Int. J. Advanced Design and Manufacturing Technology, 2024, Vol. 17, No. 2, pp. 23-31

DOI: 10.30486/ADMT.2024.873798 *ISSN: 2252-0406* *https://admt.isfahan.iau.ir*

Tolerance Geometrical Interference Analysis of Specific Movement Mechanism Under Thermal Strain

Ali Lashkari Zadeh, Seyed Yousef Ahmadi Brooghani, Mojtaba Sheikhi Azqandi*
Department of Mechanical Engineering,
University of Birjand, Birjand, Iran
E-mail: alashkarimec@birjand.ac.ir, syahmadi@birjand.ac.ir, mojtabasheikhi@birjand.ac.ir
*Corresponding author

Received: 11 September 2023, Revised: 7 April 2024, Accepted: 20 April 2024

Abstract: In sensitive mechanisms, achieving the required accuracy of the final parts at the most suitable price is a complex process. The tolerance design process is essential in design and manufacturing to achieve a quality and low-cost product. Allocation of appropriate tolerances is always time-consuming and challenging, especially for complex products, as it involves many aspects of design, manufacturing, and quality issues. In this research, a program has been written in MATLAB, in which by only one geometry model, all possible interferences of assembled parts for all tolerances and nominal sizes in different production modes have been investigated. By using this program, it is possible to add or reduce applied tolerances to nominal sizes and check the interference of parts for all tolerances. Also, temperature effects must be considered when designing a product that operates in a wide temperature range. This approach could ensure that the values of the output parameters of the mechanism remain stable with various temperature changes. The considered case study is particular clock whose structural parameter tolerances often significantly affect the accuracy of the timing output of the entire movement of the mechanism in the program; the tolerance of the parts of the delay clock mechanism has been studied. By analysing the tolerance of geometric non-interference and considering the strain of the parts due to the operating temperature of the mechanism, 24% of the examined sizes should be reduced to avoid interference and the tolerance of the axis position of the clock should be reduced to ±0.01.

Keywords: Geometric Interference Analysis, Mechanism, Operating Temperature Range, Operational Tolerance, Tolerance Analysis

Biographical notes: Ali Lashkari zadeh is a PhD student of Mechanical Engineering at the University of Birjand, Iran. His current research focuses on Tolerance analysis. **Seyed Yousef Ahmadi-Brooghani** is Professor of Mechanical engineering at the University of Birjand, Iran. He received his PhD in Mechanical engineering from University of Sheffield. His current research focuses on solid mechanics. **Mojtaba Sheikhi** received his PhD in Mechanical Engineering from University of Semnan in 2013. He is currently Associate Professor at the Department of Mechanical Engineering, University of Birjand, Iran. His current research interest includes optimal design of mechanical parts.

Research paper

1 INTRODUCTION

There are various unknown factors, such as manufacturing errors, which modify the performance of a produced mechanism. Forces, acceleration, and temperature changes cause changes in dimensions and, hence, design tolerances. As a result, they adversely affect the performance of the product. The tolerance analyses of many products have indicated a failure to meet design requirements, signaling the inefficient performance of the system under different operational conditions.

Tolerance analysis methods include worst-case tolerance analysis and statistical methods [1]. Worst-case tolerance analysis considers the worst possible combination of part tolerances, which leads to highly strict tolerances and increases manufacturing costs. On the other hand, statistical tolerance analysis is a more practical and economical tolerance adjustment approach that ensures good efficiency. Some researchers have used this method to examine product tolerances. For instance, Dantan et al. [2] employed the worst-case method based on the Monte Carlo simulation for tolerance analysis. Tolerance analysis can be performed using highly accurate methods of sensitivity analysis [3-4]. Doltsinis and Kong [5] utilized optimization methods, including sensitivity analysis and participation analysis, to design structural parameters. Tolerance analysis was also used by Zang et al. [6], Huang and Zhong [7], Zhou Kai et al. [8] to achieve an optimal design and Shao et al presented a tolerance analysis approach to spur gears [9].

In mechanism design, the best tolerance combination must be applied to manufacture parts by considering multiple objective functions, such as the provision of suitable spaces for parts, conformance to operational requirements, improvement of product quality, and reduction of manufacturing costs [10-12]. Numerous models have so far been developed to represent geometric features, cumulative deviation, and tolerance interval estimation [13].

Conventional tolerance modeling methods are generally classified into tolerance map (T-map), small displacement torsor (SDT), homogeneous transformation matrix (HTM), and other heuristic methods.

T-maps consider vectors for surfaces of various components. However, they use only five standards: planes, cylinders, spheres, cones, and toruses [14-17]. In addition, modeling techniques such as neural networks [18], volumetric envelopes [19], modal analysis [20-21], and graph theory [22-23] have been used to model tolerances between parts and assemblies. Previous research indicates that skin model shapes are very useful for representing product geometry by considering geometric deviations and form errors [24-26]. For instance, finite element analysis has been introduced to help reduce modeling errors and enhance model quality [27]. Researchers have demonstrated that discrete skin model shapes have a higher potential for assembly simulations, performance analysis, and tolerance analysis [28-30]. Regarding part contact and interference, Faraji and Abbasi [31] stated that the use of the average diameter for common form deviations cannot guarantee the acceptance or rejection of a part in geometric conformity quality control since the relationship between joint strength and average interference is not linear in these cases. This is due to the changes in the friction coefficient at the contact surface based on changes in the radius of curvature and the local contact pressure between the two surfaces. Hence, in the case of oval defects that reduce interference between the two parts below the nominal values, it is best to promptly address the factors causing these defects in the production line and remove defective parts from the production cycle. Researchers demonstrated that geometric parameters and tolerance analysis have a significant effect on the efficiency of mechanical systems [32-33].

In the analysis of mechanical assemblies, one must first perform an interference tolerance analysis before addressing dynamic requirements. Mechanical systems are in a static state before operation. Assimilability analyses of mechanical systems are performed under stationary conditions and fully depend on the tolerance ranges of parts. On the other hand, under static conditions, factors such as gear engagement or position sensor placement are of considerable importance at the beginning of motion or in other initial controls. Given the significance of dimensional and geometric tolerances in these cases, it is necessary to perform tolerance analyses of stationary mechanical systems in addition to examining the effects of external forces on them. For this purpose, the lack of part interference is defined as an objective function in the tolerance analysis of geometric interference.

In this research, a novel tolerance analysis method for mechanisms was presented in the form of a MATLAB program and was used to analyze a model created in SolidWorks. To implement this method, the program creates different tolerances in the three-dimensional model and examines their effect on the lack of interference between the model parts. It must be noted that the proposed technique and the written program do not belong to a specific mechanism and can be applied to any mechanism with any operation. This program calls the three-dimensional model created in SolidWorks only once. It then extracts the nominal values and part tolerances of the three-dimensional model as inputs. Next, each tolerance interval is divided into 10 or 100 equal parts, one or more of these parts are added to the nominal value step by step, and a geometric interference analysis is performed in each step.

For example, if a mechanism is composed of 10 parts, and each part has 10 significant dimensions, there will be a total of 100 significant dimensions in the problem. Assuming the tolerance intervals of each part are divided into 10, the parts can be assembled with different (permitted and unpermitted) tolerance configurations, leading to a very large possible design space. Accordingly, only one geometric model is created in the MATLAB program, with the nominal and upper and lower tolerance bounds defined as inputs. The nominal dimensions of this model are changed by a tenth (or hundredth) of the corresponding tolerance interval, and geometric interference is examined for each state. Subsequently, tolerances causing part interference in the mechanism are decreased by a tenth (or hundredth) in each step until no interference is observed. A major advantage of this method is the shorter time it needs to reach a set of tolerances that causes no interference. Another advantage is that the designer can ensure that the selected tolerance combination does not result in the locking of the mechanism during and after assembly.

Sensitivity analysis on mechanical systems, especially sensitive mechanisms, has certain complications [34-35]. According to the research done so far, the effect of temperature on the performance of sensitive mechanisms has not been done. So, the present research considers the effect of changes in temperature and the appearance of thermal strains and, thus, elastic deformation in the parts during operation. This is an important factor, especially in sensitive mechanisms, which must perform well under a wide range of temperatures. For this purpose, the temperature-related tolerance must be deducted from the design tolerance in part interference analysis. The proposed method is capable of simultaneously changing the design parameters. By analyzing the mechanism's performance under a wide range of temperatures, one can ensure the suitable performance of the product under this temperature range during its service life.

2 TOLERANCE ANALYSIS VIA PROGRAMMING

The proposed tolerance analysis is carried out as a systematic computer procedure. The first step involves an algorithm for computations designed to obtain the tolerances affecting part interference. The design specifications usually depend on the inherent properties of the system. Therefore, the dimensional and geometric deviations of the influential variables due to various sources can directly affect the performance of a mechanical system.

In a geometric interference analysis, possible part interference during assembly is examined by considering the tolerances of the manufactured parts. Using this part, one can modify the tolerances or the nominal dimensions of parts to prevent geometric interferences between them. To analyze geometric interference using the written MATLAB program, first, all the nominal part dimensions are applied to the initial model. Then, each dimension is changed based on the corresponding defined tolerance. Next, the new model is inspected in terms of interference. The main advantage of the proposed technique (including the MATLAB program) is that remodeling by the user is not required after changing the tolerances and the three-dimensional model.

3 TOLERANCE-BASED INTERFERENCE ANALYSIS

Before the tolerance-based interference analysis, the SolidWorks assembly must be inspected concerning interference in the nominal dimensions. First, the MATLAB program calls SolidWorks in the background and stores all the part dimensions in an Excel file. Next, the upper and lower tolerances suggested for each dimension are entered by the user (or these tolerances are extracted from the model if they have been entered in the model). Subsequently, the MATLAB program divided the interval between the nominal dimension and the upper bound into 10 or 100 increments based on the user's requirements and adds one increment to the nominal value in each step. Then, the three-dimensional model is updated, and the interferences in the model are examined. If any tolerance increment causes interference in a step, the tolerance value of the previous step is stored in an Excel file as the final value, and the program moves on to the next dimension. After all the upper tolerances have been inspected, the program begins examining the lower tolerances. In this case, a tenth or hundredth of the tolerance interval is deducted from the nominal value in each step, and the optimal lower bound is also stored in the Excel file. Figure 1 displays the flowchart of the tolerance-based interference analysis using the written program.

It must be noted that the program modifies the tolerances if this resolves interferences. Sometimes, constraint-based interferences may arise, in which case the constraints must be corrected. Finally, changes in the part lengths due to thermal strain within the operational range of the mechanism are deducted from the tolerance intervals, and the optimal part tolerances appropriate for that operational range are applied to the manufacturing drawings.

Get the 3D file

↓

Upload the Excel file of tolerance parameters

↓

Apply upper limit tolerance to parameters

↓

Model structural analysis

↓

Checking the interference of parts

↓

Reducing the tolerance of the parameters in order to eliminate the geometric interference of the parts

↓

Apply lower limit tolerance to parameters

↓

Model structural analysis

↓

Checking the interference of parts

↓

Reducing the tolerance of the parameters in order to eliminate the geometric interference of the parts

↓

Excel output with tolerance without geometric interference

Fig. 1 Steps of the tolerance-based interference analysis.

4 FULL DESCRIPTIONS OF THE TOLERANCE-BASED INTERFERENCE ANALYSIS

4.1. Geometric Modeling

In this part, the MATLAB program calls SolidWorks. MATLAB is linked with SolidWorks and performs the required modifications via related programs on the parts and assemblies designed in SolidWorks to obtain the desired outcome from the tolerance analysis.

Every part has a simple sketch and a few features. In SolidWorks, each dimension in the sketch is exclusively specified with a name. Each of these names is assigned a value, which is the nominal size of the corresponding dimension. For instance, if a part is stored with the name Part.1 and includes a rectangular shape, each dimension of the rectangle is given a name, such as D1. The MATLAB program recognizes these dimensions with the following expressions. An example of this dimensional designation is D1@Sketch1@1.Part.

Next, MATLAB reads the nominal dimensions of each part and places them in front of the corresponding dimension name stored in the previous step. Finally, the names and dimensions are stored in a file.

The upper and lower tolerance bounds of each nominal size are manually entered into the columns of the above-mentioned Excel file. Subsequently, the written MATLAB program examines the possible interferences caused by the tolerances in SolidWorks by considering their upper and lower bounds. Then, the tolerances written in the Excel file are read, and changes based on these tolerances are applied to the newly created model.

4.2. Changes in The Assembly Based on The New Dimensions

SolidWorks updates the model based on the tolerances in the Excel file for new interference analysis.

4.3. Inspection of Possible Interferences

In this step, the updated model is examined by the proposed method for possible interferences.

4.4. Storing of the Final Tolerance in The Output Excel File

After the final tolerance interval is determined from the interference analysis of the previous step, the new upper and lower bounds are stored in the final output file.

5 CASE STUDY

To demonstrate the implementation steps of the proposed algorithm and its effectiveness in tolerance analysis, this section investigates the mechanism of a highly sensitive and precise mechanical watch. The advanced analysis of geometric interference tolerance considers the presence of temperature variations exceeding one hundred and disregards the impact of component weights. The mechanism depicted in “Fig. 2” is a time-delay clock mechanism designed to function effectively within a temperature range of -33 to +71°C. This mechanism typically comprises three primary components: gear, weight gear, and escapement.

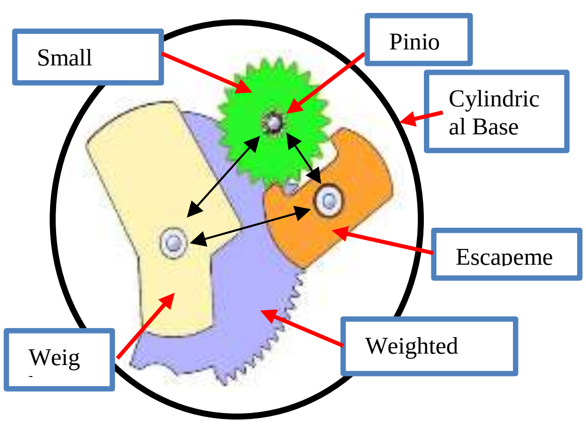

Fig. 2 Geometric model of the various components of the clock mechanism.

All the components of this mechanism are mounted on a cylindrical base plate, and based on experimental tests, its rotational speed is measured at 15000 rpm (revolutions per minute). In this mechanism, the weight positioned outside the center of the weighted component acts as the driving force, initiating the motion. The rotation of the weight and the weighted gear occurs slowly due to the interplay of gears among the components, and it completes its motion at a specific time. As shown in “Fig. 3”, the movement of the mechanism is considered complete when the initial position of the target circle aligns with the center of the time-delay set.

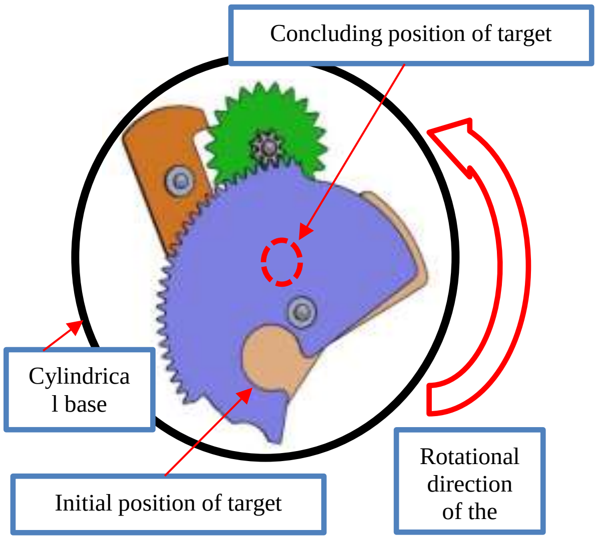

Fig. 3 The starting and ending positions of the motion.

According to the conducted experimental test, the target circle takes 0.07 seconds to reach the center of the assembly accurately. The weight located outside the assembly's center (weight and the weighted gear) is in the form of a bean-shaped weight, which is placed on the weighted gear and is also known as the weighted component. The weight initiates rotation along with the entire assembly. The motion of the weight positioned outside the center of the weighted component is transmitted to other parts by connecting it to the weighted gear, serving as the driving mechanism for the time-delay clock. This situation is depicted in “Fig. 3”.

The escapement is a mechanism that converts continuous rotational motion into intermittent movement. This component serves as an energy regulator and acts as a mechanical oscillator. Due to its control over the rotational speed of the weight, it is often recognized as the heart of the clock. One end of the escapement connects to the rotating base plate of the clock, while the other end engages with the small gear. By combining both rotational and linear forces, this mechanical system generates an oscillating motion. The functionality of this mechanism is significantly influenced by temperature and friction.

The escapement, positioned between the gear set and the oscillator, plays a crucial role in the motor's function. The regular back-and-forth motion of the oscillator determines the time interval that the escapement transfers to the weight through the gear set. This alternating movement, coupled with the friction between components, leads to a significant energy loss. As a result, the escapement holds the most potential for enhancing the performance of mechanical clocks. With each movement, when the escapement engages in one direction, the small gear returns it to its initial position. As shown in “Fig. 4”, the axes of the time-delay clock mechanism must be installed on a plate with a coordinate tolerance of 0.05 mm, after which the assembly is subjected to angular velocity. The material and mechanical properties of the components of the time-delay clock mechanism are provided in “Table 1”.

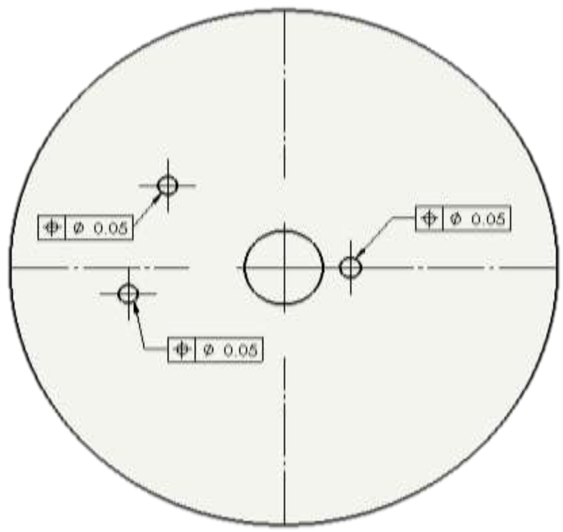

Fig. 4 The tolerance of axis positions (base plate).

Table 1 Material and mechanical properties of the components of the time-delay clock mechanism

Row	Name	Material	Young's modulus	Poisson's ratio	coefficient of thermal expansion
1	Small Gear	C26000	$10 * 10^{-11}$	0.32	$1.9 * 10^{-5}$
2	Pinion	AISI416	$2.07 * 10^{-11}$	0.29	$1.11 * 10^{-5}$
3	Escapement	C26000	$10 * 10^{-11}$	0.32	$1.9 * 10^{-5}$
4	weight	C26000	$10 * 10^{-11}$	0.32	$1.9 * 10^{-5}$
5	weight gear	C26000	$10 * 10^{-11}$	0.32	$1.9 * 10^{-5}$
6	Base Plate	AL2024-T6	$70 * 10^{-11}$	0.25	$2.38 * 10^{-5}$

As a continuation of this research, two components of the mechanism, namely the weighted component and the base plate, were selected. Six samples of each component were placed in a refrigerator at a temperature of -33°C for a duration of six hours, and one dimension of both components was measured using an optical measurement device. The results related to the weighted component are presented in “Table 2”, while the results related to the base plate are shown in “Table 3”.

Table 2 Effect of temperature on one dimension of six samples of the weighted component

Part Name	Weight piece	Nominal dimension (mm)	4.1mm
Sample	+21°C	-33 °C	Dimension difference
1	4.115	4.114	0.001
2	4.108	4.095	0.013
3	4.096	4.087	0.009
4	4.105	4.096	0.009
5	4.117	4.098	0.019
6	4.078	4.067	0.110

Table 3 Effect of temperature on a dimensional of six cylindrical base plate samples

Part Name	Base Plate	Nominal dimension (mm)	4.45mm
Sample	+21°C	-33 °C	Dimension difference
1	4.494	4.485	0.009
2	4.477	4.476	0.001
3	4.476	4.474	0.002
4	4.484	4.476	0.008
5	4.480	4.472	0.008
6	4.492	4.489	0.003

To highlight the importance of accounting for thermal strain within the operating temperature range of -33°C to +71°C, compared to the design tolerance of time-delay clock mechanism components, an analysis was conducted on the displacement of the main axes of the mechanism concerning the center point of the base plate. The analysis was carried out using the Static-General method. Initially, the ambient temperature was set at 21°C, and then the component experienced thermal stress, reaching temperatures of +71 and -33°C, respectively. The thermal analysis employed the RT8D3C mesh. To validate the results obtained in “Figs. 5a and 5b”, the mesh sensitivity analysis is presented at two temperatures of -33 and 71 degrees Celsius. As can be seen, there is a good compatibility between these two conditions.

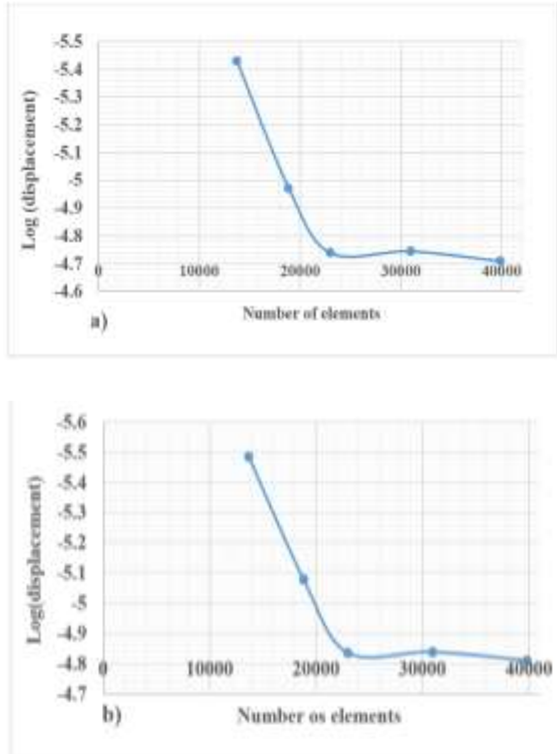

Fig. 5 Mesh sensitivity analysis at temperatures: (a): -33 degree Celsius and, and (b): +71 degree Celsius.

The deformation of the mechanism's base plate and the positioning of the axes were analysed at a temperature of -33°C, and the element displacements at this temperature are illustrated in “Fig. 6”. The positions of the axes are marked with red circles in the Figure.

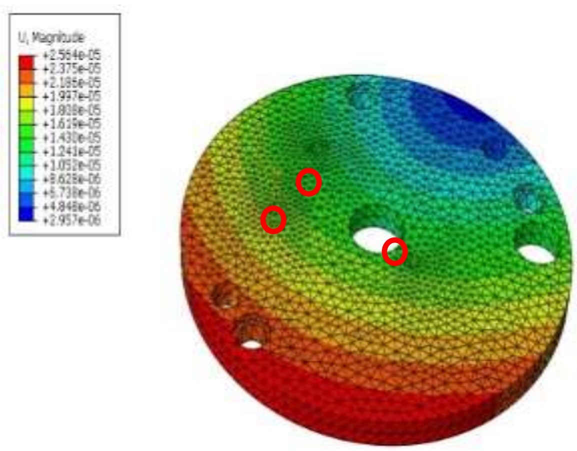

Fig. 6 Change in coordinates of axis positions on the below plate, axis positions at -33°C.

The analysis focused on the change in coordinates of the axis positions on the base plate of the clock mechanism, where the mechanism's axes are located. This analysis was conducted at a temperature of +71°C, and “Fig. 7” illustrates the displacements of the elements at this temperature.

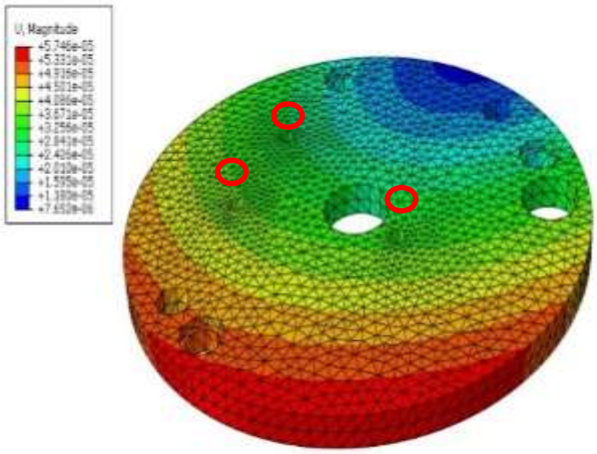

Fig. 7 Change in coordinates of axis positions on the below plate, axis positions at +71°C.

Table 4 displays the positions of the main axes at temperatures of -33 and +71°C. The results in "Table 4" indicate that the positions of the main axes fall within the manufacturing position tolerance specified by the designer, even at temperatures exceeding 100°C. This underscores the importance of conducting a thermal tolerance analysis.

Table 4 Changing the position of the axis of the delay clock in the working temperature range

Row		Dimension (mm)	change of position, mm	change of position, mm
T, °C		+21°C	-33 °C	+71°C
Position 1	X	-6.09	0.018	0.0151
	Y	-4.59		
Position 2	X	-8.15	0.0127	0.0121
	Y	1.48		
Position 3	X	3.50	0.0132	0.0110
	Y	0		

In the clock mechanism, there are 33 dimensions for all its components that require tolerance considerations during the design phase. By dividing each nominal dimension's tolerance range into ten subdivisions, approximately 10^{33} three-dimensional models need to be analysed to investigate geometric interference within both the upper and lower tolerance limits. However, the program has been developed to create only one SolidWorks geometry model using nominal dimensions. All the nominal dimensions, along with their upper and lower tolerance limits, have been inputted into the program through an Excel file. After implementing the program and conducting the analysis to prevent component interference, it was found that only in one base plate component, the axis position tolerance needs to be adjusted from the initial value of ±0.05 (before the geometric interference tolerance analysis) to ±0.03. This modification ensures that all components remain free from geometric interference at ambient temperature. However, considering that the axis positions undergo a maximum change of ±0.02 due to thermal strain, the designer must exclude the thermal tolerance from the geometric interference tolerance assessment. As a result, a tolerance of ±0.01 should be applied to the manufacturing drawings to ensure there are no interference issues during production.

6 CONCLUSIONS

In this research, for presenting the proposed method, coding has been carried out in a way that involves constructing only one geometric model in SolidWorks software. By providing nominal dimensions and upper and lower tolerance limits in an input file, tolerance analysis of the mechanism can be performed. To achieve this, the program can adapt the geometric model for multiple desired configurations, enabling the detection and adjustment of tolerance values that could cause component interference within the mechanism. As a result, during the assembly of the mechanism with different manufacturing tolerances for each component, the movement of various parts will be possible without any interference, ensuring the mechanism functions without any locking issues. In the current study, a precise and sensitive mechanical clock mechanism system was analyzed to showcase the effectiveness of the proposed tolerance analysis method, particularly under conditions of geometric interference. Given the requirement for precise functionality in this mechanism, any changes in the mechanism's shape due to variations in the operating temperature should not be disregarded during the design process. 24% of the examined sizes should be reduced and the tolerance for the clock axis positions in this mechanism has been modified to ±0.01. The tolerances have been carefully evaluated to ensure no component interference, and the tolerance for the displacement of the clock axis positions resulting from the operating temperature of the mechanism has also been calculated. The implemented program can be utilized for different types of mechanisms; it is not restricted to a specific one. Consequently, the proposed method is well-suited for precise mechanisms with a substantial number of involved components, demonstrating excellent performance.

REFERENCES

[1] Wu, F., Dantan, J. Y., Etienne, A., Siadat, A., and Martin P., Improved Algorithm for Tolerance Allocation Based on Monte Carlo Simulation and Discrete Optimization, Computers & Industrial Engineering, Vol. 56, No.4, 2009, pp. 1402-1413.

[2] Dantan, J. Y., Qureshi, A. J., Worst-Case and Statistical Tolerance Analysis Based on Quantified Constraint Satisfaction Problems and Monte Carlo Simulation, Computer-Aided Design, Vol. 41, No. 1, 2009, pp. 1-12.

[3] Sheikhi Azqandi, M., Hassanzadeh, M., first-and Second-Order Sensitivity Analysis of Finite Element Models using Extended Complex Variables Method, Archive of Applied Mechanics, Vol. 91, 2021, pp. 4263-4277.

[4] Sheikhi Azqandi, M., Hassanzadeh, M., and Arjmand, M., Calculation of Design Shape Sensitivity in Solid Mechanics Through a Novel Hybrid Method using CVM and DSM, Journal of Stress Analysis, Vol. 5, No. 1, 2020, pp. 11-20.

[5] Doltsinis, I., Kang, Z., Robust Design of Structures Using Optimization Methods, Computer Methods in Applied Mechanics and Engineering, Vol. 193, No. 23, 2004, pp. 2221-2237.

[6] Zang, C., Friswell, M., and Mottershead, J., A Review of Robust Optimal Design and Its Application in Dynamics, Computers and structures, Vol. 83, No. 4-5, 2005, pp. 315-326.

[7] Huang, X., Zhang, Y., Robust Tolerance Design for Function Generation Mechanisms with Joint Clearances, Mechanism and machine theory, Vol. 45, No. 9, 2010, pp. 1286-1297.

[8] Zhou, K., Cai, Y., Zhang Z. J., and LI, X. Y., Application of the Theory of Robust Design in Tolerance Analysis, Transactions of Beijing Institute of Technology, Vol. 23, 2003, pp. 557-560.

[9] Shao, N., Ding, X., Liu, J., Tolerance Analysis of Spur Gears Based on Skin Model and Boundary Element Method, Mechanism and Machine Theory. Vol. 144, 2020, pp. 103658.

[10] Geetha, K., Ravindran, D., Kumar, M. S., and Islam, M. N., Concurrent Tolerance Allocation and Scheduling for Complex Assemblies, Robotics and Computer-Integrated Manufacturing, Vol. 35, 2015, pp. 84-95.

[11] Shoukr, D. S. L., Gadallah, M. H., and Metwalli, S. M., The Reduced Tolerance Allocation Problem, ASME International Mechanical Engineering Congress and Exposition, Phoenix, IMECE2016-65848, V011T15A023, Arizona, USA, 2016.

[12] Khodaygan, S., Meta-Model Based Multi-Objective Optimization Method for Computer-Aided Tolerance Design of Compliant Assemblies, International Journal of Computer Integrated Manufacturing, Vol. 32, No. 1, 2018, pp. 27-42.

[13] Delos, V., Arroyave-Tobón, S., and Teissandier, D., Introducing a Projection-Based Method to Compare Three Approaches to Computing the Accumulation of Geometric Variations, In 38th International Design Engineering Technical Conferences and Computers and Information in Engineering Conference, Quebec City, Canada, 2018.

[14] Lin, E. E., Zhang H. C., Theoretical Tolerance Stack Up Analysis Based on Tolerance Zone Analysis, The International Journal of Advanced Manufacturing Technology, Vol. 17, No. 4, 2001, pp. 257-262.

[15] Wang, Y., Closed-loop analysis in semantic tolerance modeling, Journal of Mechanical Design, Vol. 130 No.6, pp. 061701-061711. 2008.

[16] Geis, A., Husung, S., Oberänder, A., Weber, C., and Adam, J., Use of Vectorial Tolerances for Direct Representation and Analysis in CAD-Systems, Procedia Cirp, Vol. 27, 2015, pp. 230-240.

[17] Heling, B., Aschenbrenner, A., Walter, M., and Wartzack, S., On Connected Tolerances in Statistical Tolerance-Cost-Optimization of Assemblies with Interrelated Dimension Chains, Procedia Cirp, Vol. 43, 2016, pp. 262-267.

[18] Kopardekar, P., Anand, S., Tolerance Allocation Using Neural Networks, The International Journal of Advanced Manufacturing Technology, Vol. 10, No. 4, 1995, pp. 269-276.

[19] Luo, C., Franciosa, P., Ceglarek, D., Ni, Z., and Jia, F., A Novel Geometric Tolerance Modeling Inspired by Parametric Space Envelope, IEEE Transactions on Automation Science and Engineering, Vol. 15, No. 3, 2018, pp. 1386-1398.

[20] Samper, S., Formosa, F., Form Defects Tolerancing by Natural Modes Analysis, Journal of Computing and Information Science in Engineering, Vol. 7, No. 1, 2007, pp. 44-51.

[21] Homri, L., Goka, E., Levasseur, G., and Dantan, J.-Y., Tolerance Analysis—Form Defects Modeling and Simulation by Modal Decomposition and Optimization, Computer-Aided Design, Vol. 91, 2017, pp. 46-59.

[22] Lin, E. E., Graph-Matrix-Based Automated Tolerance Analysis and Setup Planning in Computer-Aided Process Planning, Ph.D. Dissertation, Industrial Engineering, Texas Tech University, 2000.

[23] Zhang, K., Li, Y., and Tang, S., An Integrated Modeling Method of Unified Tolerance Representation for The Mechanical Product, The International Journal of Advanced Manufacturing Technology, Vol. 46, No. 1, 2010, pp. 217-226.

[24] Schleich, B., Wartzack, S., Anwer, N., and Mathieu, L., Skin Model Shapes: Offering New Potentials for Modeling Product Shape Variability, In 35th Computers and Information in Engineering Conference, Boston, Massachusetts, USA, 2015.

[25] Schleich, B., Anwer, N., Mathieu, L., and Wartzack, S., Skin Model Shapes: A New Paradigm Shifts for Geometric Variations Modeling in Mechanical Engineering, Computer-Aided Design, Vol. 50, 2014, pp. 1-15.

[26] Yacob, F., Semere, D., and Nordgren, E., Octree-Based Generation and Variation Analysis of Skin Model Shapes, Journal of Manufacturing and Materials Processing, Vol. 2, No. 3, 2018, pp. 52-73.

[27] Yan, X., Ballu, A., Generation of Consistent Skin Model Shape Based on FEA Method, The International Journal of Advanced Manufacturing Technology, Vol. 92, No. 1, 2017, pp. 789-802.

[28] Schleich, B., Anwer, N., Mathieu, L., and Wartzack, S., Contact and Mobility Simulation for Mechanical Assemblies Based on Skin Model Shapes, Journal of Computing and Information Science in Engineering, Vol. 15, No. 2, 2015, pp. 021009-021009-7.

[29] Dantan, J. Y., Comparison of Skin Model representations and Tooth Contact Analysis Techniques for Gear tolerance analysis, Journal of Computing and Information Science in Engineering, Vol. 15, No. 2, 2015, pp. 021009-7.

[30] Yan, X., Assembly Simulation and Evaluation Based on Generation of The Virtual Work Piece with Form Defect, Ph.D. Dissertation, Mechanical Engineering, Bordeaux, France, 2018.

[31] Faraji, F., Abbasi, K., Experimental Study of the Effect of Conventional Manufacturing Geometric Deviation in the Fitted Cylindrical Joint on its Strength, Karafan Quarterly Scientific Journal, Vol. 19, No. 1, 2022, pp. 107-123.

[32] Ghodsbin, J. A., Hatami, H., Numerical Behaviour Study of Expanded Metal Tube Absorbers and Effect of Cross Section Size and Multi-Layer Under Low Axial Velocity Impact Loading, Amirkabir Journal of Mechanical Engineering, Vol. 49, No. 4, 2018, pp. 245-248.

[33] Hatami, H., Fathollahi, A. B., The Theoretical and Numerical Comparison and Investigation of The Effect of Inertia on The Absorbent Collapse Behavior of Single Cell and Two-Cell Reticular Under Impact Loading, International Journal of Impact Engineering, Vol. 109, 2017, pp. 224-239.

[34] Hassanzadeh, M., Mazare, M., Computation of Design Sensitivities in Steady-State Incompressible Laminar Flows Based Oon New Semi-Analytical Method, Journal of Scientific Computing, Vol. 83, No. 12, 2020.

[35] Sheikhi Azqandi, M., Hassanzadeh, M., and Arjmand, M., Sensitivity Analysis Based on Complex Variables in FEM for Linear Structures, Advances in Computational Design, Vol. 4, No. 1, 2019, pp. 15-32.