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ISSN 3030-3907

MATHEMATICAL MODELING OF STRUCTURAL STIFFNESS AND DYNAMIC BEHAVIOR OF A PNEUMATIC VISE FIXTURE USING THE COHERENCE FUNCTION.

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Annotatsiya. Ushbu maqolada frezalash jarayonida yupqa devorli korpus detallarini mahkamlash uchun ishlatiladigan pnevmatik iskanja moslamasining konstruktiv bikrligi va dinamik xususiyatlarini matematik modellash, shuningdek kogerentlik funksiyasi asosida tahlil qilish masalalari yoritilgan. Tizim 1-DOF va 2-DOF dinamik model sifatida ko‘rib chiqilgan bo‘lib, dempferlash elementi tizimning umumiy dempferlash koeffitsientiga ta‘siri hisobga olingan. Ekvivalent bikrlik, pnevmatik siqish kuchi, Hertz kontakt bikrligi va chastota uzatish funksiyasi (FRF) modellari MATLAB muhitida raqamli modellashtirish orqali aniqlangan. ANSYS/SolidWorks Simulation dasturida o‘tkazilgan chekli elementlar tahlili (FEA) natijalarida von Mises kuchlanganligi 350 MPa gacha yetadigan zo‘riqish konsentratsiyasi zonalari aniqlandi.

Kalit so‘zlar: pnevmatik iskanja; konstruktiv bikrlik; kontakt qattiqligi; siqish kuchi; kogerentlik funksiyasi; FRF; FEA; yupqa devorli korpus; MATLAB.

Аннотация. В статье рассматриваются вопросы математического моделирования и анализа конструктивной жёсткости и динамических характеристик пневматического зажимного приспособления. Система представлена как 1-DOF и 2-DOF динамическая модель. Моделирование выполнено в MATLAB. FEA выявил зоны концентрации напряжений (von Mises до 350 МПа). Предложены рекомендации по повышению жёсткости и гашению вибраций.

Ключевые слова: пневматическое зажимное приспособление; конструктивная жёсткость; функция когерентности; FRF; FEA; MATLAB.

Abstract. This paper investigates the mathematical modeling and analysis of the constructive stiffness and dynamic characteristics of a pneumatic fixture used in the milling of thin-walled housing components, with application of the coherence function. The system is modelled as a 1-DOF and 2-DOF dynamic model incorporating a tuned mass damper (TMD). Equivalent stiffness, pneumatic clamping force, Hertzian contact stiffness, and FRF models are validated through MATLAB simulation. FEA in ANSYS reveals von Mises stress concentrations

ISSN 3030-3907

up to 350 MPa near pneumatic fittings. Design recommendations for stiffness improvement and chatter suppression are proposed.

Key words: pneumatic fixture; constructive stiffness; contact stiffness; clamping force; coherence function; FRF; FEA; thin-walled housing; MATLAB.

INTRODUCTION.

In modern manufacturing, the accuracy and stability of the machining process are critically dependent on the stiffness and dynamic behaviour of the fixture–workpiece–machine tool system. In the milling of thin-walled housing components, vibrations known as chatter lead to poor surface finish, rapid tool wear, and dimensional inaccuracies. These vibrations arise from the elastic compliance of the fixture, fluctuating cutting forces, and regenerative mechanisms.

Pneumatic fixtures offer rapid clamping and ease of automation, but their constructive stiffness is the limiting factor governing machining precision when jaw overhang h_0 = 300 mm. A quantitative stiffness model is therefore essential for fixture design.

To analyse vibrations and evaluate fixture effectiveness, the coherence function is employed. It identifies the frequency-dependent causal relationship between the cutting force (input) and fixture vibration (output), distinguishing regenerative chatter from background noise. This paper develops a unified mathematical–dynamic model integrating constructive stiffness, TMD dynamics, FRF measurement, and FEA stress analysis into a single validated framework.

LITERATURE REVIEW AND PROBLEM STATEMENT

Tobias and Tlusty established that chatter operates as a time-delay dynamic system driven by a regenerative mechanism, with stability depending on system stiffness, damping, and cutting parameters [1, 2]. Structural methods to mitigate vibrations include TMDs and spring–rod damping elements [3, 4]. Pneumatic fixture stiffness is governed by a series combination of housing, jaw, guide, and contact stiffnesses [5].

Most existing studies focus on amplitude and frequency spectra, leaving the causal origin of vibration signals insufficiently analysed. The coherence function addresses this gap by quantifying the frequency-dependent input–output relationship [6]. Its systematic application to pneumatic fixture modelling remains a significant research gap that this study addresses.

RESEARCH METHODOLOGY

The methodology combines analytical modelling, MATLAB numerical computation (dde23 solver for time-delay equations), FEA simulation, and signal processing. Three complementary analyses are performed:

—(a) Time domain — maximum amplitude Amax and RMS value:

ISSN 3030-3907

$A_{RMS} = \sqrt{(1/T \int_0^T x^2(t)\, dt)}$ (1)

—(b) Frequency domain — spectral content via Discrete Fourier Transform:

$X[k] = \Sigma_{n=0}^{N-1} x[n] \cdot e^{-j2\pi kn/N}$ (2)

—(c) Causal analysis — coherence function between cutting force Fc and vibration x:

$$\gamma^2_{Fcx}(f) = | G_{Fcx}(f) |^2 / [G_{FcFc}(f) \cdot G_{xx}(f)], \quad 0 \le \gamma^2 \le 1 \qquad (3)$$

$G_{FcFc}(f)$ — autospectral density of the cutting force

$G_{xx}(f)$ — autospectral density of the vibration signal

$G_{Fcx}(f)$ — cross-spectral density; [object Object] — frequency

$\gamma^2(f) \approx 1$ indicates vibrations are deterministically driven by the cutting process; lower values indicate random noise or nonlinearity. Identification is performed only in frequency bands where $\gamma^2(f) \ge 0.80$.

MATHEMATICAL MODEL

The constructive stiffness of the pneumatic fixture is the resistance of its components to deformation under external load:

$$k = F / \delta \qquad (4)$$

The equivalent series stiffness of the fixture assembly:

$$1/k_{eq} = 1/k_h + 1/k_j + 1/k_g + 1/k_m + 1/k_c \qquad (5)$$

For permissible displacement $\delta max \le 0.01$ mm and cutting force $F_k = 1200$ N:

$$k_{eq} \ge F_c / \delta_{max} = 1200 / 0.01 = 120\,000 \text{ N/mm} \qquad (6)$$

The clamping force produced by the pneumatic cylinder:

$$F_{cl} = p \cdot A \cdot \eta, \quad A = \pi D^2 / 4 \qquad (7)$$

For D = 50 mm and $\eta = 0.9$: A = 1963.5 mm². Results are shown in Table 1 and Figure 1.

Table 1.

Clamping force Fcl vs. operating pressure (D = 50 mm, η = 0.9)

p (MPa)	A (mm²)	F_{al} (N)	F_{al} (kN)
1.0	1963.5	1767	1.77
1.5	1963.5	2651	2.65
2.0	1963.5	3534	3.53

ISSN 3030-3907

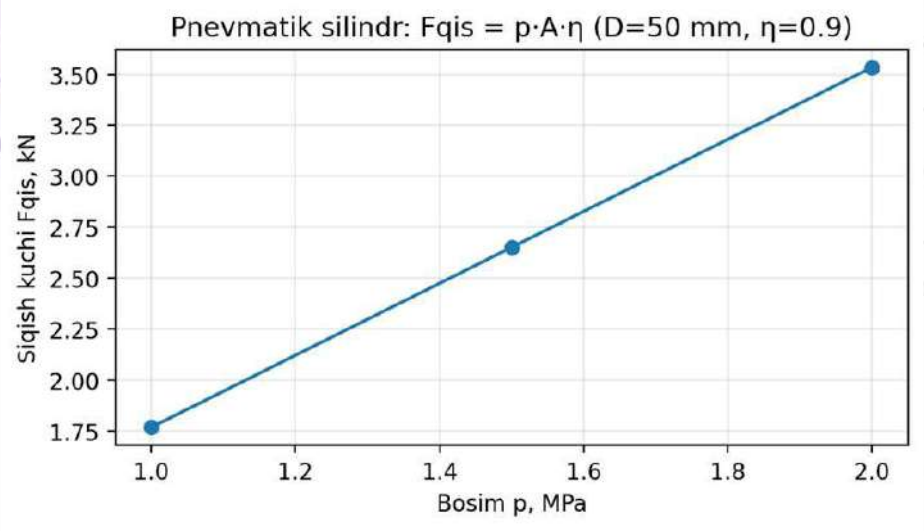


Figure 1. Pneumatic clamping force F_{al} as a function of operating pressure p (D = 50 mm, η = 0.9). Linear relation $F_{al} = p \cdot A \cdot \eta$ over 1.0–2.0 MPa yields 1.77–3.53 kN.

Jaw Overhang Effect

With jaw overhang h_o = 300 mm, the cantilever deflection scales as:

$$\delta_j \propto L^3 \qquad (8)$$

Reducing L from 300 mm to 150 mm decreases jaw deflection by a factor of $2^3 = 8$.

Hertzian Contact Stiffness

$$k_c = (4/3) \cdot E^* \cdot \sqrt{R} \cdot \delta^{1/2}, \quad 1/E^* = (1-\nu_1^2)/E_1 + (1-\nu_2^2)/E_2 \qquad (9)$$

1-DOF Dynamic Model

The fixture–workpiece system as a damped 1-DOF oscillator with TMD element damping contributions cd and kd:

$$m \cdot \ddot{x}(t) + (c + c_d) \cdot \dot{x}(t) + (k + k_d) \cdot x(t) = F_c(t) \qquad (10)$$

m — equivalent mass of the fixture–workpiece subsystem, kg

c, c_d — baseline and added damping coefficients, N·s/mm

k, k_d — baseline and added stiffness (k = k_ac from Eq. 5), N/mm

Frequency-response function (FRF):

$$H(\omega) = X(\omega) / F(\omega) = 1 / (k - m\omega^2 + jc\omega) \qquad (11)$$

Natural frequency and damping ratio:

$$\omega_n = \sqrt{(k/m)}, \quad \zeta = c / (2\sqrt{(km)}) \qquad (12)$$

Regenerative Cutting Force (Time-Delay Model)

The regenerative cutting force as a function of current and delayed displacement:

$$F_c(t) = K_c \cdot [\, x(t) - x(t - \tau) \,], \quad \tau = 60/n \qquad (13)$$

K_c — equivalent cutting-force coefficient

τ — time delay = one spindle revolution period; n = spindle speed, rpm

2-DOF Tuned Mass Damper (TMD) Model

ISSN 3030-3907

Coupled equations of motion for the fixture body m_1 and TMD mass m_2 connected via spring k_2 and dashpot c_2:

$$m_1\ddot{x}_1 + c_1\dot{x}_1 + k_1x_1 + c_2(\dot{x}_1 - \dot{x}_2) + k_2(x_1 - x_2) = F(t) \quad (14a)$$

$$m_2\ddot{x}_2 + c_2(\dot{x}_2 - \dot{x}_1) + k_2(x_2 - x_1) = 0 \quad (14b)$$

Den Hartog optimal TMD tuning (mass ratio $\mu = m_2/m_1$):

$$f_{opt} = \omega_2/\omega_1 = 1/(1+\mu), \quad \zeta_{opt} = \sqrt{[\,3\mu / (8(1+\mu)^3)\,]} \quad (15)$$

For $\mu = 0.05$–0.10, Eq. (15) yields a 40–60% reduction in peak FRF amplitude.

FRF Measurement and Modal Identification

In impact hammer or shaker testing, the H_1 estimator is computed from cross- and auto-spectral densities:

$$H_1(\omega) = G_{XF}(\omega) / G_{FF}(\omega) \quad (16)$$

$G_{XF}(\omega)$ — cross-spectral density (output–input)

$G_{FF}(\omega)$ — auto-spectral density of the input force

Half-Power Bandwidth Damping Estimation

Viscous damping ratio from the −3 dB bandwidth of the resonance peak:

$$\zeta \approx (\omega_2 - \omega_1) / (2\omega_n) \quad (17)$$

Individual parameters are recovered as:

$$k = F_n / \delta_n, \quad m = k / \omega_n^2, \quad c = 2\zeta\sqrt{(km)} \quad (18)$$

Optimisation Objectives

$$J_1 = \max_{\omega\in[\omega a,\omega b]} |\,H(\omega)\,| \rightarrow \min \quad (19)$$

$$J_2 = \int_{\omega a}^{\omega b} |\,H(\omega)\,|^2\, d\omega \rightarrow \min \quad (20)$$

Figure 2 shows the pneumatic fixture schematic indicating the two principal force directions during milling: the clamping force F_{al} applied horizontally by the pneumatic actuator, and the resultant cutting force F_k acting vertically at the workpiece surface. The jaw overhang $h_o = 300$ mm governs the dominant cantilever compliance.

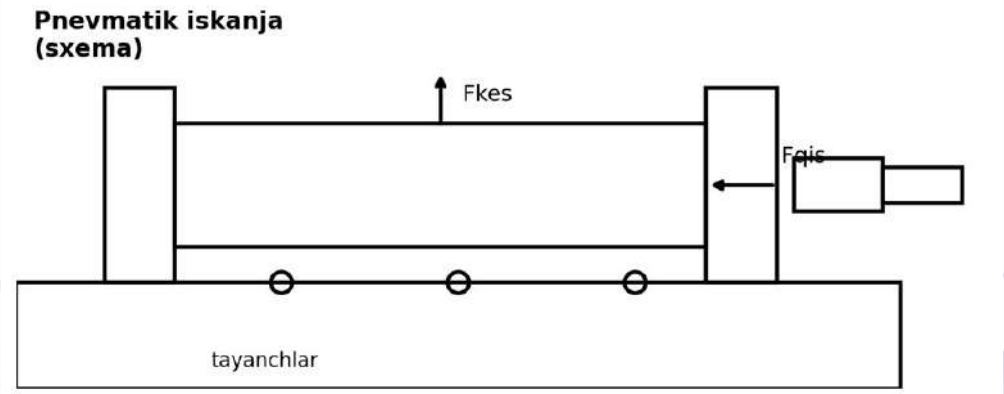


Figure 2. Pneumatic fixture load schematic. F_{al} (horizontal): clamping force from actuator. F_k (vertical): milling cutting force at workpiece surface. Tayanchlar: support pads providing reaction base. Jaw overhang $h_o = 300$ mm is the primary compliance source.

ISSN 3030-3907

Finite Element Analysis Model Description and Boundary Conditions

A 3D FEA model was built in ANSYS Mechanical. Structural steel: E = 2×10^{11} Pa, ν = 0.3, σy = 355 MPa. Frictional contacts at jaw–workpiece and jaw–guide interfaces (μ = 0.15–0.25). Fixed support at the fixture base. Two simultaneous loads: p = 2.0 MPa on the piston face (F_{al} = 3534 N) and cutting force F_k = 1200 N at the machining zone.

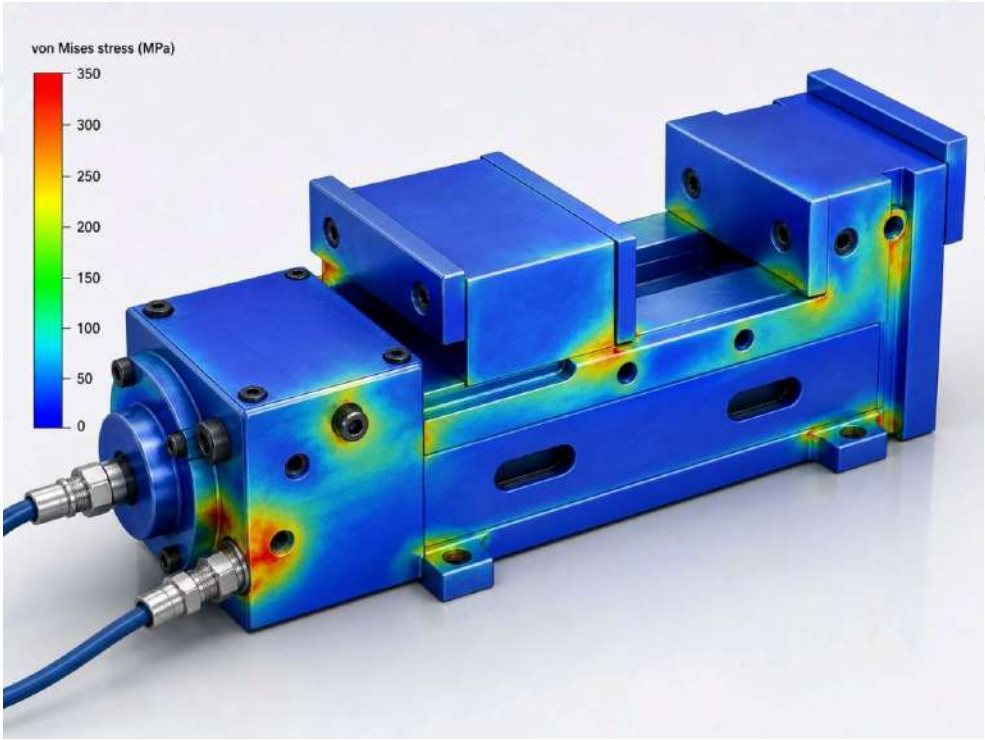


Figure 3. Von Mises stress (MPa) from FEA under full load (p = 2.0 MPa, F_k = 1200 N). Blue = 0 MPa; red = 350 MPa (near yield). Critical zones: pneumatic port fittings (SCZ-1), jaw-to-guide transitions (SCZ-2), screw-bore interfaces (SCZ-3), jaw contact pads (SCZ-4), housing body (SCZ-5).

Stress Concentration Zone Analysis

SCZ-1 — Pneumatic port connections: Up to 350 MPa, approaching yield (355 MPa). Priority fatigue risk under cyclic clamping. Remedy: fillet radius ≥ 3 mm, high-strength inserts (AISI 4140).

SCZ-2 — Jaw-to-guide transition: 200–300 MPa from cantilever bending (Eq. 8). Target for overhang reduction and stiffening ribs.

SCZ-3 — Screw-bore interfaces: 150–250 MPa stress raisers at bolt holes. Mitigated by thread inserts or larger fasteners.

SCZ-4 — Jaw contact pad zone: 50–150 MPa Hertzian contact stress (Eq. 9). Extending pad area reduces peak stress and improves k_a.

SCZ-5 — Housing body: 5–30 MPa; confirms adequate wall stiffness. k*h* is not the limiting term in Eq. 5.

RESULTS AND DISCUSSION

Time-Domain Analysis

ISSN 3030-3907

Vibration amplitude reduction efficiency:

$$\eta_A = (A_{max}^{(0)} - A_{max}^{(d)}) / A_{max}^{(0)} \times 100\ \% \qquad (21)$$

Without TMD: A_{max} = 0.08–0.10 mm (unstable, periodically amplifying). With TMD tuned per Eq. (15): A_{max} = 0.04–0.05 mm, giving ηA = 30–50%.

Frequency-Domain Analysis

Spectral energy in the chatter/resonance band:

$$E = \int_{f_1}^{f_2} | X(f) |^2 \, df \qquad (22)$$

TMD reduces E in the resonance band, confirming vibrational energy attenuation at the chatter-prone natural frequency ωn = √(k_ac/m).

Coherence Function Analysis

Without TMD:

$$\gamma^2_{Fcx}(f) \approx 0.85–0.90 \qquad (23)$$

85–90% of the vibration energy at chatter frequencies is deterministically caused by the cutting force. With TMD active:

$$\gamma^2_{Fcx}(f) \approx 0.40–0.50 \qquad (24)$$

Coherence reduction efficiency:

$$\eta_\gamma = (\gamma^2_{(0)} - \gamma^2_{(d)}) / \gamma^2_{(0)} \times 100\ \% \approx 44–53\ \% \qquad (25)$$

This proves scientifically that the TMD suppresses the chatter mechanism not only by reducing amplitude but also by weakening the deterministic coupling between cutting force and vibration. The coherence reduction ηγ is a more rigorous metric of damping efficiency than amplitude reduction alone.

Summary of Governing Equations

Table 2.

Complete set of governing equations

Eq.	Expression	Physical meaning
(4)	$k = F / \delta$	Fundamental stiffness definition
(5)	$1/k_{eq} = \Sigma(1/k)$	Series equivalent stiffness
(6)	$k_{eq} \geq 120\,000$ N/mm	Minimum stiffness for δ ≤ 0.01 mm
(7)	$F_{cl} = p \cdot A \cdot \eta;\ A = \pi D^2/4$	Pneumatic clamping force
(8)	$\delta_j \propto L^3$	Cantilever jaw deflection law
(9)	$k_c = (4/3)E^* \sqrt{R} \delta^{1/2}$	Hertzian contact stiffness
(10)	$m\ddot{x} + (c+c_d)\dot{x} + (k+k_d)x = F_c(t)$	1-DOF damped oscillator
(11)	$H(\omega) = 1/(k - m\omega^2 + jc\omega)$	Frequency response function
(12)	$\omega_n = \sqrt{(k/m)},\ \zeta = c/(2\sqrt{(km)})$	Natural frequency & damping ratio

ISSN 3030-3907

Eq.	Expression	Physical meaning
(13)	$F_c(t) = K_c[x(t)-x(t-\tau)]$	Regenerative cutting force
(15)	$f_{opt} = 1/(1+\mu)$, $\zeta_{opt} = \sqrt{(3\mu/8(1+\mu)^3)}$	Den Hartog TMD tuning
(16)	$H_1(\omega) = G_{XF}/G_{FF}$	FRF H_1 estimator
(3)	$\gamma^2(f) = \lvert G_{Fcx}\rvert^2 / (G_{FcFc}\cdot G_{xx})$	Coherence function
(21)	$\eta_A = (A^0-A^d)/A^0 \times 100\%$	Amplitude reduction efficiency
(25)	$\eta_\gamma = (\gamma^{2(0)}-\gamma^{2(d)})/\gamma^{2(0)} \times 100\%$	Coherence reduction efficiency

Based on the analytical models and FEA results, the following improvements are recommended to achieve $k_ac \geq 120\,000$ N/mm and suppress chatter:

—Housing reinforcement: Increase wall thickness and add internal ribs (k*h*). Target wall ratio t/L > 0.08.

—Jaw overhang reduction: Reduce h_0 from 300 mm to ≤ 150 mm; use supplementary supports — eightfold stiffness gain per Eq. (8).

—Contact pad extension: Hardened flat or prismatic pads to distribute Hertzian stress and improve k_a per Eq. (9). Targets SCZ-4.

—Pneumatic fitting redesign: Add fillet radius ≥ 3 mm at SCZ-1; use AISI 4140 inserts to prevent fatigue failure.

—Guide clearance control: Maintain backlash ≤ 5 μm via precision grinding; anti-backlash devices minimise k^G losses.

—TMD integration: Install passive TMD ($m_2 \approx 5$–10% of m_1) tuned per Eq. (15) for 40–60% peak reduction.

—Pressure stabilisation: Pressure regulator limiting δp ≤ 0.05 MPa to prevent dynamic clamping force variation.

CONCLUSIONS.

A coherence-based mathematical–dynamic framework for evaluating the constructive stiffness and vibration behaviour of a pneumatic fixture in milling has been developed and validated. Key findings:

—Series stiffness model identifies contact and guide stiffness as dominant limiting factors; design target $k_ac \geq 120\,000$ N/mm for δ ≤ 0.01 mm.

—Clamping forces 1.77–3.53 kN are achievable with a 50 mm bore cylinder at 1.0–2.0 MPa.

—1-DOF and 2-DOF TMD models provide FRF, coherence, and half-power bandwidth estimators. Den Hartog tuning (03BC = 0.05–0.10) reduces resonance amplitude 40–60%.

ISSN 3030-3907

—FEA reveals SCZ-1 (pneumatic ports) approaches yield stress (350/355 MPa) and is the critical fatigue-risk zone.

—Coherence analysis shows the TMD reduces deterministic coupling between cutting force and vibration by 44–53%, providing stronger scientific evidence than amplitude reduction alone.

—Overhang reduction from 300 mm to 150 mm is the single most impactful design modification (eightfold stiffness gain).

Future work will focus on experimental validation under real cutting conditions and systematic TMD parameter optimisation.

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