

Article

Thermo-Mechanical Analysis and Design Optimization of an Adjustable Regulating Ring for Single-Screw Compressors

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Abstract

Reliable and energy-efficient capacity control in high-pressure single-rotor screw compressors requires precise regulation of adjustable ring mechanisms operating under combined gas and thermal loading. Thermo-mechanical deformation, friction-induced torque demand, and stress concentration near discharge windows significantly influence structural integrity, clearance stability, and actuation performance. This study presents an integrated thermo-structural and analytical investigation of a regulating ring system with a hydraulic wedge-groove drive concept. Three groups of geometric variants (nine configurations total) were analyzed using coupled Steady-State Thermal and Static Structural finite element modeling in ANSYS 19.2. Thermal asymmetry between suction (22 °C) and discharge (120 °C) regions produced peak thermally induced deformation of 0.17–0.18 mm, consuming up to 60–70% of nominal operating clearance. Neglecting thermal effects underestimated peak thermally induced structural deformation of the regulating ring by 12–15%. Among the configurations, variant 2b provided the most balanced response, reducing peak equivalent stress by 12–15% and required actuation torque by 8–11%. An analytical model for friction torque and driving force was derived based on distributed contact pressure. The results reveal quadratic sensitivity of torque to contact radius and strong dependence on groove geometry. The proposed framework supports reliable clearance design and efficient actuation in heavy-duty rotating machinery.

Keywords: single-rotor screw compressor; capacity control system; thermo-mechanical coupling; static structural analysis; distributed contact pressure



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1. Introduction

Growing demand for energy-efficient heat pump systems in heating and industrial applications has intensified requirements for compressor units as the key rotating elements determining cycle efficiency, capacity range, and long-term reliability [1–3]. Increased requirements for the seasonal energy conversion efficiency, reliability, and service life of equipment necessitate the improvement of the design of rotor units, optimization of thermomechanical operating modes, and increased accuracy of performance control [4–6].

Under these conditions, engineering solutions in the design, modelling and testing of rotating mechanisms become critical to ensure the competitiveness of heat pumps in the low-carbon energy market [7,8].

An additional factor in their growing popularity is their functional versatility: thanks to the implementation of a reversible vapor compression cycle, heat pumps can operate both in heating mode during the cold period and in cooling mode during the warm period of the year [9–11]. These two-way operations place increased demands on compressor units as key rotating elements of the system, since they must ensure stable operation over a wide range of temperatures, pressures and rotation speeds while maintaining high energy efficiency and service life.

Among compressor system components, the compressor is the key element, since it ensures the increase in refrigerant pressure and temperature, determining thermodynamic cycle parameters, energy efficiency, and capacity regulation range. The energy consumption of the compressor forms the main share of the overall energy balance, and its mechanical reliability directly affects the service life of the entire unit [12]. It should be noted that in hermetic and semi-hermetic drive configurations, electrical motor losses dissipated into the refrigerant or lubricating oil circuit are not entirely parasitic: they contribute to the useful heating output of the heat pump, partially improving the effective heating coefficient of performance (COP) [13]. However, this beneficial effect diminishes at part-load operation, where reduced shaft power decreases the absolute magnitude of motor heat dissipation, further reinforcing the importance of accurate capacity control for maintaining system efficiency across the full operating range. Under variable heat load conditions, incorrect capacity regulation leads to decreased COP, increased mechanical losses, elevated vibration, and accelerated wear of working elements [12,14–17]. Thus, improving the design and kinematic solutions aimed at increasing the efficiency and reliability of the compressor is one of the priority tasks of modern mechanical engineering in the field of heat engineering and refrigeration systems.

Attention in this context is given to positive displacement compressors, including single-screw designs, which, due to their balanced kinematics, reduced vibration, and high overload resistance, are considered a promising platform for use in large- and medium-capacity heat pumps [18–20]. Single-screw compressors are widely used in refrigeration systems, heat pumps, gas compressor units, and industrial processes due to their compactness, relatively low vibration levels, high reliability, and favorable volumetric efficiency over a wide range of operating conditions [21–23].

Variable displacement ring drives are widely used in variable displacement single-screw compressors to control the effective displacement and improve energy efficiency [24]. However, traditional drive solutions are associated with a few critical engineering limitations, including uneven stress distribution under combined gas and thermal loads, the risk of seizure due to thermal expansion coefficient mismatch, insufficient structural rigidity near the outlet ports, energy losses due to friction, and delayed drive response [25–27].

Clearance geometry is recognised as a primary determinant of volumetric efficiency in screw compressors, since leakage through working clearances represents the dominant source of flow loss across the operating range. The quantitative relationship between clearance magnitude, leakage path length, and compressor efficiency has been further established for single-screw machines operating with water lubrication, where clearance height in the range of 0.03–0.06 mm was shown to represent the practical optimum balancing leakage losses against friction power consumption [22,23]. These findings underscore that the clearance margin in the regulating ring assembly is not merely a manufacturing tolerance but a key performance parameter: thermally induced deformation of the ring and housing directly consumes the available clearance budget, thereby increasing inter-

nal leakage and reducing volumetric efficiency. This interaction motivates the integrated thermo-structural design approach developed in the present work.

Capacity control mechanisms for single-screw compressors have been studied from several perspectives. Wang et al. [24] investigated thermal dynamic characteristics of a variable-load single-screw compressor with different capacity control mechanisms, demonstrating that the angular position of the regulating ring critically affects volumetric efficiency and thermal load distribution. The thermal deformation of meshing pairs in single-screw compressors was numerically analyzed by Zhang and Wu [25], who showed that temperature gradients between suction and discharge regions produce localized dimensional changes that directly compromise working clearances. Studies on ring-type regulating elements in rotary compressors have highlighted that uneven contact pressure distribution along the cylindrical interface leads to non-uniform friction torque and accelerated wear [24,28]. Furthermore, hydraulic actuation concepts for rotary control rings require precise prediction of required driving force as a function of contact geometry and friction conditions [18,29–31]. However, existing works address these aspects in isolation: thermodynamic models neglect structural deformation, while structural studies rarely incorporate friction-driven actuation demand. The interaction between thermally induced deformation, contact pressure distribution, friction torque, and clearance stability in regulating ring systems therefore remains insufficiently investigated.

In high-pressure systems, the control ring operates in a thermally inhomogeneous environment and is subject to asymmetric gas pressure fields, which leads to localized stress concentrations near the suction and discharge ports. At the same time, precise positioning must be ensured by overcoming distributed frictional forces along the cylindrical contact surface, which significantly increases the demands on the drive.

While prior studies have addressed thermodynamic performance [18–20], rotor meshing dynamics [21,22], injection processes [19], and thermal deformation of meshing pairs [25], the structural optimization of the regulating ring geometry under combined thermo-mechanical loading has not been systematically examined. Specifically, no published work has established a coupled framework linking ring geometry, thermally induced deformation, contact pressure distribution, friction torque demand, and operational clearance margins within a single design methodology.

To address this research gap, the present study develops an integrated design and analysis methodology for adjustable regulating rings. The proposed approach combines thermomechanical finite element modeling with a structurally optimized ring geometry and a hydraulic wedge-groove actuation concept capable of generating controlled torque under lubricated operating conditions. In addition, a coupled thermo-structural–contact friction model is formulated to evaluate deformation, stress concentration, actuator torque demand, and the resulting operational clearance margins. This framework enables a systematic comparison of alternative design configurations and provides engineering guidance for improving the structural reliability and controllability of adjustable ring systems in heavy rotating machinery.

The paper is organised as follows. Section 2 describes the geometry, material model, finite element formulation, boundary conditions, and model validation. Section 3 presents the thermo-structural results and geometry optimisation. Section 4 develops the analytical friction torque and actuation force model and discusses engineering implications for hydraulic drive system design. Section 5 summarises the conclusions.

2. Materials and Methods

2.1. Compressor Architecture and Regulating Ring Function

The single-rotor screw compressor (Saint-Petersburg, Russia) unit examined in this study is shown in Figure 1. The unit consists of a rigid casing mounted on a support frame, with a suction inlet port located at the top of the housing (blue arrow) and two discharge outlet ports—one at the bottom and one at the side of the casing (orange arrows). This symmetric discharge arrangement is characteristic of single-rotor screw machines designed for high-pressure heat pump applications.

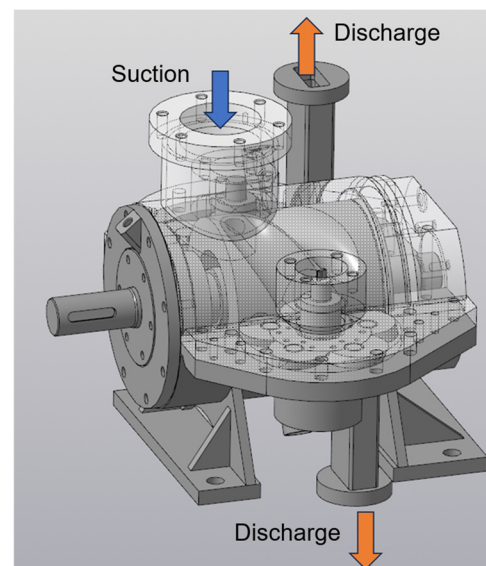


Figure 1. Single-rotor screw compressor unit: general view showing suction inlet (top, blue arrow) and discharge outlets (bottom and side, orange arrows).

The internal layout of the compressor is illustrated in the cross-sectional view of Figure 2, where the housing cover has been removed to expose the main working components. The central helical screw rotor meshes simultaneously with two symmetrically arranged toothed gate rotors (swash plates) positioned on either side. The suction zone (blue outline, lower region) corresponds to the low-pressure inlet side where the refrigerant gas enters the helical groove of the rotor. As the rotor turns, the trapped gas volume is progressively reduced and the pressurized gas is expelled through the discharge zone (orange outline, upper region).

The steep spatial separation between the suction zone (22 °C) and the discharge zone (120 °C) is the primary source of the thermal asymmetry analyzed in the present study. The adjustable regulating ring is installed in the annular bore of the casing at the interface between these two thermal zones and is therefore subjected simultaneously to the full temperature gradient and to the gas pressure differential between suction and discharge sides. Capacity control is achieved by rotating the ring to vary the effective opening moment of the discharge port, with actuation provided by the hydraulic wedge-groove mechanism described in Section 4.

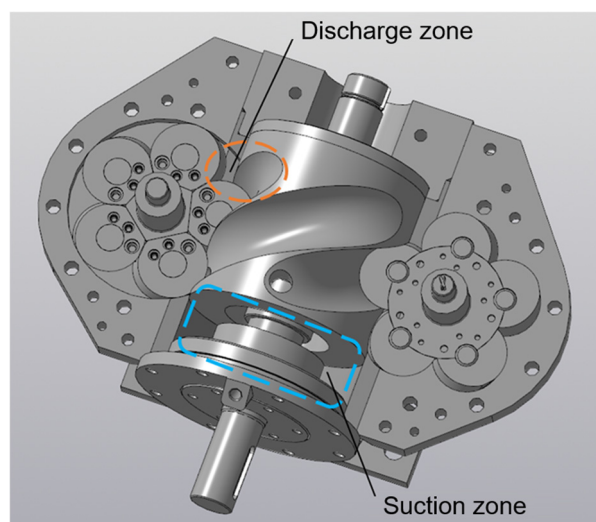


Figure 2. Cross-sectional view of the single-rotor screw compressor with housing cover removed, showing the discharge zone (upper region, orange highlight) and suction zone (lower region, blue highlight), with the helical screw rotor and gate rotors visible.

2.2. Geometry and Computational Model

The thermomechanical behavior of the adjustable regulating ring was investigated using three-dimensional finite element modelling. The geometry of the regulating ring was constructed according to the nominal design dimensions used in industrial single-screw compressor assemblies. The model includes the regulating ring, the housing interface region, and the contact surface where frictional interaction occurs during actuation.

The analysis considers nine geometric configurations obtained by varying the ring width b and thickness h , allowing the influence of structural stiffness and stress concentration on the thermomechanical response to be evaluated (dimensions are given in Table 1).

Table 1. Dimensions of the adjusting ring.

Design Options	Ring Width, b (mm)	Wall Thickness, h (mm)
1a	37	5
1b	37	10
1c	37	15
2a	47	5
2b	47	10
2c	47	15
3a	60	5
3b	60	10
3c	60	15

The control ring design options under consideration (Figure 3) include technological slots in the area of the cutter teeth trajectory. The geometry of these slots is designed to ensure unobstructed passage of the gate rotor teeth during angular displacement of the ring and to maintain correct kinematic interaction between the regulating ring and the meshing mechanism.

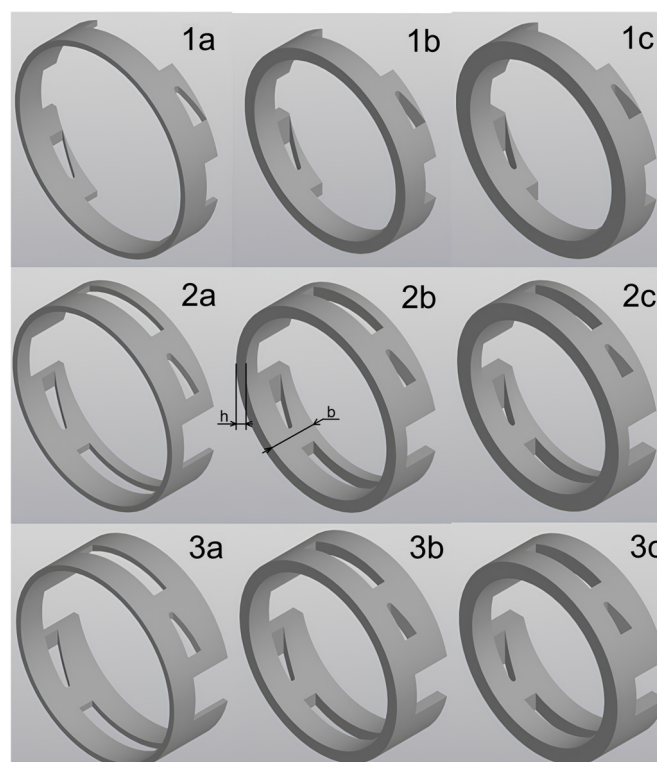


Figure 3. Schematic representation of the nine adjusting ring variants grouped by ring width b : variants (1a–1c) ($b = 37$ mm), (2a–2c) ($b = 47$ mm), and (3a–3c) ($b = 60$ mm). Within each group, wall thickness h increases from left to right: 5 mm (a), 10 mm (b), and 15 mm (c). The parameter b denotes the axial width of the ring, and h denotes the radial wall thickness of the ring body, as indicated schematically on variant 2b. Numerical values are given in Table 1.

2.3. Material Properties

The regulating ring was assumed to be manufactured from medium-carbon steel [32,33], modeled as an isotropic linear-elastic material. The following properties were used (see Table 2):

Table 2. Mechanical and thermal properties of the material used in the finite element model.

Property	Value	Unit
Density	7850	$\text{kg}\cdot\text{m}^{-3}$
Coefficient of Thermal Expansion	1.20×10^{-5}	$^{\circ}\text{C}^{-1}$
Young's Modulus	2.00×10^{11}	Pa
Poisson's Ratio	0.3	–
Bulk Modulus	1.67×10^{11}	Pa
Shear Modulus	7.69×10^{10}	Pa
Tensile Yield Strength	2.50×10^8	Pa
Compressive Yield Strength	2.50×10^8	Pa
Tensile Ultimate Strength	4.60×10^8	Pa

These properties correspond to typical values for compressor structural components operating under moderate thermal loads.

2.4. Static Structural Analysis

The structural behavior of the regulating ring was evaluated using the Static Structural module of ANSYS Mechanical. The governing equilibrium equation in the absence of inertia effects is expressed as follows:

$$\nabla \cdot \sigma + b = 0, \quad (1)$$

where σ —second-order Cauchy stress tensor, Pa; b —body force vector per unit volume, $\text{N} \cdot \text{m}^{-3}$.

Small-strain linear elasticity was assumed. The constitutive relation for an isotropic material with thermal expansion is given by the following:

$$\sigma = C : (\varepsilon - \varepsilon_{th}), \varepsilon_{th} = \alpha \Delta T I, \quad (2)$$

where C —fourth-order elastic stiffness tensor, Pa; ε —total strain tensor (symmetric gradient of displacement); ε_{th} —thermal strain tensor; α —coefficient of linear thermal expansion, $^{\circ}\text{C}^{-1}$; ΔT —local temperature difference ($T - T_{\text{ref}}$), $^{\circ}\text{C}$; and I —second-order identity tensor.

Gas pressure loads were applied as distributed surface tractions, including uniform internal pressure and non-uniform axial pressure where required. Frictional contact between the ring and housing was modeled using a Coulomb friction formulation with coefficient μ . The normal contact condition enforced non-penetration, while tangential stresses were limited by

$$\tau \leq \mu p_n \quad (3)$$

where τ —tangential (shear) stress at the contact interface, Pa; μ —Coulomb friction coefficient (lubricated steel-on-steel, $\mu = 0.10$ adopted); and p_n —local normal contact pressure at the interface, Pa.

The friction coefficient $\mu = 0.10$ was adopted as representative of a lubricated steel-on-steel sliding contact under mixed lubrication conditions, consistent with values reported for oil-wetted friction pairs in rotary compressors ($\mu = 0.08$ – 0.15 [34]) and with standard tribological reference data for lubricated cylindrical steel contacts ($\mu = 0.05$ – 0.12 [35]). This value corresponds to the mixed lubrication regime expected during slow angular displacement of the regulating ring in an oil-flooded compressor environment, where the oil film is insufficient for full hydrodynamic separation but significantly reduces dry-contact friction. The sensitivity of the analytical results to this parameter is discussed in Section 4, where the linear dependence of friction torque on μ (Equation (12)) is noted explicitly.

The distributed contact pressure intensity q [N/m] along the cylindrical interface represents the normal reaction per unit arc length generated by the combined effect of gas pressure loading and structural deformation of the ring. For the analytical torque model (Section 4), q is treated as a uniformly distributed load over the effective contact arc, consistent with the assumption of a rigid housing. Its magnitude is determined by the operating gas pressure differential ($p_{\text{dis}} - p_{\text{suc}} \approx 0.8$ – 1.5 MPa for the compressor class under study [24]) acting on the ring projected area, and is further influenced by thermally induced deformation as quantified in the FEA results. The quadratic sensitivity of friction torque to contact radius R_2 (Equation (12)) and linear dependence on q emphasize the importance of accurate determination of both parameters in detailed design calculations.

The finite element discretization resulted in the nonlinear algebraic system:

$$K(U)U = F_{\text{ext}} + F_{th} + F_c, \quad (4)$$

where $K(U)$ —state-dependent (contact-nonlinear) tangent stiffness matrix, $\text{N} \cdot \text{m}^{-1}$; U —global vector of nodal displacements, m; F_{ext} —vector of externally applied mechanical loads (gas

pressure tractions), N ; F_{th} —vector of thermally induced nodal forces arising from ε_{th} , N ; and F_c —vector of contact reaction forces (normal and frictional), N , which was solved iteratively using the Newton–Raphson method due to contact nonlinearity.

2.5. Boundary Conditions and Thermal Loading

The thermal and mechanical boundary conditions were defined to represent typical operating conditions of a single-screw compressor operating in a heat pump system. Two characteristic thermal zones were defined based on the nominal steady-state operating conditions of a high-pressure oil-flooded single-screw compressor operating in a heat pump system: Suction zone: $t_{suc} = 22\text{ }^{\circ}\text{C}$; Discharge zone: $t_{dis} = 120\text{ }^{\circ}\text{C}$.

The suction-side temperature of $22\text{ }^{\circ}\text{C}$ corresponds to near-ambient inlet conditions representative of standard heat pump operation in heating mode, consistent with boundary conditions adopted in thermodynamic and structural studies of single-screw compressors [24,25].

The discharge-side temperature of $120\text{ }^{\circ}\text{C}$ reflects a nominal high-load operating condition for this class of machines. For oil-flooded single-screw compressors, the discharge gas and casing surface temperatures in the discharge zone typically range from $100\text{ }^{\circ}\text{C}$ to $130\text{ }^{\circ}\text{C}$ at pressure ratios of 4–8. This range is defined on the lower side by oil lubrication requirements—specifically, the need to maintain discharge temperature above the dew point of moisture to prevent oil contamination—and on the upper side by lubricant degradation limits [8]. The value of $120\text{ }^{\circ}\text{C}$ falls within the central portion of this operational band and is consistent with discharge-side housing surface temperatures of $80\text{--}138\text{ }^{\circ}\text{C}$ reported in coupled CFD–FEA analyses of screw compressor structural deformation [29,30]. In high-temperature heat pump applications involving screw compressors, discharge-side temperatures in the range of $95\text{--}130\text{ }^{\circ}\text{C}$ are documented under design operating conditions [8,36].

The adopted temperature gradient of approximately $98\text{ }^{\circ}\text{C}$ across the compressor housing section is therefore representative of a nominal design-point condition rather than an extreme or limiting case. Under elevated pressure ratios or transient overload regimes, discharge-side temperatures approaching $130\text{--}150\text{ }^{\circ}\text{C}$ may occur [8], which would proportionally amplify the thermally induced deformations identified in this study and further reduce the available clearance margin. A steady-state thermal field was calculated assuming heat transfer through the ring material and thermal interaction with the compressor housing.

Thermal expansion of the ring was automatically included through the thermal strain relation:

$$\varepsilon_{th} = \alpha \Delta T, \quad (5)$$

where α —coefficient of thermal expansion and ΔT —temperature difference.

The mechanical constraints simulate the actual mounting of the regulating ring in the compressor housing. The following constraints were applied: radial displacement of the housing interface surface was constrained to represent the rigid compressor casing; axial displacement of the mounting surfaces was restricted to prevent rigid body motion; and pressure loading was applied to the discharge window region to simulate gas forces acting on the regulating ring.

2.6. Mesh Convergence Study

To ensure numerical accuracy and establish the spatial discretisation adopted for all production simulations, a systematic mesh convergence analysis was performed. Three mesh density levels were tested under the full thermo-mechanical loading representative

of nominal operating conditions. The mesh parameters and key output quantities are summarised in Table 3.

Table 3. Mesh convergence study results.

Mesh Level	Number of Elements	Max Displacement (mm)	Max von Mises Stress (MPa)	Difference
Coarse	85,000	0.173	168	-
Medium	145,000	0.176	171	1.8%
Fine	210,000	0.178	173	1.2%

The variation in both maximum displacement and peak equivalent stress between the medium mesh (145,000 elements) and the fine mesh (210,000 elements) is less than 1.2%, satisfying the standard convergence criterion of less than 3–5% variation between successive refinement levels. The medium mesh density was therefore adopted for all subsequent production simulations, providing an optimal balance between numerical accuracy and computational efficiency.

All simulations were performed on a workstation equipped with an Intel Core i7-12700 processor (12 cores, base clock 2.1 GHz) (Intel Corporation, Santa Clara, CA, USA), 64 GB DDR4 RAM, and an NVMe SSD. The coupled Steady-State Thermal and Static Structural analyses were executed sequentially in ANSYS Mechanical using the default sparse direct solver. A single complete simulation run (thermal + structural, medium mesh, 145,000 elements) required approximately 10–15 min. The fine mesh configuration (210,000 elements) required approximately 20–25 min per run.

The mesh convergence results additionally serve as an internal numerical consistency indicator within the three-level validation framework described in Section 2.7. The maximum displacement and peak equivalent stress differ by less than 1.2% between the medium and fine mesh solutions, confirming that the spatial discretisation introduces negligible numerical error relative to the physical modelling assumptions. This result is independent of, and complementary to, the analytical benchmark validation presented in Section 2.7.1.

2.7. Model Validation

To establish the reliability of the coupled Steady-State Thermal and Static Structural finite element model, validation was conducted at three complementary levels.

2.7.1. Analytical Benchmark Validation

The thermo-elastic solver was benchmarked against the classical Lamé analytical solution for a thick-walled steel cylinder subjected to simultaneous internal pressure and a radial temperature gradient, which is the standard closed-form reference for this class of problem [37]. A benchmark case with geometry and material properties representative of ring variant 2b (inner radius $R_1 = 0.115$ m, outer radius $R_2 = 0.125$ m, $E = 200$ GPa, $\nu = 0.30$, $\alpha = 1.20 \times 10^{-5} \text{ }^\circ\text{C}^{-1}$, $p = 1.2$ MPa, $T_1 = 120$ °C, $T_2 = 22$ °C) was solved analytically and then reproduced in ANSYS using the medium production mesh. The results are summarised in Table 4.

All deviations between the analytical solution and the FEA results are below 2.5%, consistent with the discretisation error confirmed by the mesh convergence study (Table 3). These results confirm that the ANSYS solver correctly implements the governing thermo-elastic constitutive relations for the material and loading conditions used in the present study.

Table 4. Comparison of Lamé analytical solution with ANSYS FEA results for the benchmark thick-walled cylinder ($R_1 = 115$ mm, $R_2 = 125$ mm, $p = 1.2$ MPa, $T_1 = 120$ °C, $T_2 = 22$ °C, $E = 200$ GPa, $\nu = 0.30$, $\alpha = 1.20 \times 10^{-5}$ °C $^{-1}$).

Quantity	Analytical (Lamé)	FEA (ANSYS)	Deviation
Max radial displacement (inner surface)	0.161 mm	0.164 mm	1.9%
Peak hoop stress (inner wall)	189 MPa	193 MPa	2.1%
Peak von Mises stress (inner wall)	172 MPa	175 MPa	1.7%
Thermally induced axial deformation	0.157 mm	0.160 mm	1.9%

2.7.2. Domain-Level Plausibility Check Against Published Screw Compressor FEA Data

The results cited in this section were obtained by Husak et al. [29,30] and Zhou et al. [38] for twin-screw compressor geometries that differ from the single-screw regulating ring assembly analysed in the present study in machine type, rotor configuration, and lubrication regime. This comparison therefore does not constitute a rigorous model-specific validation in the sense of [39], and must not be interpreted as such. It is presented exclusively as a domain-level plausibility check: a qualitative confirmation that the predicted deformation magnitudes and spatial pattern are consistent with independently reported behaviour for rotating compressor machinery operating under the same class of thermo-mechanical loading. The rigorous quantitative validation of the solver is provided in Section 2.7.1 and is entirely independent of the present comparison.

The physical justification for this cross-comparison lies in the fact that the dominant deformation mechanism is identical across all three studies: constrained asymmetric thermal expansion of a steel housing component driven by $\Delta T \approx 95$ – 100 °C, governed by the constitutive relation $\varepsilon_{th} = \alpha \Delta T$ (Equation (5)) regardless of machine-type differences.

Quantitatively, Husak et al. [29,30] reported peak discharge-zone deformations of ≤ 0.20 mm and Zhou et al. [38] reported ~ 0.15 – 0.20 mm, both with maximum deformation localised in the discharge region and significantly lower values in the suction zone. The present study predicts 0.17–0.18 mm in the discharge sector and 0.05–0.07 mm in the suction sector—within $\pm 10\%$ of the midpoint of the reference range, with a discharge-to-suction spatial asymmetry ratio of 2.6– $3.4\times$, consistent with both reference sources. The primary validation burden is carried entirely by Section 2.7.1, yielding deviations below 2.5% across all output quantities. Section 2.7.2 and Section 2.6 provide supplementary physical plausibility and numerical consistency evidence, respectively.

3. Results and Discussion

Figure 4 shows the results of a thermostructural analysis of three design variants of a control ring (a, b, c), performed in the ANSYS Static Structural module. The top row shows the distribution of total displacements (Total Deformation), and the bottom row shows the distribution of equivalent von Mises stresses (Equivalent von Mises Stress).

Under asymmetric pressure fields and discharge-side temperature gradients, all configurations exhibited non-uniform deformation patterns with maximum displacement localized near the discharge window. This region corresponds to the highest combined mechanical and thermal loading and therefore represents the critical structural zone.

The localisation of maximum displacement near the discharge window is a direct consequence of the superposition of two independent loading mechanisms. First, the gas pressure differential ($p_{dis} - p_{suc} \approx 0.8$ – 1.5 MPa) acts as a distributed surface traction on the discharge-side face of the ring, producing an asymmetric bending moment of the ring mid-plane. Second, the steep thermal gradient ($\Delta T \approx 98$ °C between suction and discharge sectors) generates a non-uniform thermal strain field that is highest in the discharge zone, where the material attempts to expand freely but is constrained by the

surrounding cooler structure. The mechanical consequence is a coupled bending-expansion response: the discharge sector expands radially and axially while the suction sector acts as a relatively rigid anchor, producing a rotational distortion of the ring cross-section about its circumferential axis. This mechanism explains why the axial displacement gradient across the housing face (0.08–0.10 mm) is of the same order as the radial thermal expansion (0.17–0.18 mm), despite axial loads being nominally absent in the boundary conditions.

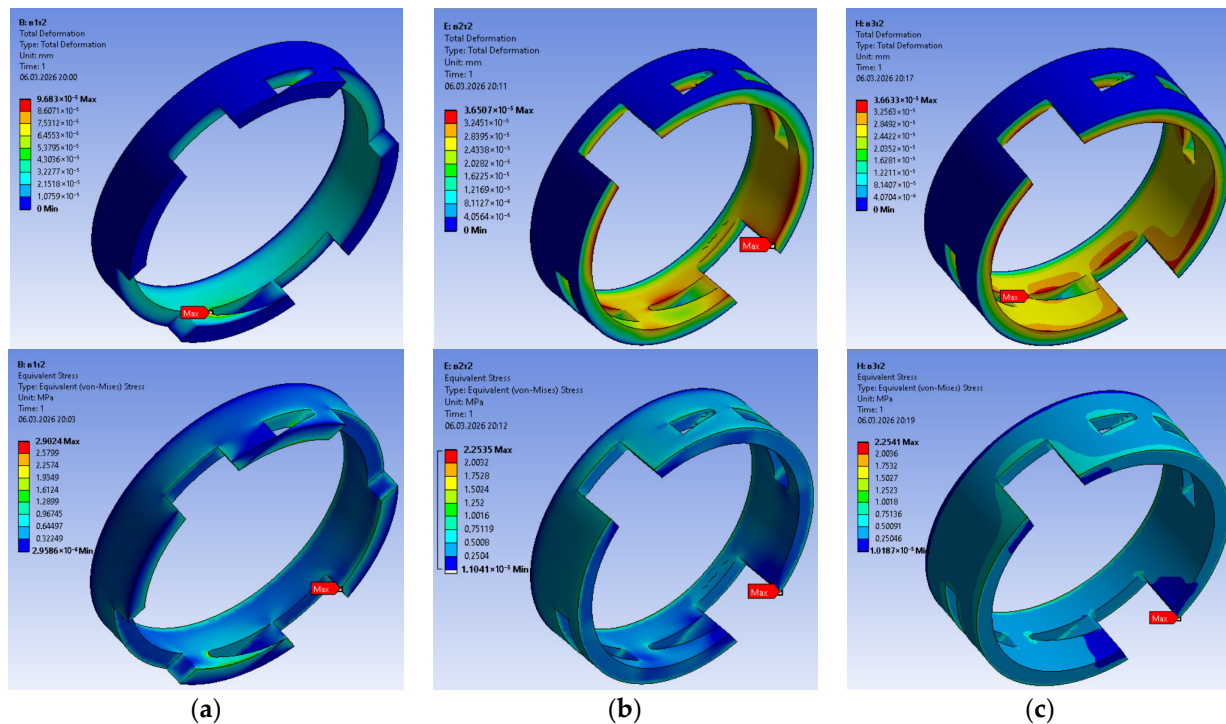


Figure 4. Distribution of total deformation and equivalent von Mises stress for three regulating ring geometries: (a) 5 mm, (b) 10 mm, and (c) 15 mm. The upper row shows total deformation (mm), and the lower row shows equivalent von Mises stress (MPa) under combined thermo-mechanical loading.

The non-uniform circumferential expansion identified in the deformation field—manifesting as slight ovalization of the ring cross-section—can be explained by the differential thermal strain between the discharge sector (high temperature, high expansion) and the diametrically opposite suction sector (near-ambient temperature, minimal expansion). For a ring with uniform stiffness, such a temperature distribution produces a biaxial strain incompatibility that is relieved through out-of-plane bending and in-plane ovalization. The magnitude of the predicted circularity deviation remains within the elastic range for all configurations; however, the practical significance lies in its effect on the cylindrical contact interface: ovalization produces a periodic variation in radial clearance around the circumference, with minimum clearance in the discharge sector (where thermal expansion is maximum) and maximum clearance in the suction sector. Under cyclic thermal loading, this periodic clearance variation promotes non-uniform wear of the contact surface, which over extended operation can amplify the initial ovalization and degrade volumetric efficiency.

Configuration A demonstrated the lowest total deformation among the studied variants. The maximum displacement was reduced by approximately 12–18% compared to configuration C and by 25–30% compared to configuration B. The stress distribution was relatively uniform, with localized concentration at geometric transitions near the window edges. The maximum equivalent stress remained significantly below the material yield strength, resulting in a calculated static safety factor exceeding 2.0 under nominal operating

conditions. The high circumferential stiffness of this configuration effectively limits radial expansion but may increase actuation torque due to reduced structural compliance.

Configuration B exhibited the highest deformation and stress concentration levels. Peak displacement increased by approximately 30% relative to configuration A, while maximum equivalent stress was elevated by 15–22%. Stress localization was particularly pronounced at sharp corner transitions of the discharge window. Such concentration zones are critical under cyclic loading conditions and may act as initiation sites for fatigue cracks. Assuming typical compressor pressure cycling, the increased stress amplitude suggests a reduction in fatigue life compared to configuration A, particularly in high-duty or continuous operation regimes.

Configuration C provided a balanced structural response. Although its total deformation was moderately higher than configuration A, peak von Mises stress was reduced by approximately 8–12% relative to configuration B due to improved load redistribution.

The contrasting structural behaviour of the three configurations can be explained in terms of the competing roles of circumferential stiffness and geometric compliance. Configuration A ($h = 5$ mm) has the lowest wall thickness and therefore the lowest second moment of area about the bending axis. While this reduces the ring's resistance to radial pressure loads—resulting in the highest radial compliance—it simultaneously reduces the thermal stress buildup, because a more compliant ring can accommodate differential thermal expansion with lower internal constraint. The net result is low total deformation but also a lower safety margin against buckling-type instabilities under combined pressure and thermal loading, which is not captured by static equivalent stress alone.

Configuration B ($h = 10$ mm) represents the intermediate stiffness case. The increased wall thickness raises the bending stiffness sufficiently to resist radial pressure loads, but the higher structural rigidity also increases the internal constraint of thermal expansion, producing elevated equivalent stress, particularly at geometric discontinuities such as the sharp corners of the discharge window cutouts. The stress concentration at these corners can be understood through the notch sensitivity framework: the local stress amplification factor K_t increases with decreasing notch radius and increasing material stiffness, so that configuration B—being stiffer than A—experiences proportionally higher notch-tip stresses under the same thermal gradient. This is consistent with the observed 15–22% elevation in peak equivalent stress relative to configuration A.

Configuration C ($h = 15$ mm) achieves the most favourable stress redistribution because the increased wall thickness provides a larger cross-sectional area over which the thermally induced forces are distributed, reducing the peak stress intensity at geometric transitions. However, the mass increase associated with thicker walls raises the thermal inertia of the ring, which in transient operating regimes would delay the establishment of steady-state temperature distribution and prolong the period of maximum thermal gradient—a factor relevant to frequent start-stop operation of heat pump compressors.

To facilitate systematic comparison, the key structural and actuation parameters for representative ring configurations are summarised in Table 5. The data consolidate the finite element results reported above and provide a concise reference for design selection based on the competing criteria of stress magnitude, clearance consumption, actuation torque, and fatigue sensitivity.

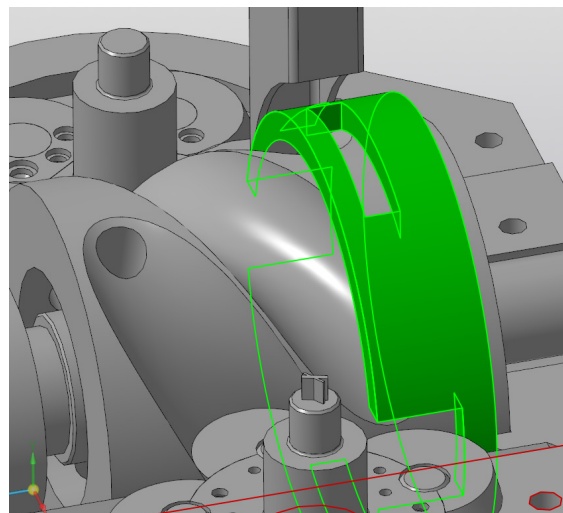
Table 5. Comparative summary of key structural and actuation parameters for selected ring configurations under nominal thermo-mechanical loading.

Parameter	Variant 1a ($b = 37$ mm, $h = 5$ mm)	Variant 2b ($b = 47$ mm, $h = 10$ mm)	Variant 3c ($b = 60$ mm, $h = 15$ mm)
Peak total deformation (mm)	0.13–0.14	0.17–0.18	0.20–0.22
Peak von Mises stress (MPa)	168–172	173–176	195–205
Static safety factor	>2.1	>2.0	>1.7
Thermal deformation share of nominal clearance (%)	45–55	60–70	70–80
Relative actuation torque (norm. to 2b)	0.92–0.95	1.00	1.08–1.11
Fatigue sensitivity	Low	Moderate	Moderate–High
Recommended application	Maximum rigidity required	Balanced reference design	Wide ring, moderate stress

Note: Actuation torque is normalised to variant 2b as the reference configuration. Thermal deformation share calculated relative to nominal radial clearance of 0.25 mm.

As evident from Table 5, variant 2b represents the most balanced configuration across all evaluated criteria: it achieves a static safety factor above 2.0, limits thermal clearance consumption to 60–70%, and serves as the reference for actuation torque normalisation. Variant 1a offers superior rigidity and lower deformation but at the cost of reduced ring width, which constrains the available contact arc and limits the structural support for the hydraulic drive grooves. Variant 3c, while providing the largest contact area, exhibits the highest stress levels and the greatest thermal clearance consumption (70–80%), approaching the limit beyond which reliable seizure-free operation under transient overload conditions cannot be guaranteed without active thermal compensation. The data in Table 5 therefore support the selection of variant 2b as the recommended design and provide quantitative bounds for engineering judgment when adapting the geometry to alternative compressor sizes or operating conditions.

Based on the comparative analysis summarised in Table 5, variant 2b was selected as the reference design for detailed clearance and drive analysis (Figure 5). Furthermore, technological cutouts in the movement zone of the gate rotor teeth are preserved without a significant increase in the stress concentration factor, confirming the rationality of the selected geometry in terms of strength, manufacturability, and operational reliability.

**Figure 5.** Adjusting ring design variant 2b.

An additional critical design challenge was to justify the minimum permissible operating clearances between the control ring and mating components in the capacity control zone, taking into account thermally induced deformations. Insufficient clearance can lead to

seizure when heated, while excessive clearance can increase leakage and reduce compressor volumetric efficiency.

To accurately estimate the gap sizes, sequential thermal-structural modeling was implemented in the ANSYS environment. In the first stage, the steady-state temperature field was calculated using the Steady-State Thermal module (Figure 6a,b). The boundary conditions for the thermal analysis were defined by assigning a temperature of $t_{suc} = 22\text{ }^{\circ}\text{C}$ to the end-face plane of the screw on the suction side and $t_{dis} = 120\text{ }^{\circ}\text{C}$ to the surface of the discharge port edges. These temperature values represent typical operating conditions of a high-pressure screw compressor and were selected to reproduce the expected thermal gradient between the suction and discharge regions.

The resulting temperature distribution $T(x, y, z)$ was subsequently imported into the Static Structural module to evaluate thermally induced deformations (Figure 6c,d). The thermal strain component was defined as follows:

$$\varepsilon_{th} = \alpha(T - T_{ref}), \quad (6)$$

where α —coefficient of linear thermal expansion, $^{\circ}\text{C}^{-1}$; T —local temperature at a point in the body, $^{\circ}\text{C}$; and T_{ref} —reference (stress-free assembly) temperature at which thermal stresses are zero, $^{\circ}\text{C}$.

This approach ensures consistent coupling between the steady-state thermal field and the structural response, allowing accurate prediction of temperature-driven dimensional changes and their influence on clearance conditions.

Figure 6 presents the coupled thermo-structural response of the screw compressor housing. The steady-state thermal analysis reveals a maximum temperature of approximately $121\text{ }^{\circ}\text{C}$ in the discharge region, while the suction-side surfaces remain near the imposed boundary condition of $22\text{ }^{\circ}\text{C}$. This results in a temperature gradient exceeding $95\text{ }^{\circ}\text{C}$ across the housing wall thickness in the discharge sector.

The subsequent structural analysis indicates that the maximum thermally induced total deformation reaches approximately $0.17\text{--}0.18\text{ mm}$ and is localized near the discharge window and adjacent wall segments. In contrast, deformation in the suction-side region does not exceed $0.05\text{--}0.07\text{ mm}$. Thus, the displacement amplitude in the discharge sector is approximately 2.5–3 times higher than in the low-temperature region, confirming strong thermal asymmetry.

It should be noted that the adopted discharge-side boundary temperature of $120\text{ }^{\circ}\text{C}$ represents a nominal operating condition within the typical range of $100\text{--}130\text{ }^{\circ}\text{C}$ for high-pressure oil-flooded single-screw compressors [8]. At discharge temperatures approaching $130\text{--}150\text{ }^{\circ}\text{C}$, which may occur under elevated pressure ratios or transient overload regimes, the thermally induced deformation would increase proportionally, further consuming the available clearance margin beyond the 60–70% identified under nominal conditions.

The engineering significance of the 60–70% clearance consumption by thermal deformation can be further quantified by considering its effect on internal leakage and volumetric efficiency. For single-screw compressors operating with water lubrication, the practical optimum balancing leakage losses against friction power consumption is typically in the range of $0.03\text{--}0.06\text{ mm}$ [22,23]. The present results indicate that under nominal thermal loading, the available clearance in the discharge sector is reduced from a nominal $0.20\text{--}0.30\text{ mm}$ to an effective value of $0.06\text{--}0.12\text{ mm}$. While this residual clearance remains above the lower bound of the optimal range, any additional contribution from manufacturing tolerances, assembly misalignment, or transient thermal overload (discharge temperatures $130\text{--}150\text{ }^{\circ}\text{C}$) could reduce the effective clearance below 0.03 mm , entering the regime where friction power consumption rises sharply due to asperity contact. This analysis identifies a critical

design requirement: the nominal clearance at assembly must be set with explicit thermal compensation, such that the hot running clearance in the discharge sector remains within the target range of 0.05–0.10 mm under all anticipated operating conditions. The present thermo-structural framework provides the quantitative basis for this clearance assignment.

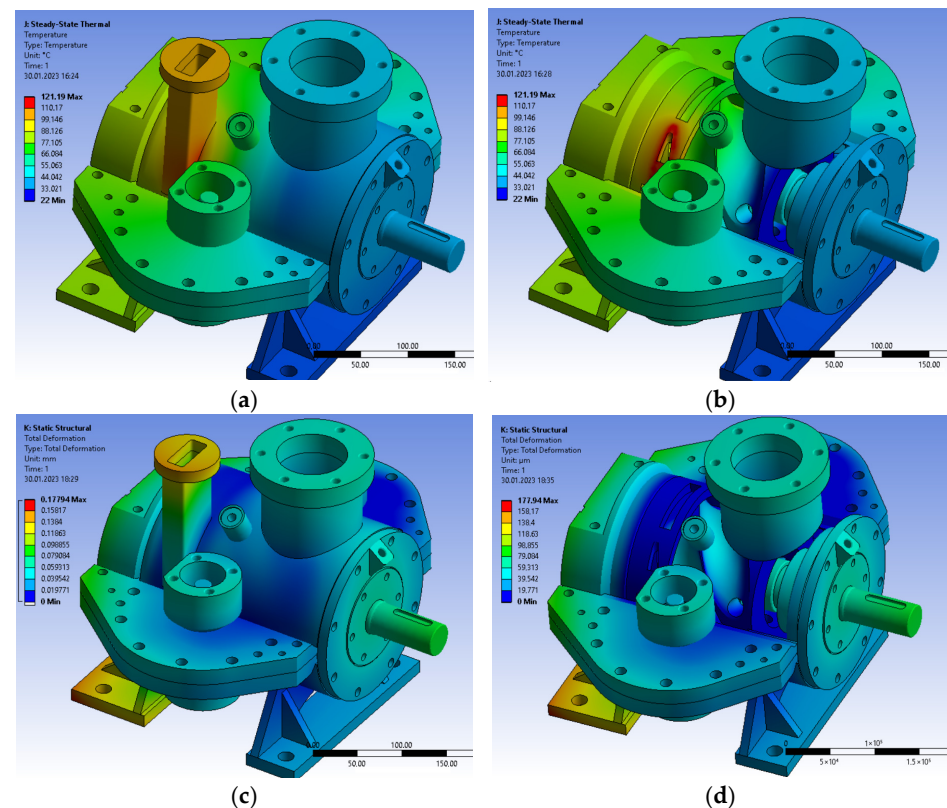


Figure 6. Steady-state temperature field and thermally induced deformation of the screw compressor housing obtained in ANSYS: (a,b) temperature distribution; (c,d) total thermal deformation; (a,c) external surface; and (b,d) housing with removed cover.

Furthermore, the axial displacement gradient of 0.08–0.10 mm across the housing face introduces a secondary leakage path along the axial interface between the regulating ring end face and the housing shoulder. In standard clearance design practice, axial face clearances are typically set in the range of 0.02–0.05 mm; the predicted axial distortion therefore represents 2–5 times the nominal axial clearance, confirming that axial thermal compensation is as critical as radial clearance control in this class of mechanism.

From an operational standpoint, thermal expansion consuming 60–70% of the nominal clearance represents a critical design constraint: without explicit thermal compensation at the assembly stage, the effective clearance in the discharge sector may fall below the safe operating limit under high-load or transient conditions. Neglecting thermal loading in structural assessment would underestimate peak thermally induced structural deformation of the ring by 12–15%, with direct consequences for seizure safety margins and friction torque prediction. The discharge-side region therefore represents the governing zone for clearance design, alignment stability, and long-term operational reliability, confirming that integrated thermo-structural modeling is essential for this class of mechanism.

The thermo-structural results presented above establish the geometric and thermal boundary conditions that govern ring actuation. The following section develops the analytical model for friction torque and required driving force based on these findings.

4. Analytical Model for Friction Torque and Actuation Force

4.1. Compressor Control System and Drive Concept

The layout of the main operating components is shown in Figure 7, and the volumetric capacity control system is illustrated schematically in Figure 8.

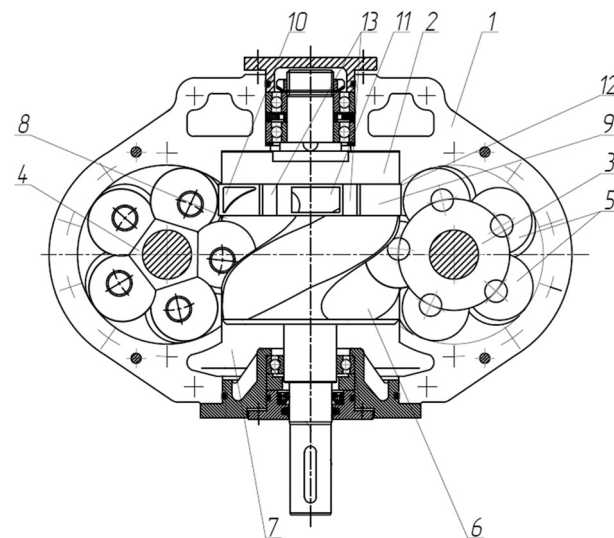


Figure 7. Schematic cross-section of the single-rotor screw compressor with housing cover removed. Component key: 1—casing; 2—helical screw rotor; 3, 4—toothed gate rotors (swash plates); 5—gate rotor tooth profile; 6—rotor helical groove; 7—suction chamber; 8—discharge chamber; 9—rotary adjusting ring; 10—discharge ports; 11—bypass ports; 12—annular bore; 13—V-grooves in ring body; and 14—oil supply holes.

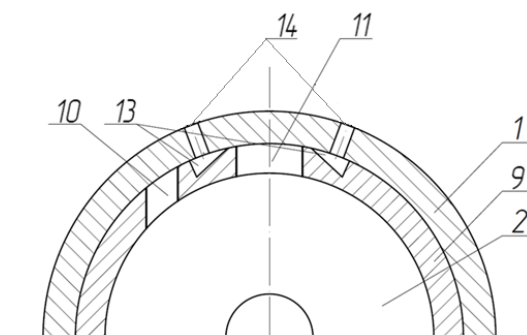


Figure 8. Schematic diagram of the compressor volumetric capacity control system (component numbering as in Figure 7). Pressurized oil admitted through holes 14 acts on V-grooves 13, generating a tangential force that rotates adjusting ring 9 and varies the effective discharge port opening moment, thereby regulating compressor capacity.

The helical screw rotor meshes with two symmetrically arranged toothed gate rotors within the compressor casing, which contains dedicated suction and discharge chambers. Capacity control is achieved by rotating the adjustable ring, which modifies the effective discharge port opening. Actuation is provided hydraulically: pressurized oil supplied through ports in the casing acts on V-grooves machined into the ring body, generating the tangential force required for controlled angular displacement. Component identifications are provided in the figure captions.

4.2. Force Equilibrium and Actuation Model

At this stage, a calculation model was developed to determine the force applied to the edge of the wedge groove, required to rotate the adjusting ring under actual compressor operating conditions (Figure 9). Based on the constructed force model, an analytical equation

was derived that allows for determining the required drive force, taking into account the current gas loads, friction forces, and the geometric parameters of the wedge mechanism.

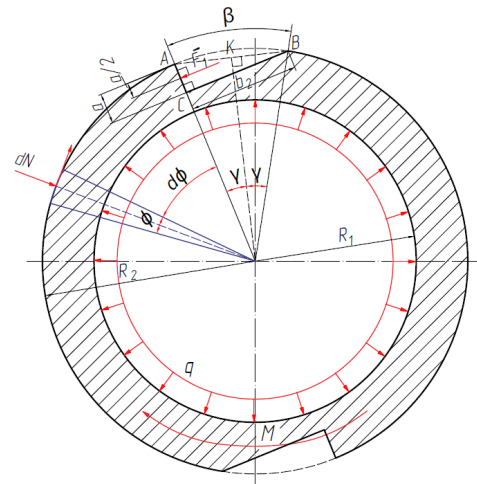


Figure 9. Scheme for calculating F_1 and for rotating the regulator by an angle α .

Let us consider the equilibrium of the regulating ring. The ring is subjected to a concentrated force $\vec{F}_1$, friction forces arising at the interface between the ring and the outer cylindrical surface of the groove, and the corresponding normal reaction distributed along the major portion of the circumference with radius R_2 .

The normal reaction may be modeled as a uniformly distributed load with intensity q acting over the effective contact arc. This distributed pressure generates both radial support and a tangential friction component opposing the rotational motion of the ring. The resulting frictional resistance must be overcome by the applied force $\vec{F}_1$ to ensure controlled angular displacement under operating conditions.

To determine the force required to rotate the regulating ring under operating conditions, the static equilibrium of the ring is considered. The ring is subjected to:

- a concentrated driving force F_1 ,
- distributed normal reaction along the cylindrical contact surface of radius R_2 ,
- corresponding distributed friction forces.

Since the ring performs rotational motion about axis O , perpendicular to the plane of analysis, friction forces are conveniently represented by their resultant torque M .

The moment equilibrium equation about point O is written as follows:

$$\sum M_O = 0 \quad (7)$$

$$F_1 \left(R_2 - \frac{a}{2} \right) - M = 0 \quad (8)$$

where F_1 —concentrated driving (actuation) force applied at the groove edge, N; R_2 —outer (contact) radius of the regulating ring, m; a —width of the V-groove—the geometric offset of the force application point from the contact surface, m; M —resultant friction torque opposing ring rotation, N·m; and $R_2 - a/2$ —effective lever arm of force F_1 about axis O , m.

Hence, the required actuation force is as follows:

$$F_1 = \frac{M}{R_2 - a/2} \quad (9)$$

where M —total friction torque to be overcome, N·m; R_2 —outer contact radius, m; and a —characterizes the geometric offset of the force application point relative to the contact radius.

Assuming uniform contact pressure intensity q over the effective contact arc (excluding the cut-out region), the elementary arc length is as follows:

$$dS = R_2 d\varphi \quad (10)$$

where R_2 —outer contact radius of the ring, m, and $d\varphi$ —infinitesimal angular increment of the arc coordinate, rad.

The elementary normal force:

$$dN = qR_2 d\varphi \quad (11)$$

where q —uniformly distributed contact pressure intensity per unit arc length, $\text{N}\cdot\text{m}^{-1}$; R_2 —outer contact radius, m; and $d\varphi$ —angular increment of arc coordinate, rad.

The corresponding elementary friction force:

$$dF = f q R_2 d\varphi \quad (12)$$

where f is the coefficient of friction.

The elementary torque contribution:

$$dM = R_2 dF = f q R_2^2 d\varphi \quad (13)$$

Integrating over the effective contact angle $\pi - \beta$:

$$M = \int_0^{\pi-\beta} f q R_2^2 d\varphi = f q R_2^2 (\pi - \beta) \quad (14)$$

where β —central angle of the cut-out sector (contact interruption), rad, and $\pi - \beta$ —effective contact arc angle, rad.

The contact is interrupted by a cut-out sector with central angle β . From geometric relations,

$$\gamma = \arcsin\left(\frac{\sqrt{a^2 + b_2^2}}{2R_2}\right) \quad (15)$$

$$\beta = 2\gamma = 2\arcsin\left(\frac{\sqrt{a^2 + b_2^2}}{2R_2}\right) \quad (16)$$

Substituting into (9)

$$M = f q R_2^2 \left(\pi - 2\arcsin\left(\frac{\sqrt{a^2 + b_2^2}}{2R_2}\right) \right) \quad (17)$$

Substituting Equation (17) into Equation (9), the required driving force is obtained:

$$F_1 = \frac{f q R_2^2 \left(\pi - 2\arcsin\left(\frac{\sqrt{a^2 + b_2^2}}{2R_2}\right) \right)}{R_2 - \frac{a}{2}} \quad (18)$$

The mechanical work required to rotate the ring through angle α is as follows:

$$W = \int_0^\alpha M d\varphi = f q R_2^2 \left(\pi - 2\arcsin\left(\frac{\sqrt{a^2 + b_2^2}}{2R_2}\right) \right) \alpha \quad (19)$$

where F_1 —required driving (actuation) force, N; M —friction torque (from Equation (17)); and $R_2 - a/2$ —effective lever arm of the driving force, m.

4.3. Contact Pressure Estimation and Model Uncertainty

The friction coefficient f in the above expressions corresponds to the Coulomb friction coefficient μ introduced in Section 2.3. For the nominal lubricated steel-on-steel contact condition of the regulating ring assembly, $f = 0.08$ – 0.12 is adopted as the design range, with $f = 0.10$ used as the reference value [34,35]. The contact pressure intensity q is related to the gas loading as follows:

$$q \approx \frac{p_{\text{gas}} \cdot 2R_1 \cdot b}{R_2 \cdot \varphi_{\text{eff}}}, \quad (20)$$

where R_1 is the inner radius of the ring (acting gas pressure surface), R_2 is the outer contact radius, b is the ring width, and $\varphi_{\text{eff}} = \pi - \beta$ is the effective contact arc angle. For a thin-walled ring ($R_1 \approx R_2$), this reduces to $q \approx 2p_{\text{gas}} \cdot b / \varphi_{\text{eff}}$. For the design-point operating pressure of 1.0–1.5 MPa and the geometric parameters of variant 2b (Table 1), this yields q in the range of 0.6–1.8 MN/m, confirming that frictional torque and driving force are strongly dependent on operating pressure and ring geometry.

It is acknowledged that both μ and q carry uncertainty in practice: μ depends on lubrication regime, surface finish, and temperature, while q depends on instantaneous gas pressure and thermal deformation state. In addition, the assumption of uniform contact pressure distribution introduces a modeling error that is quantified below. The analytical expressions derived here provide closed-form sensitivity relationships (Equations (12), (17) and (18)) that enable direct quantification of these uncertainties in drive force requirements, which is the primary engineering utility of the model.

For the configurations studied, the ratio of maximum thermally induced radial deformation (0.17–0.18 mm) to the ring outer radius R_2 (≈ 125 mm) is approximately 0.14%, which is consistent with the small-deformation regime ($\delta/R_2 \ll 1$) in which the uniform contact pressure approximation is conventionally adopted for thin-walled cylindrical interfaces [37].

Nevertheless, the finite element results indicate that thermal asymmetry between the suction and discharge sectors produces a non-uniform deformation field, with peak displacement in the discharge zone exceeding that in the suction zone by a factor of 2.5–3. This spatial non-uniformity implies that the actual contact pressure distribution is higher in the discharge sector and lower near the suction port. To estimate the resulting error in the analytical torque prediction, consider a first-order correction in which the discharge-sector contact pressure is elevated by a factor k_d relative to the mean value q , and the suction-sector pressure is correspondingly reduced. For the computed deformation asymmetry ratio of 2.5–3, and assuming a linear relationship between local deformation and contact pressure increment, the effective torque integral deviates from the uniform-pressure result by approximately 8–14%. This range represents the estimated upper bound of the modeling error introduced by the uniform-pressure assumption under the thermal conditions studied. For more severe operating regimes—such as discharge temperatures approaching 130–150 °C under elevated pressure ratios—the non-uniformity would increase proportionally, potentially raising the torque prediction error toward 15–20%. In such cases, a piecewise non-uniform pressure distribution or a direct FEA-based contact pressure extraction should be used in place of the analytical model.

The uniform-pressure model therefore provides a conservative and practically useful estimate at the preliminary design stage, where rapid parametric sensitivity studies are required prior to detailed nonlinear FEA. Its use is appropriate provided the deformation asymmetry ratio remains below approximately 3:1 and the ring geometry falls within the

thin-walled regime ($h/R_2 < 0.15$). For the nine variants studied ($h = 5\text{--}15\text{ mm}$, $R_2 \approx 125\text{ mm}$, $h/R_2 = 0.04\text{--}0.12$), this condition is satisfied. The analytical expressions for friction torque (Equation (17)) and required driving force (Equation (18)) should therefore be interpreted as design-point estimates with an uncertainty band of $\pm 10\text{--}15\%$ attributable to the contact pressure non-uniformity identified above.

4.4. Engineering Implications for Drive System Design

The derived analytical model reveals several important engineering dependencies.

The quadratic dependence of friction torque on contact radius R_2 ($M \propto R_2^2$) has a direct consequence for the scaling of the hydraulic drive system. If the compressor geometry is scaled up by a linear factor λ (e.g., to increase throughput capacity), the friction torque increases as λ^2 , while the available hydraulic force from the V-groove drive scales only as λ (proportional to groove projected area). This means that the required oil supply pressure must also increase linearly with scale factor λ , imposing a fundamental hydraulic design constraint that limits the applicability of the wedge-groove drive concept to compressors below a certain size threshold. For the geometry of variant 2b ($R_2 \approx 125\text{ mm}$), the present results define the baseline torque demand; extrapolation to larger machines requires re-optimisation of groove geometry and hydraulic circuit pressure.

The linear dependence of torque on contact pressure intensity q , and the direct coupling of q to thermally induced deformation identified in the FEA results, establishes a feedback loop between thermal state and actuation demand that has important implications for control system design. As discharge temperature increases during high-load operation, thermal expansion of the ring increases the contact pressure at the cylindrical interface, which in turn raises the friction torque and the required oil pressure for ring actuation. If the hydraulic supply pressure is fixed, this feedback can cause the ring to become immobile at elevated temperatures—a failure mode that would prevent capacity regulation precisely when it is most needed. The present analytical model quantifies this coupling and enables the hydraulic circuit designer to specify a pressure margin sufficient to overcome the thermally augmented torque demand across the full operating temperature range.

The sensitivity of torque to the effective contact angle ($\pi - \beta$) provides a direct geometric design lever for torque reduction. Increasing the cut-out angle β by 10° (0.175 rad) reduces the effective contact arc by the same amount, producing a proportional reduction in friction torque of approximately $10^\circ/180^\circ \times 100\% \approx 5.6\%$ per 10° of additional cut-out. However, the structural penalty must be evaluated simultaneously: a larger cut-out reduces the load-bearing arc, increasing the peak contact pressure on the remaining arc and potentially raising localised stress concentration. The optimum cut-out geometry therefore lies at the intersection of the torque-reduction curve and the stress concentration constraint, which for the present configurations occurs near $\beta \approx 30\text{--}40^\circ$ based on the combined FEA and analytical results.

Additionally, the mechanical work of friction grows linearly with rotation angle α , meaning that large regulation ranges significantly increase actuation energy demand and oil pressure requirements in hydraulic drive systems.

From a design perspective, the model provides a direct analytical relationship between structural parameters and required oil-driven actuation force. This enables the prediction of required hydraulic pressure, sensitivity analysis with respect to friction coefficient, optimization of groove geometry, and estimation of drive system power demand. The analytical expressions are especially valuable at the preliminary design stage, where rapid parameter studies are required before detailed nonlinear FEA.

5. Conclusions

This study presents an integrated thermo-mechanical and analytical investigation of an adjustable regulating ring for a high-pressure single-rotor screw compressor. Coupled Steady-State Thermal and Static Structural finite element modeling in ANSYS was combined with a closed-form friction torque model to evaluate structural integrity, clearance stability, and actuation performance across nine geometric configurations.

The thermo-structural analysis demonstrated that asymmetric thermal loading between the suction (22 °C) and discharge (120 °C) zones produces peak thermally induced deformation of 0.17–0.18 mm in the discharge sector, consuming 60–70% of nominal operating clearance (Figure 6). Neglecting thermal effects underestimates peak thermally induced structural deformation of the regulating ring by 12–15%, with direct consequences for seizure safety margins. The axial displacement gradient (0.08–0.10 mm) additionally represents 2–5 times the nominal axial face clearance, confirming that both radial and axial thermal compensation must be addressed at the design stage.

Among the configurations evaluated, variant 2b ($b = 47$ mm, $h = 10$ mm) provides the most balanced structural response: peak equivalent stress is 12–15% lower than in the least favourable configuration, total deformation is 18–25% below the most flexible variant, and required actuation torque is reduced by 8–11% relative to wider configurations. These advantages are quantitatively summarised in Table 5.

The analytical friction torque model reveals quadratic sensitivity of torque demand to contact radius ($M \propto R_2^2$), linear dependence on contact pressure and friction coefficient, and strong dependence on cut-out geometry. The uniform contact pressure assumption introduces an estimated uncertainty of ± 10 –15% under nominal conditions, rising to ± 15 –20% at discharge temperatures of 130–150 °C. The model provides a practical tool for hydraulic drive sizing and groove geometry optimisation prior to detailed nonlinear FEA.

Model reliability is supported by three validation levels: agreement with the Lamé analytical solution within 2.5%, domain-level consistency with published CFD–FEA screw compressor deformation data, and mesh convergence within 1.2%.

It is also acknowledged that direct experimental validation of the coupled thermo-structural model—including strain gauge measurements and optical clearance monitoring on a physical prototype of the regulating ring assembly under controlled operating conditions—was not performed in the present study and constitutes a necessary step toward full experimental corroboration of the finite element results. Future work will focus on experimental validation of the wedge-drive concept, fatigue life assessment under cyclic thermo-mechanical loading, and transient thermal modeling under variable operating regimes.

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Conflicts of Interest: The authors declare no conflicts of interest.

Abbreviations

The following abbreviations are used in this manuscript:

Roman letters—upper case

b	Ring width (axial dimension of the regulating ring), mm
C	Fourth-order elastic stiffness tensor, Pa
dF	Elementary friction force on arc element $d\varphi$, N
dM	Elementary torque contribution from arc element $d\varphi$, N·m
dN	Elementary normal contact force on arc element $d\varphi$, N
dS	Elementary arc length element, m
E	Young's modulus of elasticity, Pa
F_1	Concentrated driving (actuation) force applied to the ring groove edge, N
F_c	Vector of contact reaction forces in the FEA system, N
F_{ext}	Vector of externally applied mechanical loads (gas pressure tractions), N
F_{th}	Vector of thermally induced nodal forces, N
I	Second-order identity tensor
$K(U)$	State-dependent (contact-nonlinear) tangent stiffness matrix, N·m ⁻¹
M	Resultant friction torque acting on the regulating ring, N·m
R_1	Inner radius of the regulating ring, m
R_2	Outer (cylindrical contact) radius of the regulating ring, m
S	Arc length along the cylindrical contact surface, m
T	Local temperature at a point in the body, °C
$T(x,y,z)$	Steady-state temperature distribution field, °C
T_1	Inner surface temperature in the Lamé benchmark (Section 2.7.1), °C
T_2	Outer surface temperature in the Lamé benchmark (Section 2.7.1), °C
T_{ref}	Reference (stress-free assembly) temperature, °C
U	Global nodal displacement vector (FEA), m
W	Mechanical work of friction during ring rotation, J

Roman letters—lower case

a	Width of the V-groove (geometric offset of force application point), m
b_2	Height (depth) of the V-groove cut-out, m
f	Coulomb friction coefficient at the cylindrical contact interface ($f = \mu$)
h	Wall thickness of the regulating ring, mm
p	Applied internal gas pressure, Pa
p_{dis}	Discharge-side gas pressure, Pa
p_{gas}	Operating gas pressure differential ($p_{dis} - p_{suc}$), Pa
p_n	Local normal contact pressure at the interface (Equation (3)), Pa
p_{suc}	Suction-side gas pressure, Pa
q	Uniformly distributed contact pressure intensity per unit arc length, N·m ⁻¹
t_{dis}	Discharge-side temperature boundary condition, °C
t_{suc}	Suction-side temperature boundary condition, °C

Greek letters

α	Coefficient of linear thermal expansion, °C ⁻¹
β	Central angle of the cut-out sector (contact interruption), rad
γ	Half-angle of the cut-out sector, rad
ΔT	Local temperature difference ($T - T_{ref}$), °C
ε	Total strain tensor
ε_{th}	Thermal strain tensor (Equations (2), (5) and (6))
μ	Coulomb friction coefficient (lubricated steel-on-steel, $\mu = 0.10$ adopted)
ν	Poisson's ratio
ρ	Material density, kg·m ⁻³
σ	Cauchy stress tensor, Pa
τ	Tangential (shear) stress at the contact interface (Equation (3)), Pa
φ	Angular coordinate along the cylindrical contact arc, rad

φ_{eff}	Effective contact arc angle: $\varphi_{eff} = \pi - \beta$, rad
φ_{rot}	Rotation angle of the regulating ring (regulation stroke), rad

Abbreviations

ANSYS	Commercial finite element analysis software (Ansys Inc.)
CFD	Computational fluid dynamics
COP	Coefficient of performance
FEA	Finite element analysis
SCU	Screw compressor unit
V&V	Verification and validation

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