics on the stiffness of the movable supports and their numbers are built:$F(x, y, z) = 0$;
- Minimum frequency of the stand structural elements is determined, assuming that the stand supports are absolutely rigid $\omega = const$, which is the equation of the plane in the coordinates frequency-rigidity-channel;
- Curve obtained by the intersection of the surface $F(x, y, z) = 0$ and the plane $\omega = const$, is a restriction function $\phi(x, y, z) = 0$ in the Lagrange method;
- Substituting the obtained functions in the system of equations (4), we obtain the extreme stiffness characteristics of the movable supports of the stand;
- Analyzing the data, we obtain the optimal stiffness characteristics.

Results

The influence dependencies of changes in the stiffness of bearings and reduction gears on the frequency characteristics as a result of the study were determined. The analysis of the obtained vibration frequencies of the stands of various configurations for different stiffness parameters of the bearing units and reduction gears showed that the change in the rigidity of the kinematic units is nonlinearly dependent on the eigenfrequencies of the stand vibrations and the frequency characteristics influence to varying degrees. It was found that the tightening of the lower bearing intended for rotation of the course rack increases the lower frequency compared with the upper bearing and reduction gear, which provides rotation of the pitch and course channels.

Thus, it is necessary to use bearings with increased accuracy of the course rack to provide higher low stand frequencies. Improving the accuracy of the kinematic mechanisms of the pitch and especially roll channels do not lead to the working range expansion of the stand, increasing only the higher frequency ranges.

The applied calculation procedure is in good agreement with the available experimental data of multi-stage dynamic stands.

Conclusion

The robotic systems are complex structures with many factors affecting their performance and stiffness. The stiffness characteristics of robotic systems affect the eigenfrequencies of vibrations of these systems. The robotic systems eigenfrequency is important at the same time, since the coincidence of vibrations eigenfrequencies with the frequency of forced or working frequencies of vibrations leads to resonance phenomena that adversely affect the accuracy of the robots positioning.

The presented methodologies for modeling robotic structures using the example of a semi-natural modeling stand with composite materials having high characteristics of specific strength make it possible to arrange the basis of the composite material in such a way as to obtain the most durable and rigid structure. The developed methodologies for optimizing robotic systems allow, within the framework of constant structural elements, to achieve the maximum level of the minimum eigenfrequencies of the considered systems and thereby derive the range of eigenfrequencies of robotic systems from the range of forced or working frequencies of vibrations.

References

Berbyuk V.E. (1989). Dynamics and optimization of robotic systems. Kiev: Naukova Dumka.

Chernousko F.L., Bolotnik N.N., Gradetsky V.G. (1989). Manipulation robots: dynamics, control, automation. Moscow: Nauka.

Grover N., Singh B.N., Maiti D.K. (2013). Analytical and finite-element modeling of multilayer composite and three-layer plates: Evaluation of the new theory of shear deformation for the free vibration response. International Journal of Mechanical Sciences. 67, 89-99.

Hoopert I. (2014), Advanced analytical approach for calculating ball and roller bearings, Journal of Tribology. 136, 11105-11116.

Jones R.M. (2014). Mechanics of Composite Materials. London: Taylor and Francis.

Jonker B.A (1990). Finite element dynamic analysis of flexible manipulators. The International Journal of Robotics Research. 9, 4, 5974.

Khairnasov K.Z. (2018). Mathematical modeling of shell configurations made of homogeneous and composite materials experiencing intensive short actions and large displacements. 5th International Conference on Topical Problems of

Continuum Mechanics with a Special Session in Honor of Alexander Manzhirov's 60th Birthday 2–7 October 2017, Tsakhkadzor, Armenia. Journal of Physics: Conference Series, 991.

Khairnasov K.Z. (2019), Methodology for calculating the stress-strain state of robotic system made of composite materials, International Scientific and Practical Conference Engineering Systems. IOP Conference Series: Material Science and Engineering. 675.

Khairnasov K.Z. (2001). The methodology of static and dynamic analysis and optimization of layout schemes of multi-stage dynamic stands for semi-natural modeling. Proceedings of the International Scientific Conference "Architecture of shells and strength analysis of thin-walled building and engineering structures of complex form", Moscow.

Khairnasov K.Z. (2001). The methodology of static and dynamic analysis and optimization of layout schemes of multi-stage dynamic stands for semi-natural modeling. Proceedings of the International Scientific Conference "Architecture of shells and strength analysis of thin-walled building and engineering structures of complex form", Moscow.

Kolovsky M.Z., Sloushch A.V. (1988). The basics of industrial robot dynamics. Moscow: Nauka.

Koutromanos I. (2018). Applied Fundamentals of Finite Element Analysis. Linear Finite Element Analysis. New York: John Wiley & Sons.

Mehlenhoff T., Bloedorn C. (2010). Solution for automated drilling in composite structures with a standard articulating robot system. SAE International.

Obraztsov I.F., Volmir A.S., Khairnasov K.Z. (1982). Toroidal shells: delayed catastrophes under dynamic loading. Report of the USSR Academy of Sciences. 266(6).

Savin S.P. (2012). The use of modern polymer composite materials in the structure of the glider of the aircraft family MS-21. Bulletin of the Samara Scientific Center of the Russian Academy of Sciences. 14(4), 686 pages.

Sunada W., Dubowsky S. (1981). The Application of finite element methods to the dynamic analysis of flexible spatial and co-planar linkage systems. Journal of Mechanical Design. 103, 643-651.

Turcic D., Midha A. (1984). Generalized equations of motion for the dynamic analysis of elastic mechanism systems. ASME Journal of Dynamic Systems, Measurements and Control. 106, 243-248.

ISSN 2322- 6307