Pneumatic Industrial Manipulator End-Effector Tooling: Plate Thickness, Gripping Force and Safety Margin Coupled-Design Study

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Abstract: To address the practice of sizing custom end-effector tooling for manipulators by experience—plate thickness by rule of thumb, gripping force by simply increasing it, and safety margin by excessive conservatism— this study establishes a unified framework for coupled analysis of plate thickness, gripping force and safety margin within publicly available operating-condition boundaries. National standards, materials databases and manufacturer technical data are used to define the mechanical and thermophysical properties of commonly used structural steel, stainless steel, aluminum alloy, TPU pads and cast-iron workpieces. Verification models are established for friction gripping, cylinder thrust, bending deflection, Euler buckling, torsion, vacuum holding, and bolted, welded and pinned joints. Three worked examples—50 kg friction gripping, 80 kg rotation through 90°, and comparison of 6/8/10 mm plate thicknesses—are combined with illustrative finite-element analysis, sensitivity analysis and FMEA to form a closed analytical–simulation–test loop. Results show that, at a conservative design baseline of 0.5 MPa, plate thickness governs deflection by a cubic relationship, while eccentricity governs overturning moment linearly; these two variables often control design safety more directly than the material yield-strength grade. For vacuum holding, the worst case is usually not vertical lifting but lateral slip resistance with the suction cups in a vertical orientation. This framework provides a reproducible quantitative basis for robust design of custom end-effector tooling. Keywords: manipulator; end-effector tooling; plate thickness; clamping force; safety margin; vacuum gripping; sensitivity analysis CLC classification number: TH137; TP242 Document identification code: A

Coupled Design of Plate Thickness, Clamping Force and Safety Margin for End-Effectors of Pneumatic Power-Assist Manipulators

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Abstract: To address the empirical “thickness-by-experience, force-by-amplification, margin-by-conservatism” problem in designing non-standard end-effectors for pneumatic power-assist manipulators, a unified coupled framework relating plate thickness, clamping force and safety margin is established under publicly available operating boundaries. The mechanical and thermophysical parameters of common structural steels, stainless steel, aluminium alloy, TPU pads and cast-iron workpieces are determined from national standards, material databases and manufacturer data. Verification models for friction clamping, cylinder thrust, bending deflection, Euler buckling, torsion, vacuum adhesion and bolt/weld/pin connections are built. Three worked examples—a 50 kg friction clamp, an 80 kg 90° flip and a 6/8/10 mm thickness comparison—together with schematic finite-element analysis, sensitivity analysis and FMEA form an analysis–simulation–test closed loop. Results show that, under a conservative 0.5 MPa design baseline, plate thickness governs deflection by a cubic law and eccentricity governs the flipping moment by a linear law, both controlling safety more directly than the nominal yield grade of the material; the worst case for vacuum adhesion is generally not vertical lifting but lateral anti-slip with the cup in a vertical posture. The framework provides a reproducible quantitative basis for robust design of non-standard end-effectors.

Key words: pneumatic power-assist manipulator; end-effector; plate thickness; clamping force; safety margin; vacuum adhesion; sensitivity analysis

0 Introduction

The fundamental difference between custom handling end-effector tooling and a general-purpose robot gripper is not whether it can grip the workpiece, but whether it can maintain a verifiable safety margin at the minimum compressed-air supply pressure, the worst surface condition, the maximum eccentricity and the specified cycle time. Existing gripping studies show that The coupling method and friction mechanism will significantly affect the stable grasping under external disturbance; the sealing ring and groove parameters in the vacuum suction cup geometry also It will change the local stress and adsorption capacity [1-2]. The former shows that the force transmission chain and constraint distribution themselves are the core variables of grasping stability. The latter shows that the design of end-effector tooling cannot stay at the empirical level of "enough clamping force", but should combine structure, contact, drive, environment and Process paths are considered as coupled systems.

Public engineering information shows manipulators used for insulated transfer of high-temperature cast-iron parts, vacuum rotation of automotive headliners, Machine stator internal support type handling, car seat assembly eccentric clamping and other scenarios, and the engineering process of "analysis-design-review-production" is given, As well as rated working air pressure of 0.5 to 0.7 MPa, standard and heavy-duty load classification, conventional radius and lifting stroke range [3]. This study

These disclosures are only used as "working condition boundaries" and "application prototypes" and are not directly used as material constants or performance conclusions, so as to make the model suitable for Should be based on real industrial scenarios.

The objects studied are the custom-tooling main plate, gripper-jaw/suction-cup assembly, and drive cylinder mounted on the manipulator end flange (or flip actuator cylinder), intermediate lever/connecting rod and its connecting parts; excluding the fatigue life of the entire machine arm body, basic installation strength and the entire line The control logic only focuses on the static strength, stiffness, stability and reliability of the end-effector tooling body in the "clamping-handling-turning-release" link. Verify safety margin.

1 Boundary conditions and data sources

1.1 Disclosure of boundary conditions Table 1 gives the public boundary conditions used in this study. The calculation example uses 0.5 MPa as the conservative design baseline and 0.7 MPa as the upper limit for verification.

Table 1 The public boundary conditions used in this study

Project | Open boundaries/input | Purpose of this study Rated working pressure | 0.5□0.7 MPa | Typical design window; 0.5 MPa is conservative baseline, 0.7 MPa | Check for upper limit Load binning | Standard type <150 kg; heavy duty ≥150 kg | Determine whether the calculation example belongs to the end-effector tooling problem within the standard range Regular scope of work | Radius ≤ 3 500 mm; lifting stroke ≤ 2 000 mm | end-effector tooling needs to balance light weight and stiffness environmental boundaries | −15□50 □; relative humidity <90% | Open working condition boundary at normal temperature; high temperature workpiece will be revised separately Public application prototype | High temperature cast iron parts, car roofs, stators, car seats | chair | Corresponds to four scenarios: high temperature, vacuum, large eccentricity, and internal support. engineering process | Analysis-Design-Review-Production | Organize the verification and delivery logic of this study Note: The public working conditions, pressures and application examples in the table are all from the public page of the manufacturer's official website [3].

1.2 Data sources and citation priority In order to reduce the risk of "empirical parameter drift" in enterprise technical analysis, this study adopts a hierarchical number selection strategy: the first priority is national standards and official standard platform for grade definition, thickness-related strength lower limit and test methods [4-5]; the second priority is material manuals and data library for typical values necessary for structural design such as elastic modulus, Poisson's ratio, density and thermal expansion coefficient [6]; the third priority is the manufacturer's technology technical information for the engineering selection rules for vacuum gripping and tooling/actuators [13]; the fourth priority is peer-reviewed papers for supplementary Full grasping stability, suction cup geometric effect and other mechanisms [1-2]. Any values that are typical values rather than measured values in the delivery state are clearly marked in the text.

2 Material parameters and theoretical models

2.1 Symbol description Table 2 gives the main symbols and units uniformly used in this study.

Table 2 Main symbols and units

symbol | meaning | unit | symbol | meaning | unit m / Workpiece mass / kg / P / Compressed-air supply pressure / MPa mg / Tooling/rotator-head mass / kg / D / Cylinder bore / mm a | motion acceleration | m/s² | d | Piston rod diameter | mm Fcyl / Theoretical cylinder thrust / N / i / Mechanism amplification ratio / - ηm / Mechanical efficiency / - / NΣ / Total normal force / N symbol | meaning | unit | symbol | meaning | unit μ | Friction coefficient | - | S | safety factor | - L | Cantilever length/span | mm | b | Effective width of board | mm t | Plate thickness | mm | I | Sectional moment of inertia | mm⁴ σ | normal stress | MPa | τ | shear stress | MPa δ | Deflection | mm | e | center of gravity eccentricity | m Md | Design turning moment | N·m | J | Torsional/polar moment of inertia | mm⁴ Δp / Effective vacuum differential pressure / kPa / Aeff / Effective suction area / m² ηs / Sealing/suction efficiency / - / Pcr / Critical buckling load / N

2.2 Material parameters and data sources This study does not equate "high material strength" with "tooling is better": for tooling, stiffness, thermal stability, wear resistance and connection technology are the same. Just as important. Steel grade upgrading usually significantly increases the yield strength but has little effect on the elastic modulus and therefore does not significantly change the same geometric structure. The deflection of the structure; on the contrary, although aluminum alloys significantly reduce weight, the stiffness is usually only about one-third of steel when the geometry remains unchanged [6]. Figure 1 gives the board The coupling relationship between thickness-drive-friction-safety margin.

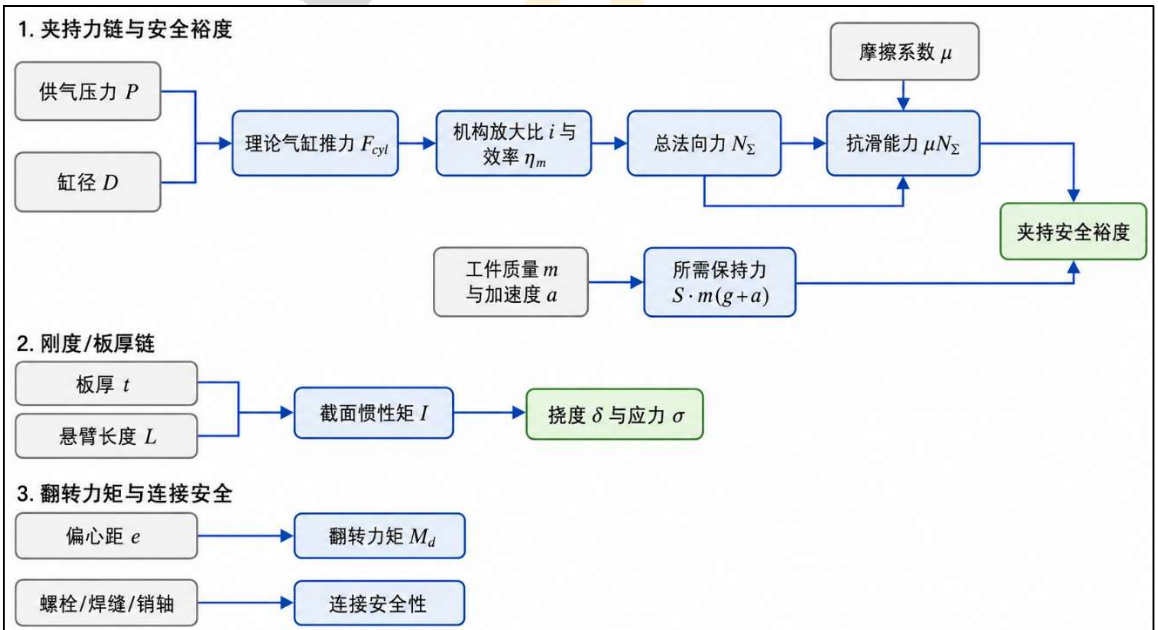

Figure 1 Schematic diagram of coupling relationship of end-effector tooling

Table 3 summarizes the representative material parameters used in the calculation examples in this study.

Table 3 Representative material parameters

Material | E/GPa | ν | ρ/(kg·m−3) | σy/MPa | α/(10−6·□−1) | design description Q235 structural steel 190□210 0.27□	
Q355 low alloy steel 190□210 0.27□	0.30 | 7 850 | 235 | ≈12 | Universal welded panels 0.30 | 7 850 | 355(≤16);345(>16~40) | ≈12 | Increased strength, stiffness | Near General Steel
45 steel 190□210 0.27□	0.30 | 7 850 | ≈355 | ≈12 | Suitable for pins and trunnions
304 stainless steel | 193 | 0.29 | 8 000 | 215 | 17.3 | Corrosion resistant, relatively thermal expansion | Big	

Material | E/GPa | ν | ρ/(kg·m−3) | σy/MPa | α/(10−6·□−1) | design description 6061-T6 aluminum alloy | 68.9 | 0.33 | 2 700 | 276 | ≈23 | Lightweight and strong | falling TPU 95A type pad | layer | Level 0.14 | - | 1 210 | - | - | Contact buffer layer, need | Consider creep Cast iron workpiece | 80□160 | 0.2□0.3 | 6 900□7 400 | Changes with brand number | 11□14 | High temperature parts need to be insulated and | Derate Note: E, ν, ρ, α of structural steel and 45 steel adopt the typical range of the material properties appendix [6]; Q355 thickness-related yield strength comes from GB/T 1591-2018 [5]; 304 and 6061-T6 use publicly available material data sheets [7-8]; TPU uses the Elastollan C95A data sheet [9]; the room-temperature cast-iron range uses the NIST thermophysical-property compilation [10]. All values above are typical pre-design values; formal engineering work shall be governed by material certificates and any required incoming inspection.

It should be noted that flexible cushioning such as TPU cannot be regarded as an accessory that "only changes the touch feel". In public data sheets, Elastollan The tensile modulus of C95A is about 140 MPa, the Shore A hardness is about 96, and the 23 □ creep modulus-time curve is attached, indicating that under continuous clamping Considerable time-varying compliance and stress relaxation will occur in the contact layer under high-frequency cycling [9]. For high temperature castings, the NIST summary shows that cast iron room temperature Young's modulus is about 80 to 160 GPa, linear expansion coefficient is about 11 to 14×10−6·□−1, and temperature rise will significantly change the contact conditions and dimensional chain; Public cases of high-temperature iron castings also use insulation structure and cooling pneumatic circuit to reduce its impact [10][3].

2.3 Friction clamping and cylinder thrust model For bilateral friction clamping, ignoring uneven local contact pressure, the basic anti-slip condition is:

$$\mu N_{\Sigma} \geq Sm(g+a) \tag{1}$$

The total normal force required is

$$N_{\Sigma} \geq \frac{Sm(g+a)}{\mu} \tag{2}$$

When the two sides are symmetrical, the unilateral normal force is

$$N_{side} = \frac{N_{\Sigma}}{2} \tag{3}$$

Note the definition used: some standards define gripping force as the force on one side, whereas slip resistance is normally determined from the sum of the normal forces on both sides. This study consistently uses NΣ for total normal force and Nside for one-side force. As a published test reference for grippers, T/SZRA 001— 2024 recommends calculating the rated load with a coefficient of friction no greater than 0.2 and a safety factor no less than 2 [11]. This study uses those values as a conservative pre-design line.

The theoretical thrust of the cylinder is determined by the effective pressure area and air supply pressure. For the extension side and the return side, respectively

$$F_{cyl,out} = \frac{P\pi D^2}{4} \tag{4}$$

$$F_{cyl,in} = \frac{P\pi (D^2 - d^2)}{4} \tag{5}$$

If there are levers, wedges or linkage amplification, the total normal force available is

$$N_{\Sigma} = i\,\eta_m F_{cyl} \tag{6}$$

In the formula: i is the mechanism amplification ratio; ηm is the mechanical efficiency, which is used to uniformly absorb sealing, guide and mechanism losses [12]. The coefficient of friction is not Can be fixed based on experience: The manufacturer's technical information emphasizes that it must be tested and determined under the "original workpiece-real surface-real pollution state". Public reference intervals are roughly 0.1 for oily surfaces, 0.2 to 0.3 for wet surfaces, 0.5 for dry wood/metal/glass/stone, and rough surfaces. About 0.6[13].

2.4 Bending, deflection, buckling and torsion models

The tooling main plate is approximated as a cantilever beam with a rectangular cross-section. The cross-sectional moment of inertia, maximum bending stress at the root and free end deflection are respectively

$$I=\frac{bt^3}{12} \tag{7}$$

$$\sigma_{max}=\frac{6FL}{bt^2} \tag{8}$$

$$\delta_{max}=\frac{4FL^3}{Ebt^3} \tag{9}$$

Equations (8) and (9) reveal the key facts of plate thickness design: stress decreases with the square of t, while the deflection decreases with the cubic of t. Therefore, in tooling, which requires clamping parallelism, positioning accuracy and flipping stability, the plate thickness is first of all the "stiffness variable", and its importance is very high on the "intensity variable". For slender rods and thin-walled supports under pressure, Euler buckling should be checked at the same time; for flip shafts, flange sleeves and pins, The Saint-Venant torsion relation may be used:

$$P_{cr}=\frac{\pi^2 EI}{(KL)^2} \tag{10}$$

$$\tau_{max}=\frac{Tr}{J}, \quad \varphi=\frac{TL}{JG} \tag{11}$$

Since the elastic modulus of most steels is between 190 and 210 GPa, the difference in E between structural steel and high-strength steel is much smaller than the difference in yield strength. are different, so the improvement in deflection by "raising the steel grade" is usually much smaller than "increasing the plate thickness" or "shortening the cantilever"; while the E of 6061-T6 is only about 68.9 GPa, when the geometry remains unchanged, the deflection of aluminum tooling is about 3 times that of steel [6].

2.5 vacuum gripping model The normal holding force of vacuum gripping comes from the pressure difference

$$F_N=\Delta p\, A_{eff}\, \eta_s \tag{12}$$

where Δp is the effective differential pressure, Aeff is the effective suction area, and ηs is the sealing efficiency. Vacuum percentage is a relative value referenced to ambient pressure. As a relative quantity, the atmospheric pressure at sea level is about 101.3 kPa, so the 60% vacuum degree can be approximately taken as Δp≈60.8 kPa [14]. For three typical handling scenarios, The theoretical holding forces are: vertical lifting (suction cup horizontal), horizontal transfer (suction cup horizontal), the most unfavorable working conditions (suction cup vertical, Withstand vertical force)

$$F_H=m(g+a)\,S \tag{13}$$

$$F_H=m\left(g+\frac{a}{\mu}\right)S \tag{14}$$

$$F_H=\frac{m}{\mu}(g+a)\,S \tag{15}$$

If n pieces of suction cup are used, the theoretical holding force required for a single cup is

$$F_{H,1}=\frac{F_H}{n} \tag{16}$$

The calculation should be based on the most unfavorable load situation in the entire handling sequence, rather than just looking at the lifting moment. Safety system for smooth and dense surfaces The number should be at least 1.5, and for porous, rough, oily or heterogeneous surfaces, it should be 2.0 and above; porous materials should be in the 30% to 55% vacuum range To ensure flow compensation, the dense surface should obtain higher holding power in the 55% to 80% vacuum range [13,15].

A supplementary example illustrates the importance of the pose path: use 6 suction cups, each 60 mm in diameter, at 60% vacuum with ηs=0.85. The theoretical force per cup normal force approx.

$$F_{N,1}=0.85\times 60.8\times 10^3\times\pi\times 0.03^2\approx 146\ \mathrm{N}$$

The total normal force is about 876 N, and the static load ratio for a 25 kg dense plate is about 3.57. It seems to be fully safe according to the vertical lifting condition; but when the work

When the parts enter the most unfavorable working condition of "suction cup vertical and longitudinal stress", when μ=0.25, a=1 m/s², and S=2, the required total holding force increases to about 2 162 N, 6 ϕ60 suction cup is obviously insufficient. It can be seen that the design object of vacuum tooling is "complete action sequence" rather than "single lifting action".

2.6 Connection check In addition to judging the nominal load of the connector, looseness, hole wall crushing, fatigue and assembly deviation should also be considered. This study uses the following names The first-order screening is performed using the correct calibration formula - the average shear stress of the bolt, the pressure-bearing stress of the hole wall, and the average shear stress of the double shear pin are respectively

$$\tau_b = \frac{V}{n\,A_s} \tag{17}$$

$$\sigma_{br} = \frac{V}{n\,d\,t} \tag{18}$$

$$\tau_p = \frac{2V}{\pi d^2} \tag{19}$$

The effective throat thickness, effective area and nominal shear stress of the fillet weld are

$$a = 0.707\,z \tag{20}$$

$$A_w = 0.707\,z\,L_w, \quad \tau_w = \frac{V}{A_w} \tag{21}$$

Public welding design data generally use 0.707z as the approximate effective throat thickness of fillet welds [17]; for bolted connections that rely on friction and anti-slip, The slip coefficient is significantly affected by surface conditions and should be obtained through representative tests rather than directly copying empirical values [16].

3 Typical calculation examples and finite element solutions

The three sets of calculation examples all use 0.5 to 0.7 MPa as the air supply window and 0.5 MPa as the conservative baseline. The pre-designed baseline for friction clamping is μ≤0.2, S≥2[3,11]. All geometric dimensions, mechanism efficiency and eccentricity are engineering example values set to illustrate the method.

3.1 Calculation example 1: 50 kg double-sided friction clamping Question: A two-sided friction gripper handles a 50 kg workpiece. Let a=1.0 m/s², μ=0.20, S=2.0, i=3.0 and ηm=0.85. The calculation proceeds as follows: Calculate as follows.

$$N_\Sigma = \frac{S\,m(g+a)}{\mu} = \frac{2 \times 50 \times (9.81 + 1.0)}{0.20} = 5405\ \text{N}$$

$$N_{side} = N_\Sigma / 2 = 2702.5\ \text{N}$$

$$F_{cyl,\,min} = \frac{N_\Sigma}{i\,\eta_m} = \frac{5405}{3.0 \times 0.85} = 2119\ \text{N}$$

Comparison of candidate bore diameters is shown in Table 4.

Table 4 Calculation Example 1 Comparison under different bore sizes and pressures

Bore diameter	Pressure/MPa	Fcyl/N	i ηm Fcyl/N	Margin to NΣ
ϕ63	0.5	1 559	3 974	0.74
ϕ63	0.6	1 870	4 769	0.88
ϕ63	0.7	2 182	5 564	1.03
ϕ80	0.5	2 513	6 409	1.19
ϕ80	0.6	3 016	7 691	1.42
ϕ80	0.7	3 519	8 972	1.66

Conclusion: If the freezing plan is based on the 0.5 MPa baseline, ϕ63 is obviously insufficient and ϕ80 has an available margin; even if the on-site supply is stable at 0.7 MPa, ϕ63 only reaches the boundary value and is not suitable for conservative design. It can be seen that the value of the design baseline in friction clamping is more important than the "highest possible pressure" key.

3.2 Example 2: Rotation of an 80 kg workpiece through 90° Question: 80 kg workpiece is flipped 90°, the eccentricity of the center of gravity of workpiece is e=0.30 m; the mass of the flip head and local components is mg=15 kg, the offset center-of-gravity eccentricity eg=0.15 m; combined rotation design factor κM=2.2; effective moment arm r=0.12 m; ηm=0.85.

$$M_s = mge + m_g g\, e_g = 80 \times 9.81 \times 0.30 + 15 \times 9.81 \times 0.15 = 257.5\ \mathrm{N \cdot m}$$

$$M_d = \kappa_M M_s = 2.2 \times 257.5 = 566.5\ \mathrm{N \cdot m}$$

$$F_{cyl,min} = \frac{M_d}{r\eta_m} = \frac{566.5}{0.12 \times 0.85} = 5554\ \mathrm{N}$$

Comparison of candidate bore diameters is shown in Table 5.

Table 5 Calculation Example 2 Comparison under different bore sizes and pressures

Bore diameter	Pressure/MPa	Fcyl/N	Available load moment/(N·m)	Margin to Md
ϕ100	0.5	3 927	400.6	0.71
ϕ100	0.6	4 712	480.7	0.85
ϕ100	0.7	5 498	560.8	0.99
ϕ125	0.5	6 136	625.9	1.10
ϕ125	0.6	7 363	751.0	1.33
ϕ125	0.7	8 590	876.2	1.55

Conclusion: The primary variable in the 90° flip problem is the eccentricity rather than the yield strength of the steel plate. If the eccentricity is reduced from 0.30 m to 0.20 m, When other conditions remain unchanged, the static gravity moment is reduced by about 30%, which is often more effective in reducing the system than upgrading the plate from Q235 to Q355. risk.

3.3 Calculation example 3: 6/8/10 mm plate thickness comparison Question: Compare three plate thicknesses of 6/8/10 mm for the same main plate, treat it as a cantilever beam with a rectangular cross-section, and take the effective width b=160 mm, Cantilever length L=150 mm, end equivalent load F=1 000 N, E=210 GPa. The calculation results from equations (7) to (9) are shown in Table 6.

Table 6 Calculation Example 3 Calculation results of different plate thicknesses

Plate thickness/mm	I/mm4	σmax/MPa	δmax/mm	relative stiffness
6	2 880	156.3	1.86	1.00
8	6 827	87.9	0.785	2.37
10	13 333	56.3	0.402	4.63

If the process requires that the end deflection does not exceed 0.50 mm, then 10 mm meets the requirement, 8 mm is close to the upper limit, and 6 mm is obviously insufficient. It is worth noting that although the maximum bending stress of 6 mm plate is 156 MPa, which does not exceed the typical yield strength of Q235 of 235 MPa, its deflection is much larger. Exceeding the process allowable value, this is a typical situation where "plate thickness is first controlled by stiffness rather than strength".

3.4 Finite element simulation solution In formal projects, the above analytical model should be used for quick screening rather than as a substitute for detailed simulation. The recommended finite element scheme is shown in Table 7.

Table 7 Suggested finite element modeling scheme

Project Recommended settings model The main plate, flange, ear plate, connecting seat and gripper jaw bracket constitute a three-dimensional solid model Material For structural steel, take the median value of E=206 GPa, ν=0.30, ρ=7 850 kg·m□^{3}; replace 304 or 6061-T6 according to Table 3 unit Second-order tetrahedron or hexahedron; overall mesh 4~5 mm, hole edge/round corner/ear plate root part 1~1.5 mm

Project Recommended settings contact gripper jaw-workpiece is friction contact; the welding area is first simplified into a merged body, and then a local sub-model is made for the welding toe. constraint The end flange hole ring is fully constrained; the flip head is hinged or coupled constrained according to the actual rotation axis load Gravity, clamping equivalent normal force, eccentric moment, overturning inertia load; lateral impact amplification is taken into account when necessary output Total displacement, Mises stress, maximum principal stress, contact pressure, hole edge and ear plate root stress, connection load grid independence Local refinement is about 30%, and the change in key stress and displacement is <5%, which is considered passed.

The reason for using the median values E and v of structural steel is that the elastic modulus of steel is 190~210 GPa and Poisson's ratio is 0.27~ 0.30, taking the median value in early simulation can avoid false precision of a certain grade [6]. Figure 2 shows the suggested finite element modeling and stress hot spots. Location indication.

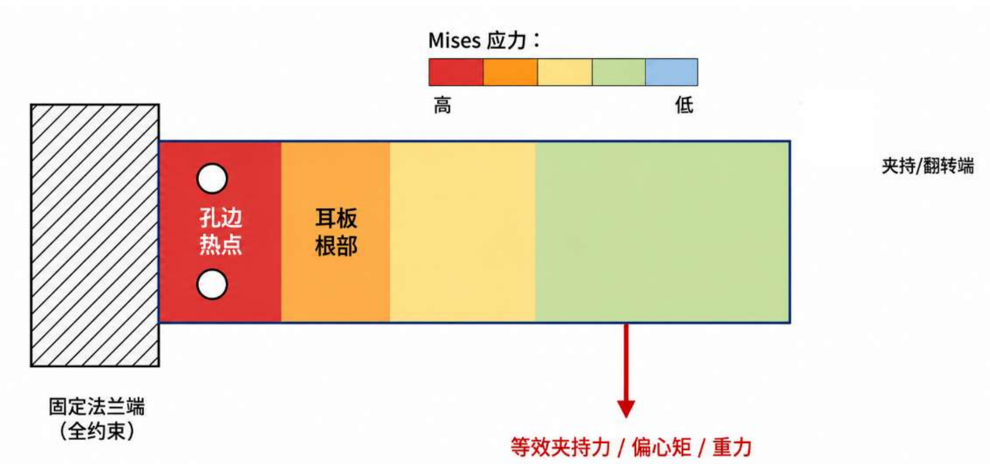

Figure 2 Finite element model and stress hot spot diagram

The results above are illustrative finite-element results used only to compare trends with the analytical solution; they are not an acceptance analysis for a specific product. The writing method and verification standards of the finite element chapter of the academic paper do not replace the actual calculation of the project. The three groups of plate thicknesses of 6/8/10 mm in Example 3 are still used. The results are shown in Table 8.

Table 8 Schematic finite element results

Plate thickness/mm | Indicates the maximum Mises stress/MPa | Indicates maximum displacement/mm | Stress ratio for Q235 | Main hotspots 6 | ≈170 | ≈1.93 | 0.72 | Flange hole edge, cylinder | root of ear plate 8 | ≈96 | ≈0.81 | 0.41 | Hole edge transition fillet 10 | ≈61 | ≈0.42 | 0.26 | Flange connection area part

The schematic value remains of the same order as the analytical solution: when the plate thickness increases, the displacement decreases according to the cube of t, the stress decreases according to the quadratic power of t, and the hole The local peak value at the edge and ear plate root is slightly higher than that of the analytical beam model, which is due to the stress concentration caused by geometric discontinuity. Formal projects should be supplemented with at least Local refinement of the welding toe and opening angle, bolt pre-tightening and contact nonlinear quadratic recalculation; thermal load and contact should also be superimposed under high temperature conditions Thermal derating analysis.

4 Sensitivity analysis, experimental verification and FMEA

4.1 Sensitivity analysis The simple first-order sensitivity can be directly given by the model. The friction clamping safety margin, plate deflection and flipping moment are respectively

$$\gamma_g \propto \frac{PD^2 \mu i \eta_m}{m(g+a)}, \quad \delta \propto \frac{L^3}{Et^3}, \quad M_d \propto m e$$

Based on this, the first-order sensitivity of Table 9 is constructed.

Table 9 First-order sensitivity of main variables

variable | Main influence on response | mathematical sensitivity | Response changes when variable +10% Air supply pressure P | Clamping safety margin | +1 | +10% Bore diameter D | Clamping safety margin | +2 | +21% Friction coefficient μ | Clamping safety margin | +1 | +10% Mechanism amplification ratio i / Gripping safety margin / +1 / +10% Mechanical efficiency η□ | Clamping safety margin | +1 | +10% Workpiece mass m / Gripping safety margin / −1 / −9.1% Plate thickness t | Deflection | −3 | −24.9% Cantilever length L | Deflection | +3 | +33.1% Modulus of elasticity E | Deflection | −1 | −9.1% eccentricity e | Turning moment | +1 | +10%

Table 9 reveals three key facts: first, the primary geometric variable for clamping safety is the bore, which enters thrust as the square; second, The stiffness of the main plate is extremely sensitive to the plate thickness and cantilever length, because it enters the deflection with the third power; thirdly, the flipping scene is most afraid of eccentricity drift. Therefore, priority should be given to "compressing the eccentricity" in the planning stage rather than "compensating" with higher pressure at the end.

4.2 Experimental verification plan T/SZRA 001-2024 stipulates the public test prerequisites for gripper jaw equipment: ambient temperature 20±2 □, relative humidity 20%~ 80%, air pressure 86~106 kPa, and it is recommended to fully warm up before testing, and use 50 cycle statistics for force control/repetitive positioning/peak torque [11]. For vacuum mode, please refer to the leakage rate test path of GB/T 34878-2017 [18]; for fatigue of materials or representative components, ASTM E466 constant amplitude axial fatigue test is supplemented [19]. The recommended verification process is shown in Table 10.

Table 10 Recommended experimental verification process

stage | target | Main equipment/instruments | output Confirmation of incoming materials Confirm plate thickness, material, heat treatment Status

Calipers, ultrasonic thickness gauges, hardness testers, material certificates | instructions | Actual t, material status

Clamping force calibration | Establish pressure-clamping force curve | Pressure sensor, tension pressure sensor, acquisition system | system | P-Nside curve Slip-resistance test / Back-calculate equivalent coefficient of friction μeq / Vertical loading tooling, weights/servo actuator, displacement sensor / Incipient-slip load and μeq Deflection test | Verify plate thickness selection | Laser displacement meter, standard load block | F-δ curve flip test | Verify 0°/45°/90° pose moment | Torque sensor, inertia simulation block, angle encoding | device | M-θ curve and overshoot repeatability test | Verify clamping and positioning discreteness | Automatic circulation platform, camera/displacement meter | Positioning error repeated 50 times Vacuum-holding test / Verify vacuum retention and leakage / Vacuum gauge, flowmeter, leak-test equipment / Pressure-drop–time curve and effective differential pressure Durability and fatigue / Verify service-life risk / Cycle test bench; coupon fatigue testing if required / Life curve and loosening threshold

It is recommended to perform static clamping force calibration in three gears of 0.5, 0.6, and 0.7 MPa (each ≥ 5 steady-state points), and then perform calibration on the real workpiece or equivalent Anti-slip loading was performed on the specimen, and μeq was back-calculated in four states: dry at room temperature, slightly oily, grinding dust and high temperature respectively; then the deflection and

Flip test to verify analytical solution and simulation trend, repeatability test ≥50 times. The vacuum module maintains pressure and leaks under different vacuum degrees. Test, and convert the pressure drop slope into effective pressure difference attenuation. Finally, parameters such as μ, ηm, and κM can be backfilled into the analytical model to “Experience value” is converted into “field identification value” [11].

4.3 Failure Mode and Effects Analysis (FMEA) Table 11 gives the FMEA recommendations for end-effector tooling. S, O, and D range from 1 to 10, and the risk priority number RPN=S×O×D.

Table 11 End-Effector Tooling FMEA

failure mode | Main causes/consequences | S | O | D | RPN | Recommended actions Friction loss | μ Dropping, oil pollution, insufficient pressure/falling | 10 | 4 | 4 | 160 | Real workpiece measures μ; 0.5 MPa base | Line; add anti-skid teeth/covering Excessive board deflection | Insufficient plate thickness, too long cantilever/alignment error | 8 | 5 | 4 | 160 | Prioritize thickening or shortening the cantilever; add ribs instead of | Not just upgrading steel grade Insufficient turning torque | The eccentricity is estimated to be too small and the dynamic load is missed or stagnant. | 9 | 3 | 5 | 135 | Test the center of gravity first; introduce κM; leave enough margin | Degree Weld fatigue cracking / Weld-toe stress concentration and residual stress/fracture / 9 / 3 / 6 / 162 / Weld-toe dressing, fillet transitions and hot-spot fatigue reassessment Bolt-preload loss / Vibration, settlement and thermal cycling/loosening / 7 / 5 / 5 / 175 / Preload-torque control, locking measures, periodic inspection and retightening Pin wear and increased clearance | Frequent turning, insufficient lubrication/deterioration of accuracy | 7 | 4 | 5 | 140 | Replaceable bushing; improve surface hardness Vacuum leakage / Suction-cup aging and uneven surface/dropped workpiece / 10 / 3 / 4 / 120 / Pressure-retention test, zoned vacuum and periodic replacement High temperature seal derating | Insufficient thermal isolation/clamping force drift | 8 | 3 | 6 | 144 | Heat shielding, cooling pneumatic circuit, select heat-resistant and | Derate

As far as RPN is concerned, bolt preload attenuation, weld fatigue cracks, excessive plate deflection and friction slip are the most worthy of priority control-it Their average stress may not be the highest, but it is most likely to amplify into failure after accumulation of cycles, heat, contamination and assembly deviations.

5 Discussion

This study discusses materials, mechanics, contact and technology in the same coordinate system. Its limitation is that it is still a preliminary engineering analysis framework. Rather than the final solution for a real three-dimensional product. First, E, ρ, ν, and α in the material table are mostly typical values, which are affected by the manufacturing process, Defects, temperature and load history are affected. For formal analysis, the material supplier should be consulted; 304 and 6061-T6 data correspond to specific states. Aluminum Alloys are sensitive to post-weld softening, and base metal data cannot be equated with post-weld regional properties [6][7-8].

Second, both the friction and vacuum models adopt the first-order approximation of "Coulomb friction + effective pressure difference". The manufacturer's data emphasizes that the friction coefficient must be within Experiments on the original workpiece determined that the vacuum degree selection is also directly related to whether the surface is porous [13-15]. The model can distinguish "which variable is more sensitive", But it cannot replace the calibration test under real surface, temperature and contamination conditions; especially for high-temperature castings, insulation, cooling and sealing Derating may be more effective than “increasing the clamping force”[10][3].

Third, the finite element part is a schematic result rather than an acceptance report. It is deliberately kept at the same level as the analytical solution and is intended to illustrate the writing method of the paper. Boundary conditions and hotspot identification. The formal project should at least add weld geometry and residual stress, bolt pretension and contact slip, and flexible pads. Four types of corrections are superimposed on layer viscoelasticity or creep, high temperature and cyclic loading, and are backfilled with field test data before they can be converted to Reproducible engineering standard parts.

6 Conclusion

(1) There is a clear coupling relationship between end-effector tooling plate thickness, clamping force and safety margin. The influence of plate thickness on deflection is cubic level, in most working conditions requiring parallelism and low jitter, plate thickness is first a matter of stiffness.

(2) For steel tooling, only upgrading Q235 to Q355 significantly increases the yield strength but hardly changes the same geometric stiffness; "upgrading "Steel grade" is usually not as effective as "increasing plate thickness, reducing cantilever, and reducing eccentricity".

(3) In the 50 kg friction-gripping example, under the conservative assumptions of 0.5 MPa and μ=0.20, a ϕ80 cylinder bore meets the requirement whereas a ϕ63 No, it means that the design baseline should be the lowest guaranteed pressure rather than the highest possible pressure.

(4) In the 90° flip example of 80 kg, eccentricity is the dominant variable, and the risk of flipping is mainly affected by mass, eccentricity and comprehensive dynamics. load factor control.

(5) For vacuum gripping, the most unfavorable working conditions usually come from lateral anti-skid in the complete action sequence rather than single vertical lifting, friction The coefficients and safety factor must be calibrated by real workpiece testing. In summary, the path to robust design is not to “infinitely increase the clamping force”; “First identify the real boundary, and then use analysis-simulation-test closed loop to freeze plate thickness, cylinder diameter, connection and surface parameters” [6]. Design and verification The closed loop is shown in Figure 3.

Figure 3 Design and verification closed-loop process

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