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Design, Optimization, and Analysis of Machining Fixture Layout under Dynamic Conditions

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Abstract: This paper presents a study on the optimization of fixture layout design in multi-operation machining processes to improve the dimensional accuracy, surface finish, and productivity of manufacturing industries. The study focuses on the role of fixture elements such as locators and clamps in holding, securing, and restraining the workpiece during machining operations. The improper fixture layout can increase the vibration of the fixture-workpiece system, leading to reduced accuracy and surface finish of the machined workpiece. The study employs Finite Element Method (FEM) and modal analysis to compute the natural frequencies of the fixture-workpiece system and proposes the use of a Genetic Algorithm (GA) and Particle Swarm Optimization (PSO) to optimize the fixture layout design. ANSYS 15.0 and MATLAB software are used to conduct the FEM analysis and develop the GA and PSO coding. The study also presents the validation of the proposed approach on both 2D and 3D fixture-workpiece systems. The results show that PSO-based optimization produces better results and is identified as the suitable method for fixture layout optimization problems. The proposed approach can aid designers in minimizing machining vibration and achieving the required machining accuracy in multi-operation machining processes.
Keywords: Finite Element Method, EM, GA, PSO, Finite Element Analysis.

I. INTRODUCTION

This paper discusses the importance of fixtures in modern manufacturing industries to improve dimensional accuracy, surface finish, and productivity during the multi-operation machining process. Fixtures are work-holding devices used to reduce cycle time in automated manufacturing, inspection, and assembly operations. However, the design, fabrication, and testing of fixtures consume a major portion of new product development time. In a machining system, fixtures are designed to be flexible to reduce lead time, and fixture design and analysis software can assist in performing fixture planning, design, and verification functions in a short time. The paper explains the fundamentals of fixtures, including the different fixture elements such as locators, clamps, supports, and fixture body, which are used to hold, locate, and restrain the workpiece during the machining process. The stability of the workpiece during the machining process is achieved by placing the fixture elements appropriately in their location to constrain the workpiece. The paper also highlights the fixture requirements, which are necessary to fulfill several needs to retain the workpiece in a constant position during machining. The benefits of fixtures in manufacturing are explained, which include improved accuracy and uniform quality of components machined using fixtures, minimized labor cost, and reduced production cost by maximizing productivity and minimizing operating cost. The paper emphasizes the importance of fixture layout design, which is essential for machining the product and process designs while maintaining an optimal design for function and structural performance. A proper fixture layout design ensures the workpiece is positioned precisely in its position with respect to the cutting tool, improving machining accuracy and reducing the cost of quality control of the machined part. Finally, the paper discusses the need for Computer Aided Fixture Layout Design (CAFLD) to determine the optimal fixture layout for dynamic machining conditions. The CAFLD procedure can integrate the concepts in CAD and fixture layout design to find a feasible solution for complicated problems in fixture layout design. In conclusion, this paper provides a comprehensive overview of the fundamentals of fixtures and their importance in modern manufacturing industries.

II. LITERATURE REVIEW

This literature review focuses on the design and optimization of the machining fixture layout. The primary objective is to develop a frequency-based approach to minimize workpiece vibration during machining. The approach involves altering the natural frequency of the workpiece by placing fixture elements at various positions, making the difference between the natural frequency of the workpiece and the exciting frequency of the cutter to be maximum. This is achieved through modal analysis using FEM and integrating it with evolutionary techniques. The review highlights that most machining forces are dynamic, and the dynamic behavior of the fixture-workpiece system needs to be considered to design a fixture layout that can improve machining accuracy.

The research aims to overcome the difficulties identified in the research gap, and the sub-objectives are framed to achieve the primary objective of the research. The first sub-objective is to create a finite element model of the 2D and 3D fixture-workpiece system using ANSYS 15.0 finite element analysis software. The second sub-objective is to conduct preliminary experiments of finite element simulations to study the dynamic behavior of the fixture-workpiece system. The third sub-objective is to determine and record the first three natural frequencies for all load steps and exciting frequencies. The fourth sub-objective is to calculate the value of the objective function, which is the difference between the natural frequencies of the fixture-workpiece system and the exciting frequency of the cutter. The fifth sub-objective is to apply GA and PSO-based optimization algorithms to find the near-optimal position of the locator and clamps to maximize the objective function. The sixth sub-objective is to analyze and justify the values of control parameters used in GA and PSO. Finally, the seventh sub-objective is to compare the performance of GA and PSO for identifying the suitable algorithm for the optimization of the machining fixture layout under dynamic conditions. The review highlights that some researchers have calculated the natural frequencies for their analysis, but the difference between natural and exciting frequencies has not been appropriately explored for analyzing the vibrations. Therefore, the frequency-based approach proposed in this research work could fill this research gap and improve the machining accuracy.

The research proposes using ANSYS 15.0 finite element analysis software to create a finite element model of the 2D and 3D fixture workpiece system. This is suitable software for simulating the dynamic behavior of the fixture-workpiece system, as it can analyze both linear and nonlinear static and dynamic behavior. Moreover, the proposed approach involves integrating FEM with evolutionary techniques, which could lead to more accurate and reliable results than using FEM alone. The review also highlights that GA and PSO are two optimization algorithms that can be used to find the near-optimal position of the locator and clamps. Both algorithms have their advantages and disadvantages, and the selection of the suitable algorithm depends on the specific problem and the required performance criteria. In conclusion, this literature review highlights the importance of considering the dynamic behavior of the fixture-workpiece system to design and optimize the machining fixture layout. The proposed frequency-based approach, which involves modal analysis using FEM and integrating it with evolutionary techniques, could improve the machining accuracy by minimizing workpiece vibration during machining. The sub-objectives proposed in this research work could achieve the primary objective of the research, and the performance of GA and PSO could be compared to identify the suitable algorithm for the optimization of machining fixture layout under dynamic conditions.

III. FINITE ELEMENT METHOD

Finite Element Method (FEM) is a numerical technique used to solve complex engineering problems, which are otherwise difficult to solve analytically. It is based on dividing the problem domain into smaller subdomains, called elements, which are easier to analyze. FEM involves solving a set of algebraic equations, based on a mathematical model that represents the physical behavior of the system. Dynamic analysis using FEM is used to study the response of structures to time-varying loads.

In dynamic analysis, when loads are variable with respect to time, mass and acceleration come into effect. The response of a structure to dynamic loads is studied by determining its natural frequencies and mode shapes. When a structure is elastically deformed and released suddenly, it vibrates about its equilibrium position. This is called free vibration, and the frequency corresponding to free vibration is the natural frequency of the structure.

The FEM is based on shape functions that represent the displacement of the structure at each node. For example, in a four-node quadrilateral element, the shape function for each node is defined as a constant value at that node and zero at all other nodes. The constant values are determined by the condition that the shape function is equal to 1 at the node it represents. The displacement of any point within the element is then determined using the shape functions and the nodal displacements.

The stiffness matrix of an element represents the resistance of the element to deformation. It is determined based on the strain energy of the element. The strain energy is the work done on the element to deform it, and it is calculated based on the strain-displacement relations of the element. The mass matrix of an element represents the mass distribution within the element, and it is determined based on the shape functions of the element.

In hexahedral or brick elements, the connectivity of nodes is defined by a consistent numbering scheme, and the shape functions are represented by Lagrange shape functions. The mass and stiffness matrices are then determined based on the shape functions and strain-displacement relations of the element.

Dynamic analysis using FEM involves solving the equations of motion for the system under the given time-varying loads. The kinetic and potential energy terms of the system are determined, and the Lagrangian equation is used to solve for the displacements of the system over time. The response of the system is then studied using frequency and time domain analysis techniques.

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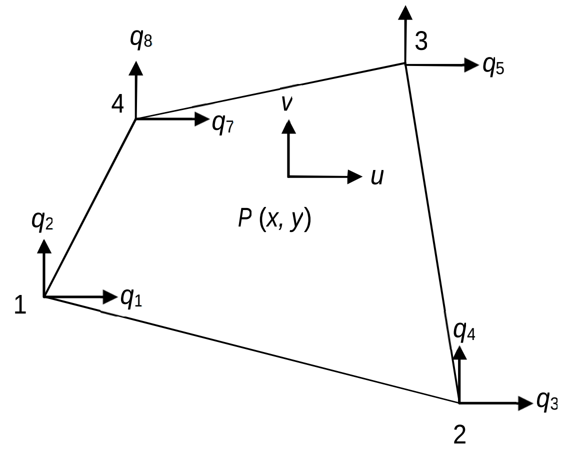

Figure 3.1 Four-node quadrilateral element

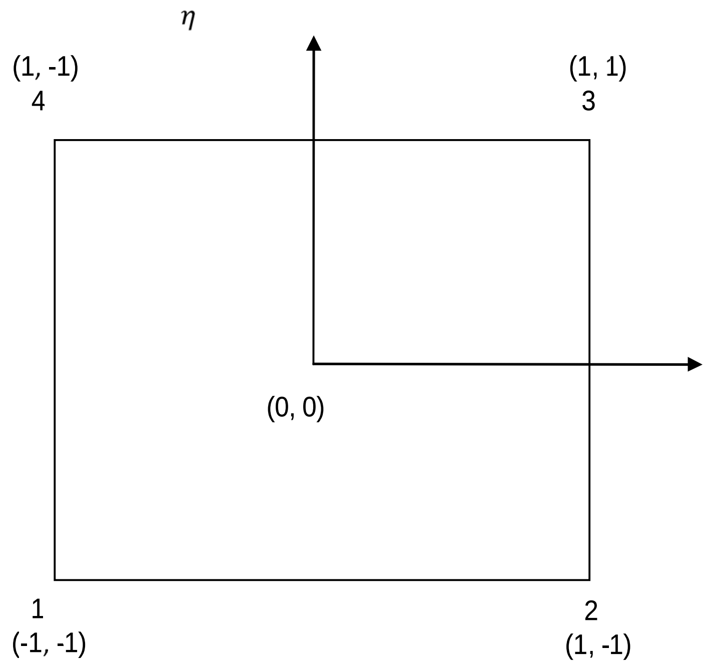

Figure 3.2 Parent element

The nodes are numbered in a counter-clockwise manner and the coordinates of node *i* are defined as (x_i, y_i) and the corresponding displacement vectors are

$$q = [q_1, q_2, q_3, \dots, q_8] \tag{3.1}$$

A. Shape Functions of Four –Node Quadrilateral Element

To get the shape functions, parent element as shown in Figure 3.2 is defined in natural coordinates. The shape function N_i is equal to unity at node i and is zero at other nodes where i = 1, 2, 3 and 4. Thus, the shape function for node 1 has to be of the form.

$$N_1 = c(1-\xi)(1-\eta) \tag{3.2}$$

where, c = Constant.

The constant is calculated from the condition $N_1 = 1$ at node 1 by substituting $\xi = -1, \eta = -1$ at node 1

$$1 = c(2)(2)$$

which gives $c = \frac{1}{4}$. Thus, the shape functions for all the four nodes can be written as

$$N_1 = \frac{1}{4}(1-\xi)(1-\eta) \tag{3.3}$$

$$N_2 = \frac{1}{4}(1+\xi)(1-\eta) \tag{3.4}$$

$$N_3 = \frac{1}{4}(1+\xi)(1+\eta) \tag{3.5}$$

$$N_4 = \frac{1}{4}(1-\xi)(1+\eta) \tag{3.6}$$

The displacement vector $u = [u, v]^T$ denotes the displacement components of a point located at (ξ, η).

$$u = N_1 q_1 + N_2 q_3 + N_3 q_5 + N_4 q_7 \tag{3.7}$$

$$v = N_1 q_2 + N_2 q_4 + N_3 q_6 + N_4 q_8 \tag{3.8}$$

In matrix form

$$u = Nq \tag{3.9}$$

where,

$$N = \begin{bmatrix} N_1 & 0 & N_2 & 0 & N_3 & 0 & N_4 & 0 \\ 0 & N_1 & 0 & N_2 & 0 & N_3 & 0 & N_4 \end{bmatrix} \tag{3.10}$$

Furthermore, the same shape functions are used to describe the coordinates of a point within the element in terms of nodal coordinates.

$$x = N_1 x_1 + N_2 x_2 + N_3 x_3 + N_4 x_4 \tag{3.11}$$

$$y = N_1 y_1 + N_2 y_2 + N_3 y_3 + N_4 y_4 \tag{3.12}$$

A function f = f(x,y) considered to be an implicit function of ξ and η as

$$f = f[x(\xi, \eta), y(\xi, \eta)] \tag{3.13}$$

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While using chain rule of differentiation

$$\frac{\partial f}{\partial \xi} = \frac{\partial f}{\partial x}\frac{\partial x}{\partial \xi} + \frac{\partial f}{\partial y}\frac{\partial y}{\partial \xi} \tag{3.14}$$

$$\frac{\partial f}{\partial \eta} = \frac{\partial f}{\partial x}\frac{\partial x}{\partial \eta} + \frac{\partial f}{\partial y}\frac{\partial y}{\partial \eta} \tag{3.15}$$

or

$$\begin{Bmatrix} \frac{\partial f}{\partial \xi} \\ \frac{\partial f}{\partial \eta} \end{Bmatrix} = \mathbf{J} \begin{Bmatrix} \frac{\partial f}{\partial x} \\ \frac{\partial f}{\partial y} \end{Bmatrix} \tag{3.16}$$

where, J = Jacobian matrix

$$J = \begin{bmatrix} \frac{\partial f}{\partial \xi} & \frac{\partial y}{\partial \xi} \\ \frac{\partial f}{\partial \eta} & \frac{\partial y}{\partial \eta} \end{bmatrix} \tag{3.17}$$

$$J = \frac{1}{4}\begin{bmatrix} -(1-\eta)x_1 + (1-\eta)x_2 + (1+\eta)x_3 - (1+\eta)x_4 & -(1-\eta)y_1 + (1-\eta)y_2 + (1+\eta)y_3 - (1+\eta)y_4 \\ -(1-\xi)x_1 - (1+\xi)x_2 + (1+\xi)x_3 + (1-\xi)x_4 & -(1-\xi)y_1 - (1+\xi)y_2 + (1+\xi)y_3 - (1-\xi)y_4 \end{bmatrix} \tag{3.18}$$

$$J = \begin{bmatrix} J_{11} & J_{12} \\ J_{21} & J_{22} \end{bmatrix} \tag{3.19}$$

$$\begin{Bmatrix} \frac{\partial f}{\partial x} \\ \frac{\partial f}{\partial y} \end{Bmatrix} = J^{-1} \begin{Bmatrix} \frac{\partial f}{\partial \xi} \\ \frac{\partial f}{\partial \eta} \end{Bmatrix} \tag{3.20}$$

$$\begin{Bmatrix} \frac{\partial f}{\partial x} \\ \frac{\partial f}{\partial y} \end{Bmatrix} = \frac{1}{\det J}\begin{bmatrix} J_{22} & -J_{12} \\ -J_{21} & J_{11} \end{bmatrix} \begin{Bmatrix} \frac{\partial f}{\partial \xi} \\ \frac{\partial f}{\partial \eta} \end{Bmatrix} \tag{3.21}$$

$$dx\, dy = \det J\, d\xi\, d\eta \tag{3.22}$$

B. *Stiffness Matrix of Four-Node Quadrilateral Element*

Strain energy can be used to find the stiffness matrix for the fournode quadrilateral element, given as

$$U = \int_V \frac{1}{2}\sigma^T \epsilon\, dV \tag{3.23}$$

$$U = \sum_e t_e \int_e \frac{1}{2}\sigma^T \epsilon\, dA \tag{3.24}$$

where, t_e= thickness of the element

The strain –displacement relations are

$$\begin{Bmatrix} \epsilon_x \\ \epsilon_y \\ \gamma_{xy} \end{Bmatrix} \epsilon = \begin{Bmatrix} \frac{\partial u}{\partial x} \\ \frac{\partial v}{\partial y} \\ \frac{\partial v}{\partial x} + \frac{\partial u}{\partial y} \end{Bmatrix} \tag{3.25}$$

By $\begin{Bmatrix} \frac{\partial u}{\partial x} \\ \frac{\partial u}{\partial y} \end{Bmatrix} = \frac{1}{\det J} \begin{bmatrix} J_{22} & -J_{12} \\ -J_{21} & J_{11} \end{bmatrix} \begin{Bmatrix} \frac{\partial u}{\partial \xi} \\ \frac{\partial u}{\partial \eta} \end{Bmatrix}$ considering $f \equiv u$

Equation $\begin{Bmatrix} \frac{\partial v}{\partial x} \\ \frac{\partial v}{\partial y} \end{Bmatrix} = \frac{1}{\det J} \begin{bmatrix} J_{22} & -J_{12} \\ -J_{21} & J_{11} \end{bmatrix} \begin{Bmatrix} \frac{\partial v}{\partial \xi} \\ \frac{\partial v}{\partial \eta} \end{Bmatrix}$ obtained from above 3.26,3.27,3.28

where, A is given as

$$\epsilon = A \begin{Bmatrix} \frac{\partial u}{\partial \xi} \\ \frac{\partial u}{\partial \eta} \\ \frac{\partial v}{\partial \xi} \\ \frac{\partial v}{\partial \eta} \end{Bmatrix} \tag{3.29}$$

From the interpolation equations

$$\begin{Bmatrix} \frac{\partial u}{\partial \xi} \\ \frac{\partial u}{\partial \eta} \\ \frac{\partial v}{\partial \xi} \\ \frac{\partial v}{\partial \eta} \end{Bmatrix} = Gq \tag{3.30}$$

where,

$$G = \frac{1}{4} \begin{bmatrix} -(1-\eta) & 0 & (1-\eta) & 0 & (1+\eta) & 0 & -(1+\eta) & 0 \\ -(1-\xi) & 0 & -(1+\xi) & 0 & (1+\xi) & 0 & (1-\xi) & 0 \\ 0 & -(1-\eta) & 0 & (1-\eta) & 0 & (1+\eta) & 0 & -(1+\eta) \\ 0 & -(1-\xi) & 0 & -(1+\xi) & 0 & (1+\xi) & 0 & (1-\xi) \end{bmatrix} \tag{3.31}$$

$$\epsilon = Bq \tag{3.32}$$

where,

$$B = AG \tag{3.33}$$

The stress

$$\sigma = DBq \tag{3.34}$$

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The strain energy becomes

$$U = \sum_e \frac{1}{2} q^T \left[t_e \int_{-1}^{1} \int_{-1}^{1} B^T D B \det J \, d\xi d\eta \right] q \tag{3.35}$$

$$= \sum_e \frac{1}{2} q^T k^e \, q \tag{3.36}$$

where,

$$k^e = t_e \int_{-1}^{1} \int_{-1}^{1} B^T D B \det J \, d\xi d\eta \tag{3.37}$$

The nodes in hexahedral elements shown in Figure 3.3 is defined by a consistent node numbering scheme.

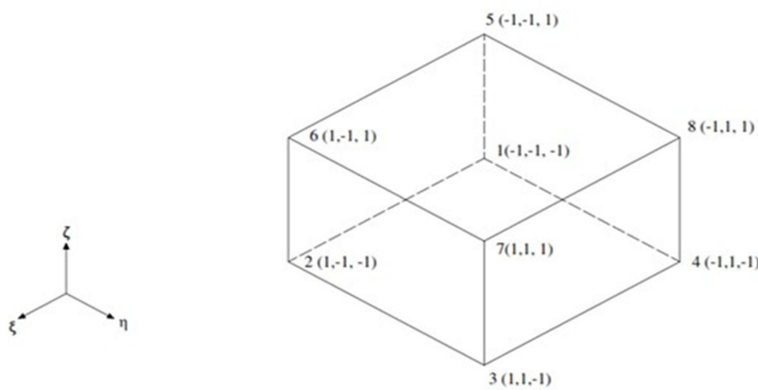

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Figure 3.3 Hexahedral element

The standard representation of Lagrange shape functions of the parent element

$$N_i = \frac{1}{8}(1 + \xi_i \xi)(1 + \eta_i \eta)(1 + \zeta_i \zeta) \tag{3.38}$$

where, i = 1 to 8

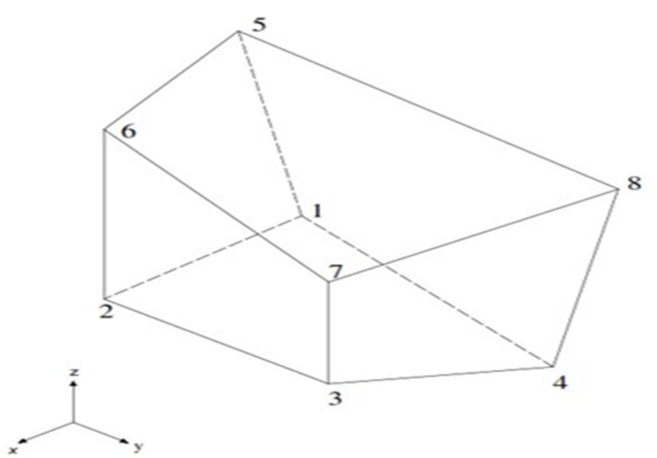

Figure 3.4 Master element

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The displacements of the nodes are represented by the displacement vector.

$$q = [q_{1,}\ q_{2,}\ q_{3,}\ \ldots.,\ q_{24}] \qquad (3.39)$$

Shape functions N_i are used to determine the displacement at any point inside the element.

$$u = N_1 q_1 + N_2 q_4 + \cdots + N_8 q_{22} \qquad (3.40)$$

$$v = N_1 q_2 + N_2 q_5 + \cdots + N_8 q_{23} \qquad (3.41)$$

$$w = N_1 q_3 + N_2 q_6 + \cdots + N_8 q_{24} \qquad (3.42)$$

and

$$x = N_1 x_1 + N_2 x_2 + \cdots + N_8 x_8 \qquad (3.43)$$

$$y = N_1 y_1 + N_2 y_2 + \cdots + N_8 y_8 \qquad (3.44)$$

$$z = N_1 z_1 + N_2 z_2 + \cdots + N_8 z_8 \qquad (3.45)$$

The strain is in the form of

$$\epsilon = Bq \qquad (3.46)$$

The element stiffness matrix is given as

$$k^e = \int_{-1}^{1}\int_{-1}^{1}\int_{-1}^{1} B^T DB \det J \, d\xi d\eta \, d\zeta \qquad (3.47)$$

The mass matrix of four-node quadrilateral element is evaluated from shape functions

$$u^t = [u \quad v] \qquad (3.48)$$

$$q^t = [q_{1,} \quad q_2 \quad \cdots \quad q_8] \qquad (3.49)$$

$$N = \begin{bmatrix} N_1 & 0 & N_2 & 0 & N_3 & 0 & N_4 & 0 \\ 0 & N_1 & 0 & N_2 & 0 & N_3 & 0 & N_4 \end{bmatrix} \qquad (3.50)$$

The mass matrix is given by

$$m^e = \rho t_e \int_{-1}^{1}\int_{-1}^{1} N^T N det\, J \, d\xi \, d\eta \qquad (3.51)$$

The mass matrix of the hexahedral element is evaluated from shape functions

$$m^e = \rho \int_{-1}^{1}\int_{-1}^{1}\int_{-1}^{1} N^T N det\, J \, d\xi \, d\eta \, d\, \zeta \qquad (3.52)$$

In structures, when analysis is done while slowly applying the loads, it is called static analysis. Furthermore, when the loads are variable with respect to certain period of time, the mass and acceleration come into effect. Solid body when elastically deformed and released suddenly, vibrates about its equilibrium position. This is known as free vibration. The frequency that corresponds to free vibration is called natural frequency and the highest displacement from its equilibrium position is called amplitude of vibration. The vibration of a structure is minimized using dampers. In the simplest vibration model, damping effects are neglected and give significant information about the dynamic behavior of the structure.

The Lagrangean equation is given by

$L = T - \Pi$ (3.53) where, T = Kinetic energy

Π = Potential energy

The kinetic energy is given as

$$T = \frac{1}{2}\int_V \dot{u}^T \dot{u} \rho \, dV \tag{3.54}$$

where, ρ = density

The velocity vector of the point at x, with components $\dot{u}, \dot{v}, \dot{w}$ is given as

$$\dot{u} = [\dot{u}, \dot{v}, \dot{w}]^T \tag{3.55}$$

In FEM, u is expressed in terms of nodal displacements q and shape functions N.

$$u = Nq \tag{3.56}$$

In dynamic analysis, the nodal displacements q are time dependent. The velocity vector is given by

$$\dot{u} = N\dot{q} \tag{3.57}$$

The kinetic energy T_e in the element e is

$$T_e = \frac{1}{2} \dot{q}^T \left[\int_e \rho N^T N dV\right] \dot{q} \tag{3.58}$$

The element mass matrix

$$m^e = \int_e \rho N^T N dV \tag{3.59}$$

$$T = \sum_e T_e = \sum_e \frac{1}{2} \dot{q}^T m^e \dot{q} = \frac{1}{2} \dot{Q}^T M \dot{Q} \tag{3.60}$$

The total potential energy is given as

$$\Pi = \frac{1}{2} Q^T K Q - Q^T F \tag{3.61}$$

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The equation of motion is obtained using Lagrangean equation

$$M\ddot{Q} + KQ = F \quad (3.62)$$

For free vibrations, force F is zero

$$M\ddot{Q} + KQ = 0 \quad (3.63)$$

For steady state condition

$$Q = Usin\omega t \quad (3.64)$$

where, U = amplitude of vibration
ω = circular frequency $= 2\pi f$

$$KU = \omega^2 MU \quad (3.65)$$

The generalized eigenvalue problem

$$KU = \lambda MU \quad (3.66)$$

where, U = Eigen vector
$\lambda =$ Eigenvalue

In conclusion, FEM is a powerful numerical technique used to solve complex engineering problems, including dynamic analysis of structures under variable loads. It involves dividing the problem domain into smaller elements, determining the shape functions, and solving for the algebraic equations of motion. The stiffness and mass matrices are determined based on the strain-displacement relations and shape functions of the element.

IV. FINITE ELEMENT MODELING AND ANALYSIS OF 2D and 3D FIXTURE–WORKPIECE SYSTEM

Researchers commonly use Finite Element Method (FEM) to analyze fixture-workpiece systems and compute workpiece responses during machining operations. FEM employs an approximate function to find solutions, and the accuracy of the solutions depends on valid assumptions and proper mesh size selection. This research study utilizes the FEM tool ANSYS 15.0 to investigate the dynamic response of a workpiece during end milling operation on a 2D rectangular geometry. The analysis assumes a plane stress condition for two-dimensional problems, with the workpiece as an elastic body and the fixture elements as rigid bodies. The study also explores fixture locating principles, which aim to achieve high-precision fixture designs that minimize dimensional and machining accuracy errors and reduce workpiece vibrations. FEA can resolve these issues and determine the dynamic behavior of fixture-workpiece systems.

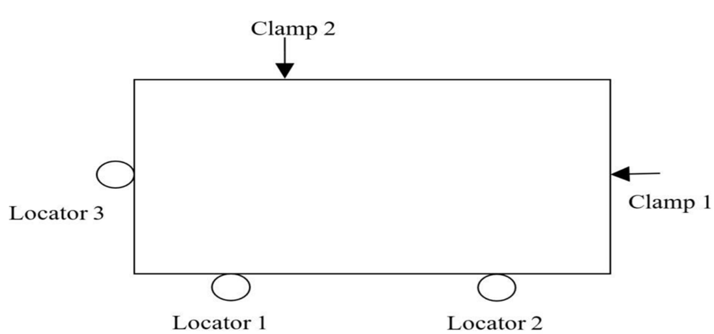


Figure 4.1 3-2-1 Fixture layout of 2D workpiece

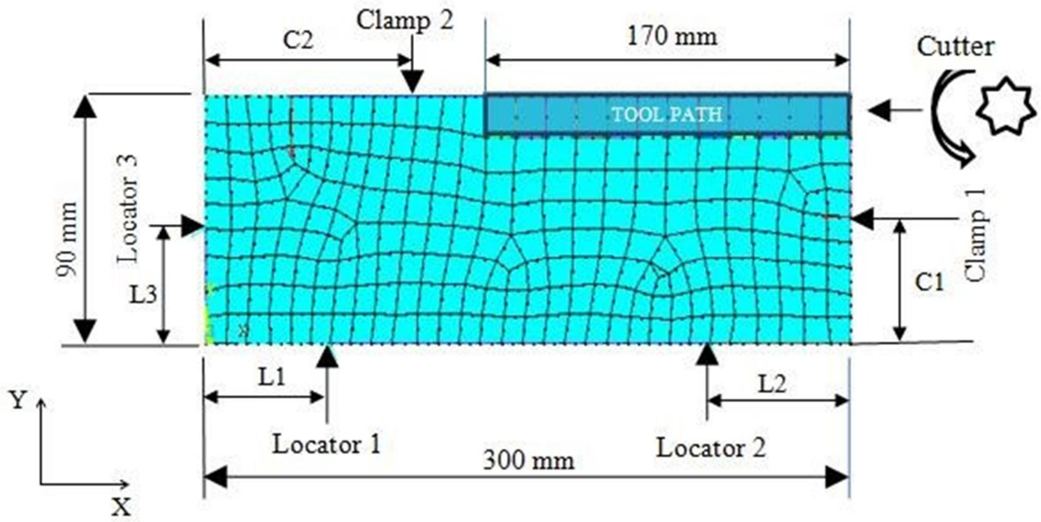


Figure 4.2 Finite element model of the 2D fixture - workpiece system

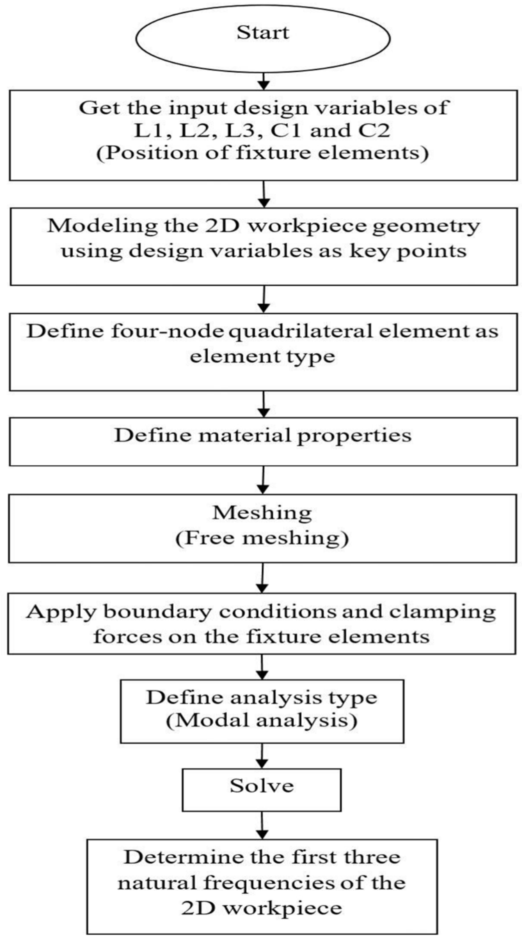
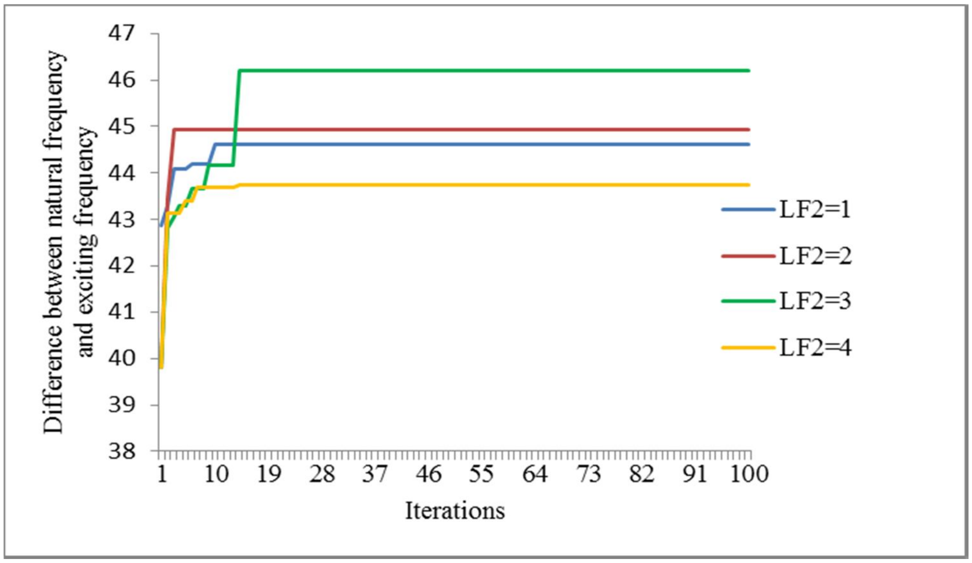
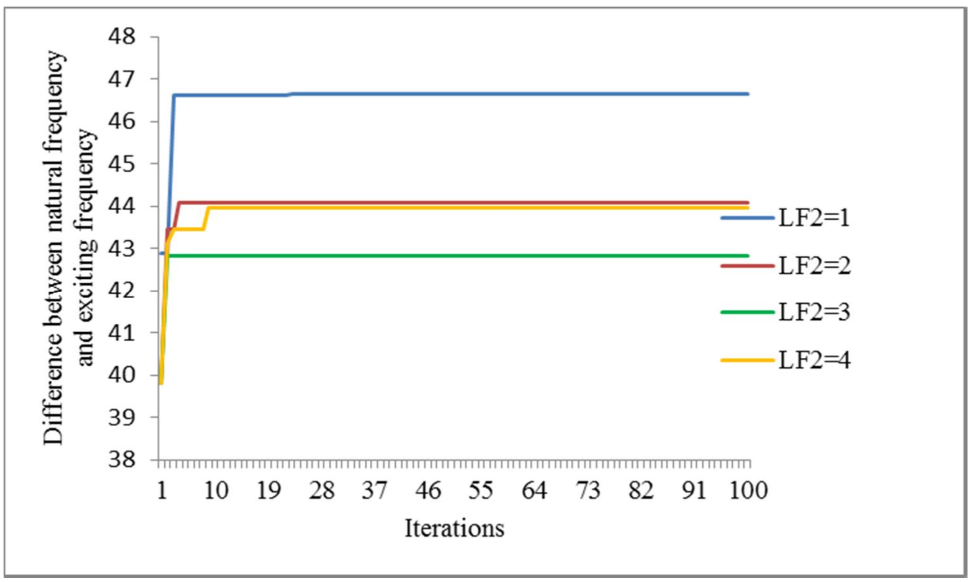


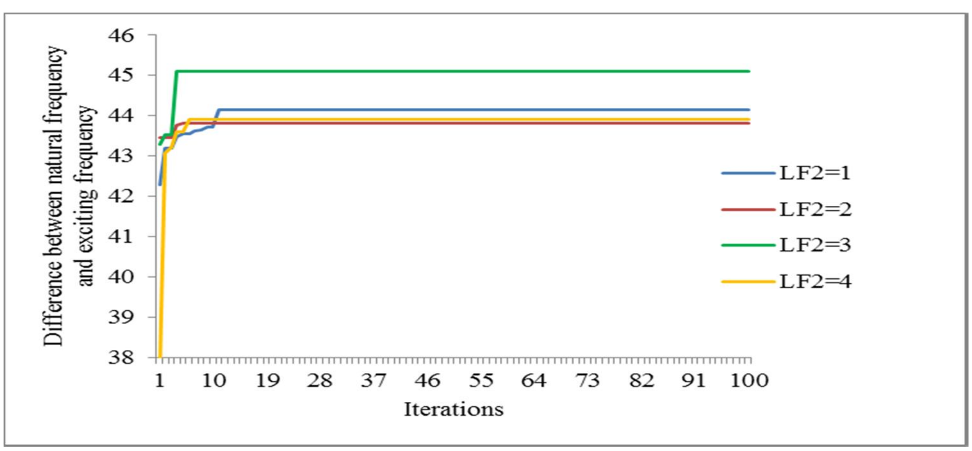
Figure 4.3 Flowchart for FEM of 2D fixture-workpiece system


Difference between natural frequency and exciting frequency
Iterations
LF2=1
LF2=2
LF2=3
LF2=4

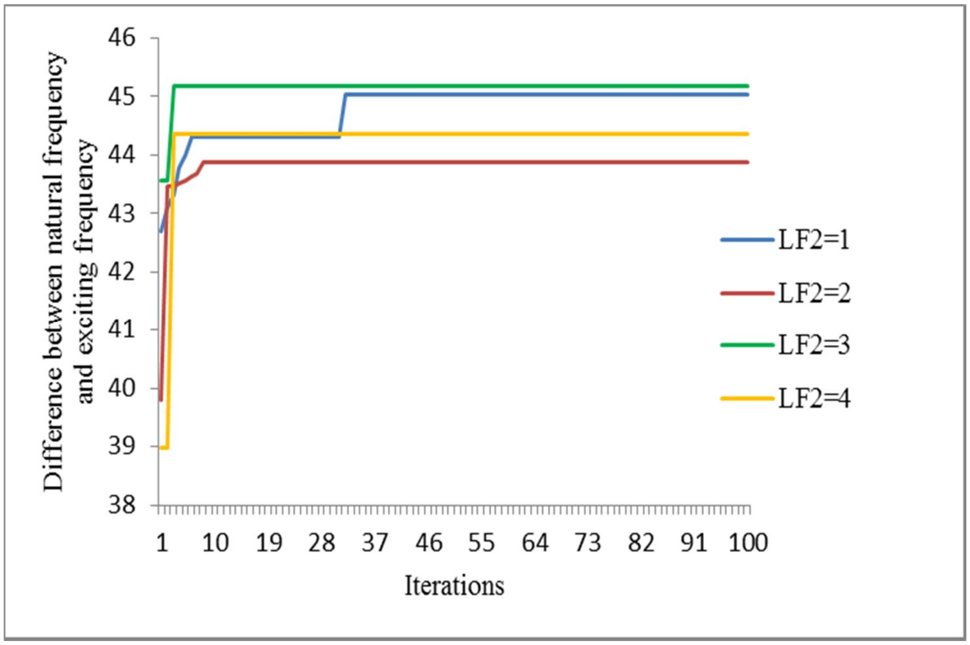
(a)


Difference between natural frequency and exciting frequency
Iterations
LF2=1
LF2=2
LF2=3
LF2=4

(b)


Difference between natural frequency and exciting frequency
Iterations
LF2=1
LF2=2
LF2=3
LF2=4

(c)



(d)

Figure 7.2 Effect of different values of LF1 and LF2 on convergence of PSO for: (a) LF1=1, (b) LF1=2, (c) LF1=3 and (d) LF1=4

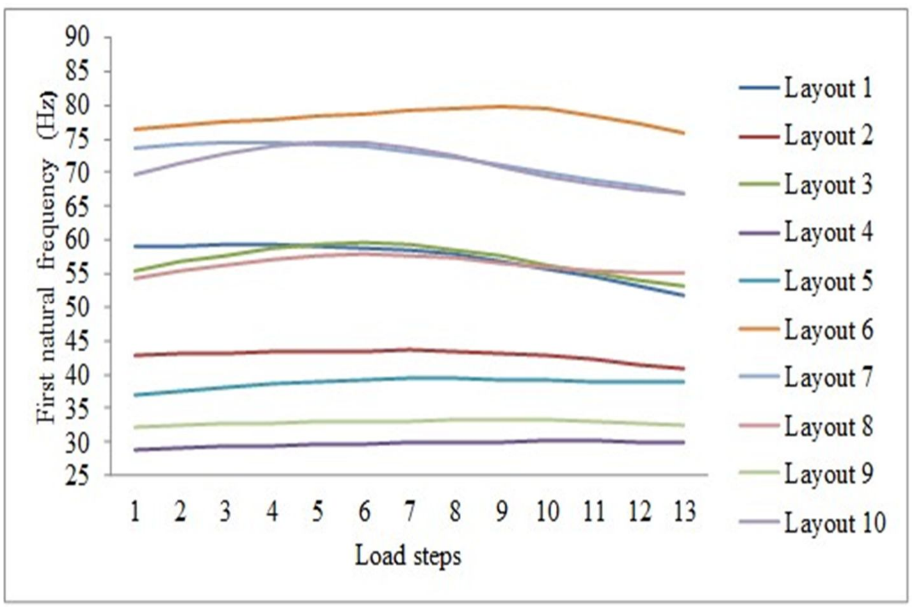


Figure 4.4 Effect of material removal on the natural frequency

The research work discussed the use of FEM for dynamic analysis of the fixture-workpiece system during the end milling operation. The study used ANSYS 15.0 to investigate the dynamic response of the workpiece and assumed a plane stress condition for the analysis. The design of the fixture must minimize dimensional and machining accuracy errors while reducing the vibration of the workpiece. The first natural frequency of the workpiece was used to formulate the objective function to minimize vibration and resulting dimensional errors. The effect of material removal on natural frequency was studied, and the appropriate position for locators and clamps was determined to optimize the objective function. The chapter concluded with an overview of the study's finite element modeling and analysis using ANSYS 15.0 and MATLAB.

V. CONCLUSION

This research work focuses on minimizing machining errors and improving product quality by minimizing the vibration of the workpiece through an accurately designed fixture layout. The proposed frequency-based methodology optimizes the machining fixture layout to minimize workpiece vibration during the machining process. It combines FEM with evolutionary techniques of GA and PSO and has been implemented on a 2D fixture-workpiece system and a 3D fixture-workpiece system. The methodology has been validated using ANSYS 15.0 software, and MATLAB has been used to run the optimization algorithms. Modal analysis has been performed to determine the natural frequencies of the workpiece. The influence of fixture layout on the dynamic behavior of the workpiece during end milling operation is studied, and the effect of material removal is analyzed. The results show that PSO produces better optimal results than GA for both 2D and 3D fixture-workpiece systems. The difference between natural frequency of the fixture-workpiece system and exciting frequency of the cutter in end milling operation has been maximized, and the near optimal results obtained by GA and PSO are compared. The results suggest that PSO is more suitable for fixture layout optimization problems than GA.

The major contributions of this research work are the development of a frequency-based approach to minimize workpiece vibration, the validation of the methodology on 2D and 3D fixture workpiece systems, and the comparison of GA and PSO for fixture layout optimization. This research work demonstrates the importance of accurate fixture design in reducing machining errors and improving product quality. The integration of FEM with the evolutionary techniques of GA and PSO provides a powerful tool for optimizing the fixture layout and reducing workpiece vibration. The results of this research work can be used to guide the design of machining fixtures and improve machining accuracy and quality.

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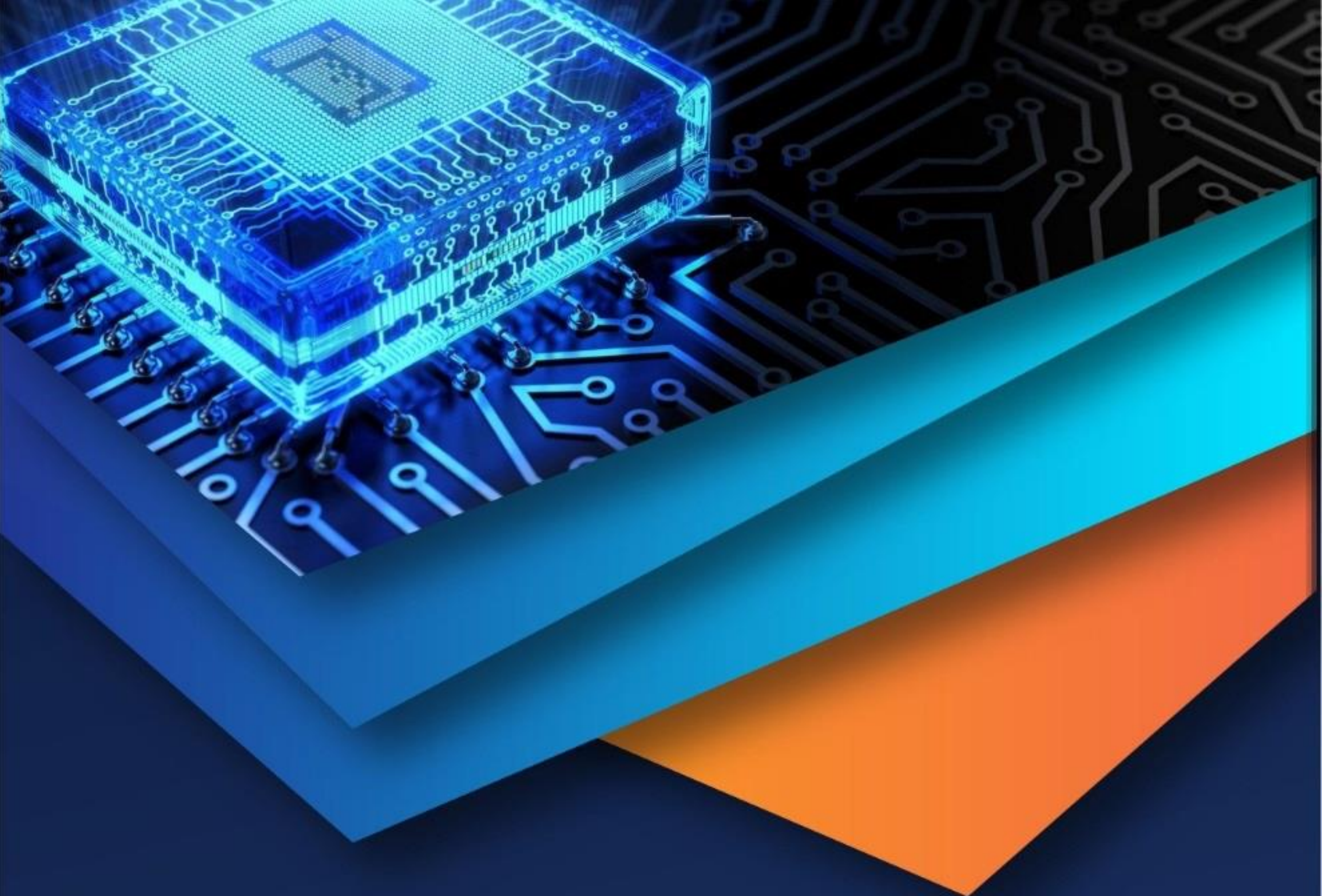

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