Design Optimization for Efficiency and Mechanical Stress Management in High-Speed Screw Spindle Compressors

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ABSTRACT

Pursuing sustainable and environmentally friendly cooling solutions has underscored the significance of exploring alternative refrigerants in various industrial applications. Previous publications showed that high-speed screw spindle compressors are a viable solution for water vapor (R718) applications. This paper introduces a design optimization strategy targeting the efficiency enhancement and mechanical stress mitigation of high-speed screw spindle compressors. As rotational speed increases, so does volume flow and efficiency, but mechanical stress becomes a challenge, especially on the root diameter of the spindle teeth. The research investigates this trade-off and proposes a design that allows higher speeds while alleviating mechanical limitations, increasing its mechanical safety.
The proposed design optimization is validated through analytical calculations and thorough FEM simulations, displaying substantial efficiency improvements and volumetric flow rate increments at elevated speeds. The observed outcomes underline the approach's potential in facilitating efficient, high-speed screw spindle compressors.

Keywords: R718, screw spindle compressor, FEM, mechanical stress

1. INTRODUCTION

R-718 (water) as a refrigerant presents a compelling alternative to refrigeration technology. It is non-flammable, has no ozone depletion potential (ODP) and minimal global warming potential (GWP = 5×10^4), which aligns with global environmental goals. Due to its high temperature range (> 200 °C) the field of application is theoretically tremendous. However, challenges of an R-718 refrigeration cycle persist, including the requirement to operate below atmospheric pressure and the need for high suction volume flows and high-pressure ratios, which introduce design complexities. The investigations of Moesch *et al.* (2019, 2022b) show that screw spindle vacuum pumps may be a feasible solution for high pressure ratios below atmospheric pressure, however the volume flows of vacuum pumps are rather limited. Moesch *et al.* (2022a) introduced a conical screw spindle design for theoretical volume flows between 10,000 m³/h and 11,000 m³/h at 16,000 rpm which corresponds to 45 kW cooling capacity at 5 °C evaporation temperature according to the results of the thermodynamic model (Moesch *et al.*, 2023). This design uses liquid injection and has a volume index of 11.3, which leads to a temperature lift from 5°C to 50°C (corresponds to $\pi = 14.2$) and a discharge superheat of only 5 K (Moesch *et al.*, 2023).
Higher suction volumes can be achieved by designing a bigger screw spindle or faster rotating speeds. A bigger screw spindle will increase the manufacturing costs, which is the reason why high rotational speeds are in this case preferred. However, the high rotational speed places significant mechanical stress on the compressor's rotating spindle, raising concerns about the component's fatigue. This paper focusses on the mechanical stress of the rotating spindle, which is quantified using the finite element method (FEM). Furthermore, an analysis and characterization of stress distribution

within the spindle structure was conducted and different stress reducing methods were compared. The results show that the stresses may be reduced significantly when using different materials or introducing design changes.

2. SCREW SPINDLE ROTOR GEOMETRY

The investigated screw spindle geometry is shown in **Figure 1**. The rotor has a variable lead, a variable crown diameter, and it is based on a cycloid profile by Rofall and Steffens (2011) as described by Moesch *et al*. (2022). The crown diameter is reduced linearly over the rotor length as shown in Eq. (1):

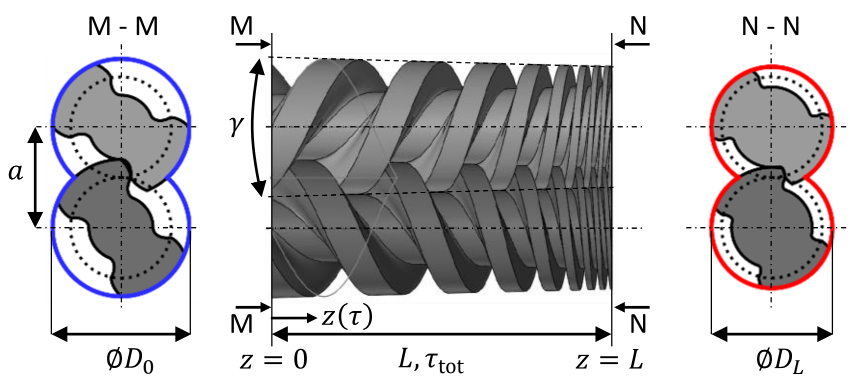
$$D(\tau)=D_0\text{-}2\cdot z(\tau)\cdot\tan(\gamma/2) \tag{1}$$

where $z(\tau)$ is the helix angle-dependent axial position in m. The axial position $z(\tau)$ is defined by a combined polynomial and trigonometric function with the coefficients $c_1 \ldots c_7$ as shown in Eq. (2).

$$z(\tau)=\frac{1}{2\pi}\left[c_1\tau^4+c_2\tau^3+c_3\tau^2+c_4\tau+c_5+c_6\cos(c_7\tau)\right] \tag{2}$$

with $c_1=2.669\times10^{-6}$; $c_2=-2.701\times10^{-4}$; $c_3=-7.304\times10^{-4}$;
$c_4=4.223\times10^{-1}$; $c_5=-1.426$; $c_6=1.426$, $c_7=1.241\times10^{-1}$

The rotor geometry leads to a built-in volume ratio V_i=11.3 and theoretical displacement volume V_{disp}=11.5 dm³ ($\dot{V}_{disp}$ = 11,039 m³/h at 16,000 rpm).



$\boldsymbol{a}$	195.0	mm
$\boldsymbol{D_0}$	265.5	mm
$\boldsymbol{D_L}$	234.1	mm
$\boldsymbol{L}$	650	mm
$\boldsymbol{\gamma}$	2.769	°
$\boldsymbol{\tau_{tot}}$	1450	°

Figure 1: Main screw spindle rotor geometry parameters and their values (right)

3. MATERIAL SELECTION BASED ON CRITICAL GEOMETRY NOTCHES

In this section, a material case study is conducted to investigate the influence of the material selection on the spindle performance subjected to high rotational speed.

Figure 2 shows the simulation model. The spindle is placed between two concentric shafts. The locating bearing is fixed on the left shaft, ensuring that if the cone-shaped spindle expands due to temperature changes, it expands to the right, where there is more clearance, avoiding contact between the spindle and housing and thus avoiding a failure of the machine.

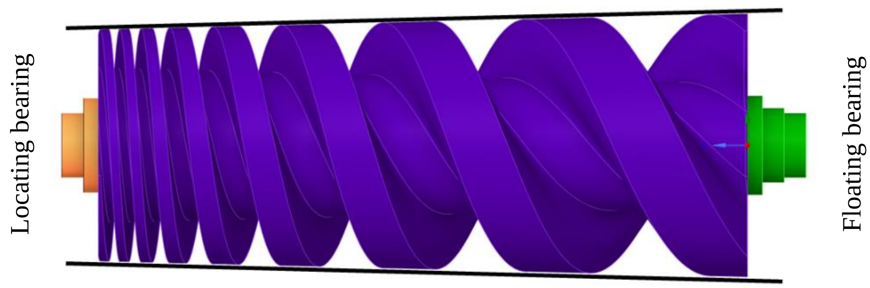


Figure 2: Screw spindle model showing locating and floating bearing positions.

Specifically, three common materials are considered: steel, aluminum, and titanium alloys. Each has distinct mechanical properties and thermal behaviors.

- **Steel, 30CrNiMo8:** Steel is a commonly used material with high strength and good wear resistance. However, it is dense compared to the other alloys used in this case study. The chosen alloy has a relatively high yield strength and is corrosion-resistant.
- **Aluminum alloy, AlZn5, 5MgCu:** Also known as EN AW-7075, is lightweight, and has a good strength-to-weight ratio. It is one of the strongest aluminum alloys that is available. A downside of aluminum alloys is generally that they have a large coefficient of thermal expansion (CTE), discussed in Section 4.
- **Titanium, Ti-6Al-4V:** This alloy exhibits a high strength-to-weight ratio and corrosion resistance, making it attractive for high-speed rotational applications.

The screw spindle geometry remains constant across the case study, allowing for a direct comparison of the materials' responses under identical boundary conditions. The relevant material properties are summarized in **Figure 3**.

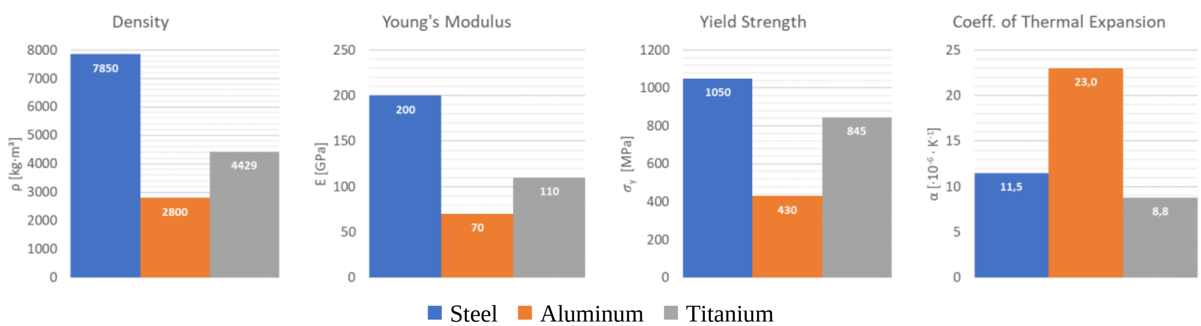


Figure 3: Relevant material properties for steel, aluminum, and titanium

Higher rotational speeds lead to higher centrifugal forces and, consequently, higher mechanical stresses within the spindle. As the spindle rotates rapidly, centrifugal forces act on its teeth, pulling them outward. The farther away the tooth's center of mass is from the axis of rotation, the higher the centrifugal force it experiences.

This outward force leads to deformation, particularly in bending the spindle teeth, if the material is not sufficiently robust or centrifugal forces are too high. Stress concentration can lead to localized yielding or fatigue failure, particularly in regions where the tooth profile undergoes sudden changes, such as notches.

Figure 4 shows the simulation results for an aluminum spindle with an inner diameter of 60 mm and the critical notches**.**

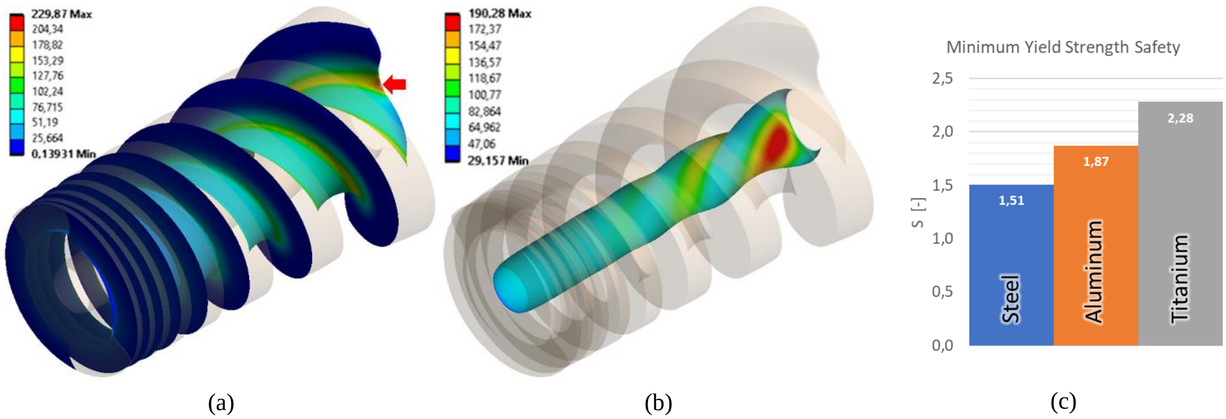


Figure 4: Stress distribution in $\boldsymbol{MPa}$ on the spindle root circle diameter (a) and inner diameter (b) and the minimum yield strength safety (c)

Notches act as stress raisers, making the spindle more susceptible to fatigue failure. The presence of notches accelerates the initiation and propagation of fatigue cracks, reducing the fatigue life of the shaft.
Figure 4 (a) shows the stress distribution at the root circle diameter of the spindle. The spindle teeth are the biggest (in thickness and height) on the suction side and the smallest on the discharge side. The spindle's mass is concentrated towards the outer rim, with less mass towards the center. Therefore, the centrifugal force acting on the mass will be highest at the outer rim of the spindle. Thus, the highest stresses occur at the root circle diameter at the suction side.
Figure 4 (b) shows the stress distribution in the hollow part of the spindle, the inner diameter. The mass of the teeth pulls on the inner diameter to the extent that it deforms into the wiggly shape shown exaggeratedly in the **Figure 4 (b)**.
The yield strength safety shown in **Figure 4** (c) is calculated with the following equation:

$$S = \frac{\sigma_y}{\sigma_{max}} \tag{3}$$

where σ_y is the yield strength and σ_{max} is the maximum stress occurring on the spindle.
After considering various materials for constructing the spindle compressor, titanium emerged as the material with the highest safety factor due to its high strength-to-weight ratio. However, due to its costs, it was deemed economically unfeasible for this application. Consequently, aluminum was chosen as the alternative material, balancing performance and cost-effectiveness. While not possessing the same level of safety as titanium, aluminum stood as the next best option.
Another case study was done on the aluminum screw spindle, varying its inner diameter. While it is thermodynamically advantageous to have a hollow screw spindle with a higher inner diameter to cool the spindle, the results of the case study in **Figure 5** show that the maximum stress on the screw spindle increases, thus decreasing its fatigue safety. An inner diameter of 20 mm has been chosen for this application.

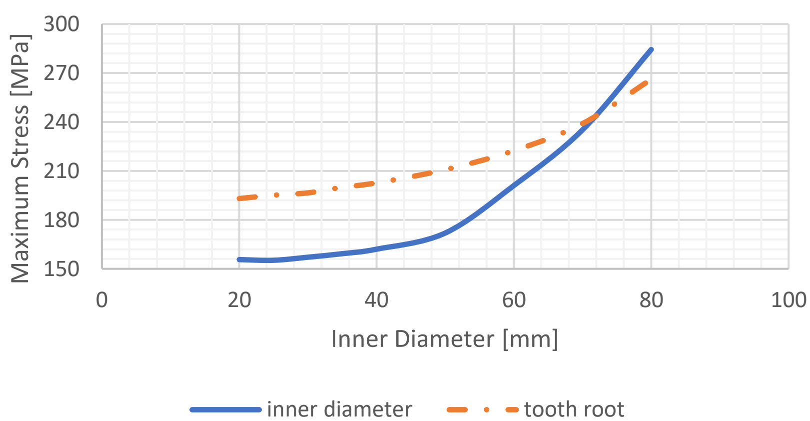


Figure 5: Maximum Stress on screw spindle at the inner diameter and the root circle diameter

4. THERMAL EXPANSION

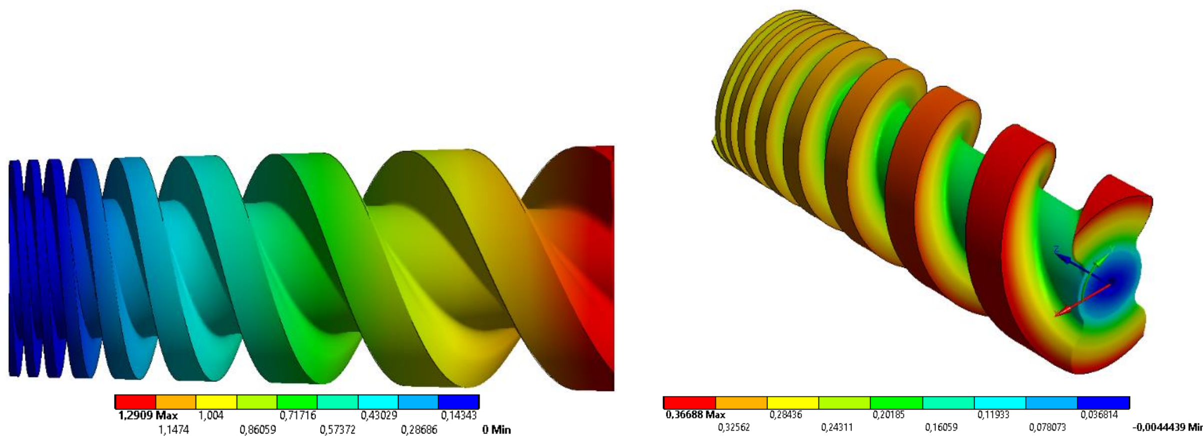
The clearances between the two rotors and between the rotors and the compressor casing are decisive to the functioning and effectiveness of screw compressors. These clearances impact the compressor's volumetric efficiency, leakage losses, and overall performance.

Tight clearances minimize leakage losses but may increase mechanical friction and the risk of rotor contact. Inadequate clearance management can accelerate wear and tear on compressor components, leading to increased maintenance requirements and potential breakdowns.

Conversely, excessive clearances reduce mechanical losses but can compromise volumetric efficiency. They disrupt the sealing mechanism between the rotors and allow gas to bypass compression stages, thereby reducing the overall compression ratio and efficiency and increasing leakage losses, especially for oil-free compressors.

As mentioned in Section 3, aluminum has a relatively high coefficient of thermal expansion. It also possesses a high thermal conductivity. The water vapor at the discharge is around 120 ℃. The worst-case scenario is that the entire aluminum spindle heats up with a $\Delta T = 100$ ℃.

Figure 5 shows the results of the thermal finite element analysis. Due to its length and the increase in temperature, the spindle will expand axially by 1.14 mm and radially up to 0.37 mm, particularly at the suction side with the highest diameter. This expansion must be considered when manufacturing the spindles. The screw spindles' tip diameter shall be slightly smaller to avoid a machine failure.



(a) (b)

Figure 6: Axial (a) and radial (b) thermal expansion of the aluminum screw spindle in mm

5. REDUCTION OF STRESS WITH GEOMETRY OPTIMIZATION

To reduce the stresses on the spindle, several approaches can be considered. The primary cause of significant stress at the foot diameter of the spindle teeth is the centrifugal force generated by the high rotational speed, which pulls the teeth away from the spindle body, thereby inducing tensile stress.

One method to mitigate these stresses involves preventing the teeth from being pulled away. This can be achieved by employing a 3D-printed spindle with hollow teeth, which reduces the centripetal force acting on them. However, this approach is expensive.

Another strategy involves using an expansion screw to pull the teeth together at both ends, thereby preventing them from expanding and further stressing the spindle body. However, this solution imposes design constraints on the rotating unit, as it would obstruct the hollow diameter.

This paper presents and analyzes two additional feasible solutions to address the issue. The designs are shown in **Figure 7**. In Design A, the spindle is shown with material anchoring on its suction side, adding more material to the notch, smoothing the notch and increasing the spindle’s stiffness. In Design B, the tooth's mass is reduced through holes in the tip circle diameter, thus reducing centrifugal force pulling on the tooth root diameter.

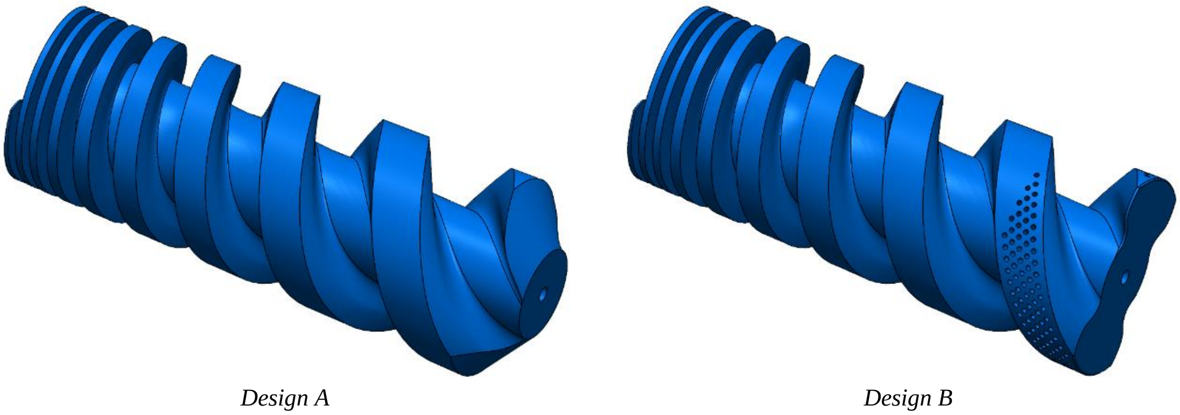


Figure 7: Alternative screw spindle designs:
Design A: material anchoring. *Design B*: mass reduction at tip circle diameter.

The boundary conditions of the analysis were not changed. The results of the FEM are shown in **Figure 8**. The case study has been conducted for the original screw spindle mentioned at the end of section 3 and the same screw spindle but with material anchoring (*Design A*) at the suction side and with a reduced mass at the tip circle diameter (*Design B*).

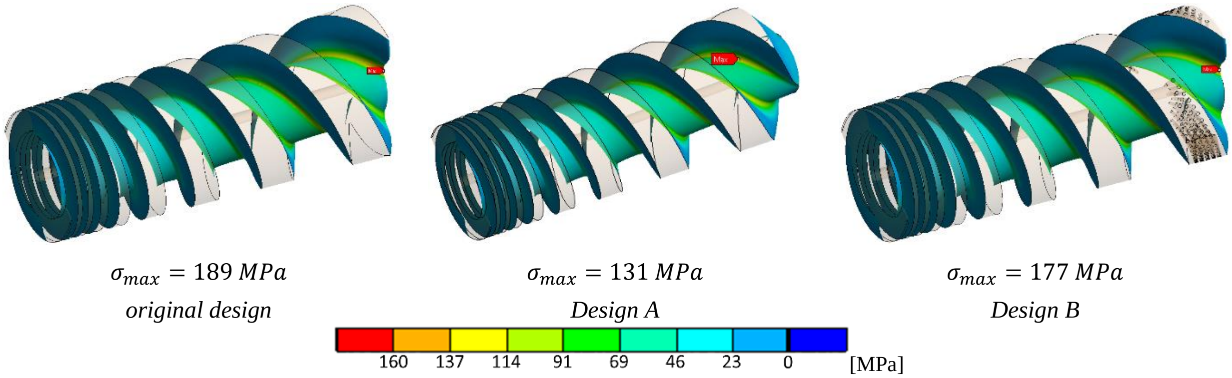


Figure 8: Stress Results of the FEM-Analysis for the original screw spindle on Design A and Design B

The results show that the stress was significantly reduced by ca. 30 % on the root circle diameter for **Design A**. The structural reinforcement helped redistribute stresses and enhanced its stiffness. The mass reduction on **Design B** had a negligible effect on the stress distribution on the root diameter, however, it did reduce the centrifugal force pulling on the inner diameter of the screw spindle. The maximum stress on the root circle diameter and the inner diameter of the screw spindle for all of the three designs are shown in **Figure 9**.
Not only is *Design A* the cheaper option due to the smaller effort in manufacturing, but it also emerged as the better notch mitigating design, demonstrating less stress on the root circle diameter. While Design B reduced the stress at the inner diameter, the overall maximum stress in Design B is higher than in Design A.

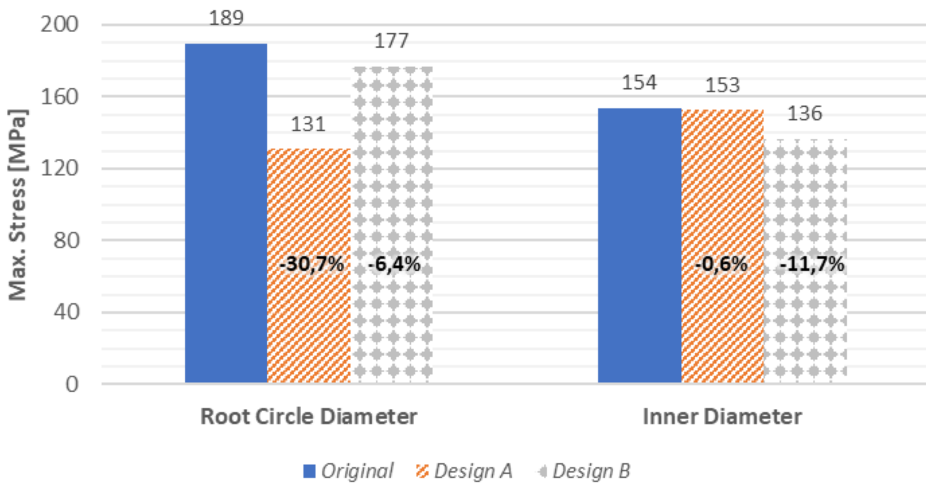


Figure 9: Comparison of maximum stresses on the root circle diameter and inner diameter of the screw spindle for all of the three designs

6. CONCLUSIONS

In conclusion, the study has delved into the details of screw spindle compressors for R-718 in refrigeration/ heat pump appliances, emphasizing the necessity for high volume flows and consequent challenges that arise with increased rotational speeds. Despite these challenges, the pursuit of efficiency and safety remains paramount in the design process.

The exploration of materials revealed titanium as the optimal choice for ensuring maximum safety against fatigue; however, its prohibitive cost necessitated a realistic shift towards the utilization of an aluminum alloy. While aluminum may not offer the same level of safety as titanium, the design compensates through careful attention to detail and optimization strategies, ensuring a balance between performance, cost-effectiveness, and mechanical robustness. Aluminum comes with its high coefficient of thermal expansion as a downside that causes the screw spindle to expand significantly. This effect on the clearances shall be compensated when manufacturing the screw spindles.

Furthermore, selecting the compressor's inner diameter is a critical parameter in stress mitigation. By minimizing the diameter, the study aims to alleviate potential mechanical vulnerabilities. This approach underscores a proactive stance towards preempting stress concentrations and structural weaknesses, thus enhancing the compressor's longevity and reliability.

Moreover, two designs are introduced and analyzed through Finite Element Analysis (FEA) to reduce the mechanical stress on the root circle diameter. *Design A* has material anchoring at the suction side and Design B has holes at the tip circle diameter. *Design A* emerges as the more successful option, exhibiting lesser stresses on the screw spindle under operational conditions. The material anchoring offered more support at the critical notch and changed its structure, reducing the maximum stress at the root circle diameter by ca. 30 %. Meanwhile, Design B, which slightly reduced the centrifugal force, reduced the stress in the same region by about 6 %.

This research encapsulates the complex interplay between design constraints, material considerations, and structural optimization in screw spindle compressor technology. Through thorough analysis and innovative design strategies, it addresses the challenges associated with high-speed rotation and presents a solution for enabling the screw spindle to rotate in higher speeds.

NOMENCLATURE

Symbols:

Symbol	**Meaning**	**Unit**
a	rotor center distance	m
$c_1 \dots c_7$	coefficients of lead function	-
D	crown diameter	m
E	Young's Modulus	GPa
L	rotor length	m
P_i	indicated power	W
$\dot{Q}_0$	cooling capacity	W
V	volume	m^3
$\dot{V}$	volume flow	m^3/h
z	axial rotor position	m
ΔT	Temperature difference	℃
α	coefficient of thermal expansion	K^{-1}
γ	rotor cone angle	rad
η	efficiency	-
ϑ	temperature	℃
ρ	density	$kg \cdot m^{-3}$
σ	Stress	MPa
τ	helix angle or wrap angle	rad

Subscripts:

Subscripts	**Meaning**
0	position z = 0
cond	condensation
disp	displacement
evap	evaporation
L	position z = L
suc	suction
tot	total

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