Building a 5-stage Axial Compressor for an in-class demonstration

Suryansh S. Shekhawat

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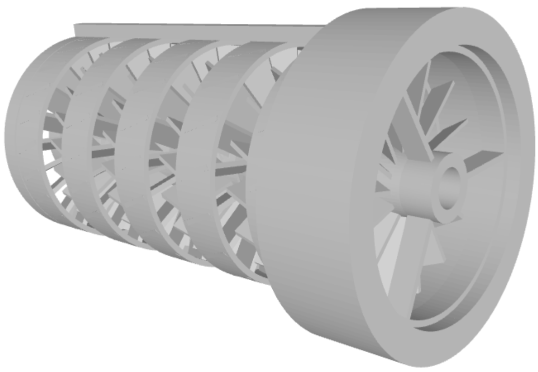
Abstract

This is a project report for a 5-stage axial compressor constructed to learn the basics of compressor mechanics, CAD design, and assembly. The report includes only details that were considered important from the perspective of learning outcomes. In this version, typical axial compressor design equations from standard references have been added for illustrative purposes.



General Characteristics

- Axial
- 5-stages (5 Rotor-Stator pairs)
- Has central stators (use bearings for mounts on main drive rod)
- Has a ducting pipe, cut in half for display purposes
- Has a complete inlet pipe with mounting marks

Components

0.1 Rotors

There are 5 rotors in this axial compressor (5-stage), with the following fin numbers:

$$\{14, 16, 18, 20, 22\}$$

These are fittingly labelled R-1, R-2, R-3, R-4, and R-5.

0.1.1 Calculations for Construction

- The diameter of the entry cone is only 1 mm more than the exit, with the height of the fin set being 10 mm, and the total height of the cone being 14 mm.
- The inlet hub diameter for the first rotor is 10 mm; by the last stage, the hub grows to 20 mm.
- The outer casing diameter is 80 mm (i.e., 40 mm radius).

0.1.2 Rotor Fin and Pitch Calculations

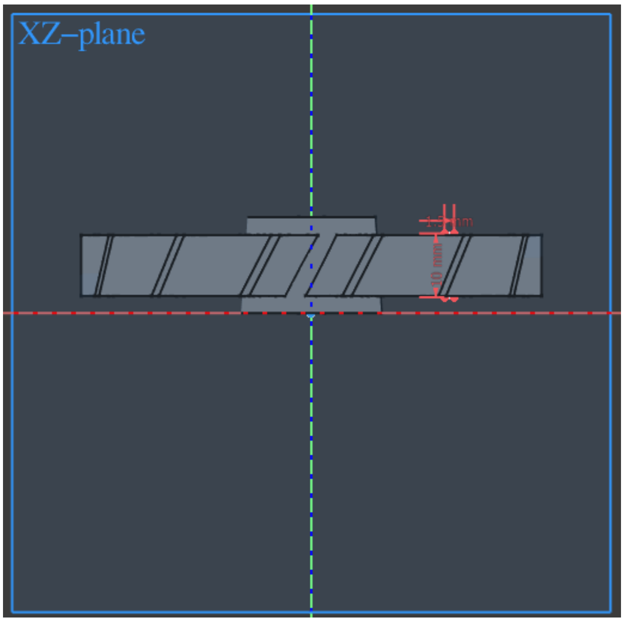
For aerodynamic performance, the spacing between rotor fins (the circumferential pitch) is key. The pitch p is calculated as:

$$p = \frac{2\pi r_{\text{mean}}}{N_{\text{fins}}}$$

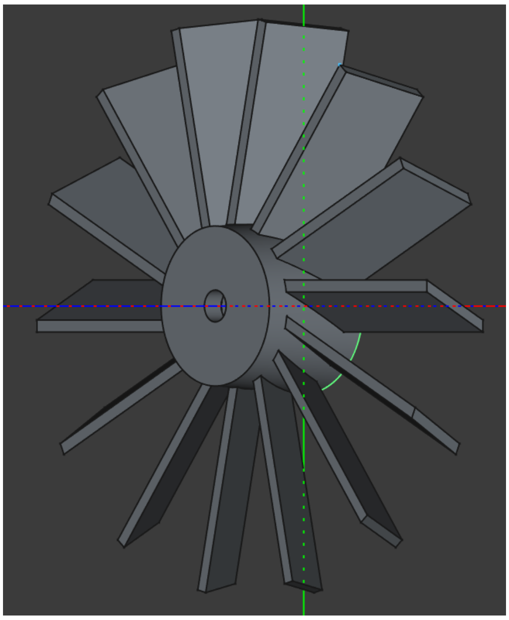
where r_{mean} is the mean radius of the rotor and N_{fins} is the number of fins. For instance, assuming a mean radius of 35 mm for rotor R-1 with 14 fins:

$$p = \frac{2\pi \times 35}{14} \approx 15.7\,\text{mm}.$$

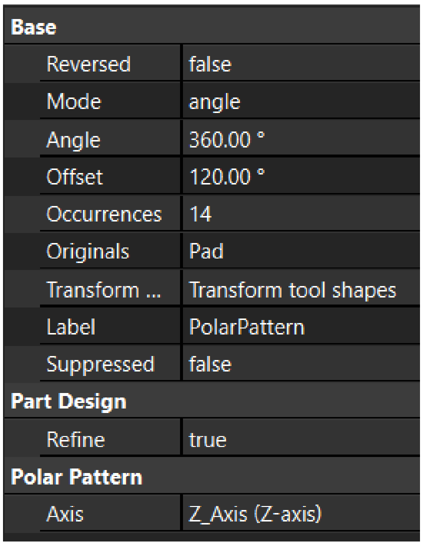
Similar calculations can be performed for each rotor stage, ensuring uniform flow conditions across the compressor.



(a) XZ axis view

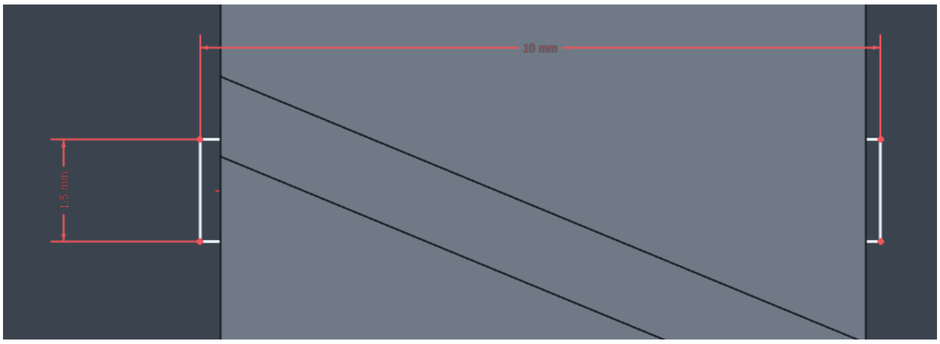


(b) Titled view of fins on a rotated fulcrum with a hole



Base	
Reversed	false
Mode	angle
Angle	360.00 °
Offset	120.00 °
Occurrences	14
Originals	Pad
Transform ...	Transform tool shapes
Label	PolarPattern
Suppressed	false
Part Design	
Refine	true
Polar Pattern	
Axis	Z_Axis (Z-axis)

(a) Settings used in Polar fins



(b) Side dimensions

0.2 Stators

There are 5 stators with the following fin counts:

$$\{15, 17, 19, 21, 23\}$$

These stators are labelled S-1, S-2, S-3, S-4, and S-5, each positioned immediately after its corresponding rotor.

0.2.1 Stator Design and Calculations

In addition to providing flow turning and diffusion, the stator fins are critical in controlling the exit flow angle. The circumferential pitch for stator fins is similarly given by:

$$p = \frac{2\pi r_{\text{mean}}}{N_{\text{fins}}}.$$

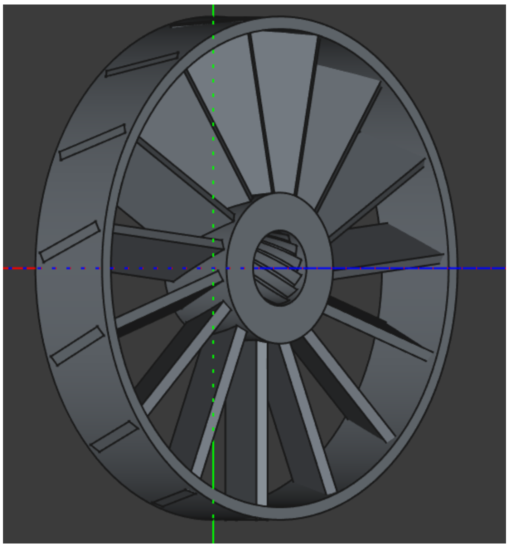
For example, for stator S-1 with 15 fins at a mean radius of 35 mm:

$$p = \frac{2\pi \times 35}{15} \approx 14.6\,\text{mm}.$$

Such calculations help in ensuring that the stator design minimizes flow separation and maintains efficiency. Additionally, the stator blade angles can be optimized based on the velocity triangles at the rotor–stator interface, a process well documented in texts such as [2].

Base	
Reversed	false
Mode	angle
Angle	360.00 °
Offset	120.00 °
Occurrences	15
Originals	Pad
Transform ...	Transform tool shapes
Label	PolarPattern
Suppressed	false
Part Design	
Refine	true
Polar Pattern	
Axis	Z_Axis (Z-axis)

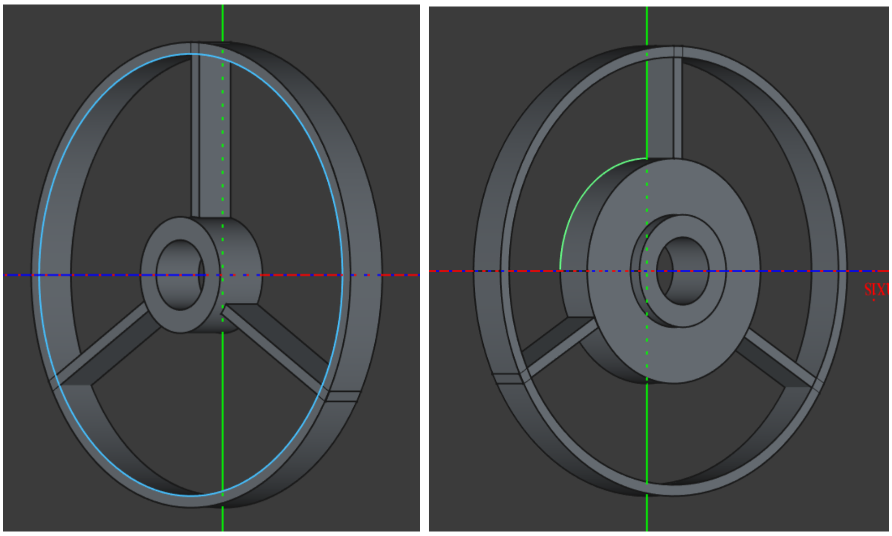
(a) Settings used in polar fins (Stator)



(b) Titled view of Stator - 1

0.3 Supports

The supports maintain alignment and structural integrity of the compressor assembly. The front and back supports are designed to manage the dynamic loads during operation.



(a) Front Support (b) Back Support

0.4 Casing

The casing houses the rotor–stator assembly and must be designed to withstand the internal pressure and mechanical stresses. The overall annulus of the compressor is defined between the variable hub radius and the fixed outer radius of 40 mm.

0.4.1 Casing Geometry and Structural Calculations

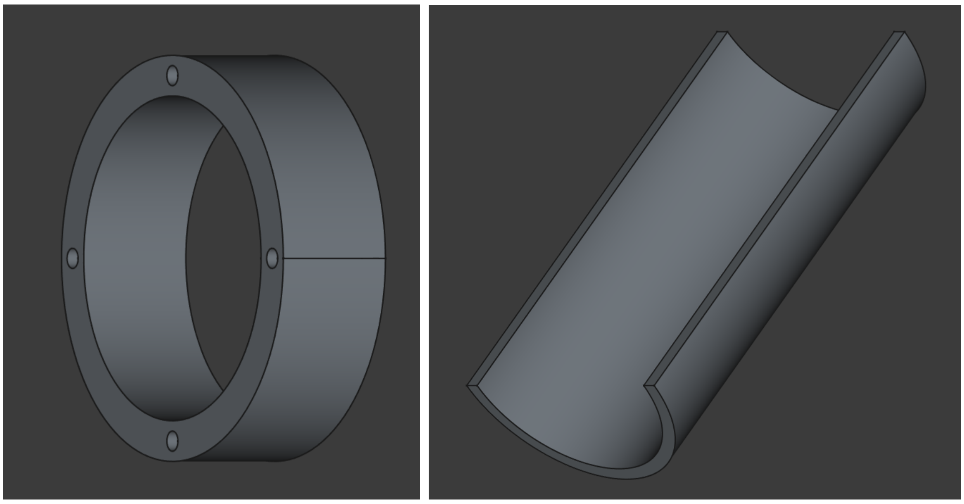
Based on the provided screenshot (see Figure **??**), the casing features reinforced sections at both the inlet and outlet. The annulus area for the compressor is given by:

$$A = \pi\left(r_{\text{out}}^2 - r_{\text{hub}}^2\right),$$

where r_{hub} varies from 5 mm (inlet) to 10 mm (exit). Additionally, the casing wall thickness t can be approximated using thin-walled pressure vessel theory:

$$t = \frac{p\,r}{\sigma_{\text{allow}}},$$

where p is the internal pressure, r is the mean radius, and σ_{allow} is the allowable stress of the casing material. Such calculations ensure that the casing will safely contain the compressor's operating pressures.



(a) Inlet Support (b) Cut-Case Support

1 Geometry and Annulus Areas

The compressor annulus is defined between the varying hub radius and the fixed outer radius $r_{\text{out}} = 40\,\text{mm}$. For instance:

1.1 Inlet Area (Stage 1)

$$r_{\text{hub},1} = \frac{10\,\text{mm}}{2} = 5\,\text{mm}, \quad r_{\text{out}} = 40\,\text{mm},$$
$$A_{\text{in},1} = \pi\left(40^2 - 5^2\right)\,\text{mm}^2.$$

1.2 Exit Area (Stage 5)

$$r_{\text{hub},5} = \frac{20\,\text{mm}}{2} = 10\,\text{mm},$$
$$A_{\text{ex},5} = \pi\left(40^2 - 10^2\right)\,\text{mm}^2.$$

A mean flow area for each stage may be estimated by interpolating between these values.

2 Compression Ratio Calculations

The overall pressure ratio of an axial compressor is influenced by the enthalpy rise per stage, blade speed, and flow angles. Detailed derivations are available in standard texts such as [1, 2, 3].

2.1 Stage Enthalpy Rise

For each stage, the ideal enthalpy rise Δh_{stage} can be estimated using the Euler turbomachine equation:

$$\Delta h_{\text{stage}} = U\,\Delta V_\theta \approx \psi\,\frac{U^2}{2}, \tag{1}$$

where

- U is the blade speed (mean radius multiplied by rotational speed),
- ΔV_θ is the change in tangential velocity,
- ψ is the stage loading coefficient (typically between 0.3 and 0.5).

2.2 Polytropic Efficiency and Stage Pressure Ratio

Assuming a polytropic efficiency η_{poly}, the stage pressure ratio π_s is given by:

$$\pi_s = \exp\left(\frac{\Delta h_{\text{stage}}\,\eta_{\text{poly}}}{R\,T_{01}}\right), \tag{2}$$

with

- $R \approx 287\,\text{J}/(\text{kg K})$ for air,
- T_{01} as the total (stagnation) temperature at the stage inlet.

For low Mach number flows, an approximate relation is:

$$\pi_s \approx 1 + \frac{\gamma}{\gamma - 1}\,\frac{\Delta h_{\text{stage}}}{c_p\,T_{01}},$$

where γ is the specific heat ratio and c_p is the specific heat at constant pressure.

2.3 Overall Compressor Pressure Ratio

For a 5-stage compressor:

$$\pi_{\text{total}} = \prod_{i=1}^{5} \pi_{s,i}. \tag{3}$$

If each stage has an identical pressure ratio π_s, then:

$$\pi_{\text{total}} = (\pi_s)^5.$$

A typical single-stage pressure ratio might be in the range 1.05 to 1.20, yielding an overall ratio approximately between:

$$(1.05)^5 \text{ and } (1.20)^5 \approx 1.28 \text{ to } 2.49.$$

2.4 Example Calculation (Illustrative)

Assume:

- $\psi = 0.45$,
- $U = 100\,\text{m/s}$,
- $\eta_{\text{poly}} = 0.85$,
- $T_{01} = 300\,\text{K}$.

Then, from Eq. (1):

$$\Delta h_{\text{stage}} = 0.45 \times \frac{(100)^2}{2} = 2250\,\text{J/kg}.$$

And from Eq. (2):

$$\pi_s = \exp\left(\frac{2250 \times 0.85}{287 \times 300}\right) \approx \exp(2.22) \approx 9.21.$$

This high value illustrates the sensitivity of the stage pressure ratio to the chosen parameters; for a demonstration setup, lower values of ψ and U would yield more realistic pressure ratios.

3 References

References

[1] H. I. H. Saravanamuttoo, G. F. C. Rogers, H. Cohen, and P. Straznicky, *Gas Turbine Theory*, 7th ed., Pearson, 2017.

[2] H. Cohen, G. F. C. Rogers, and H. I. H. Saravanamuttoo, *Gas Turbine Theory*, 4th ed., Addison Wesley Longman, 1996.

[3] S. M. Yahya, *Turbines, Compressors and Fans*, 4th ed., Tata McGraw-Hill, 2010.