TRANSONIC FAN FOR AERO-ENGINES

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OBJECTIVE STATEMENT

To design a highly efficient, light and compact aero-engine transonic fan by:

- Maximising total to total efficiency (β_{tt})
- Minimising frontal area
- Minimising weight (blade count, chord, etc.)

DESIGN PROCEDURE

The design procedure consists of 3 phases: meanline design, detailed blade design and post processing - as illustrated in Figure 4. The entire design procedure is based on a set of duty coefficients asuumed before meanline design. The meanline design phase results in a compressor size, performance and fluid dynamic parameters at meanline which are used as inputs for meangen . Here 2 iterations run to correct for loss coefficient estimates and to ensure the machine operates in the allowable massflow range.

The Detailed Blade Design phase is characterized by blade shape determination, either from assumptions or reference data, and grid point selection. After passing this data through Multall, an iteration is performed between the CFD outputs and meanline design to ensure requirement satisfaction.

The results are then post processed and visualised in the Post Processing phase.

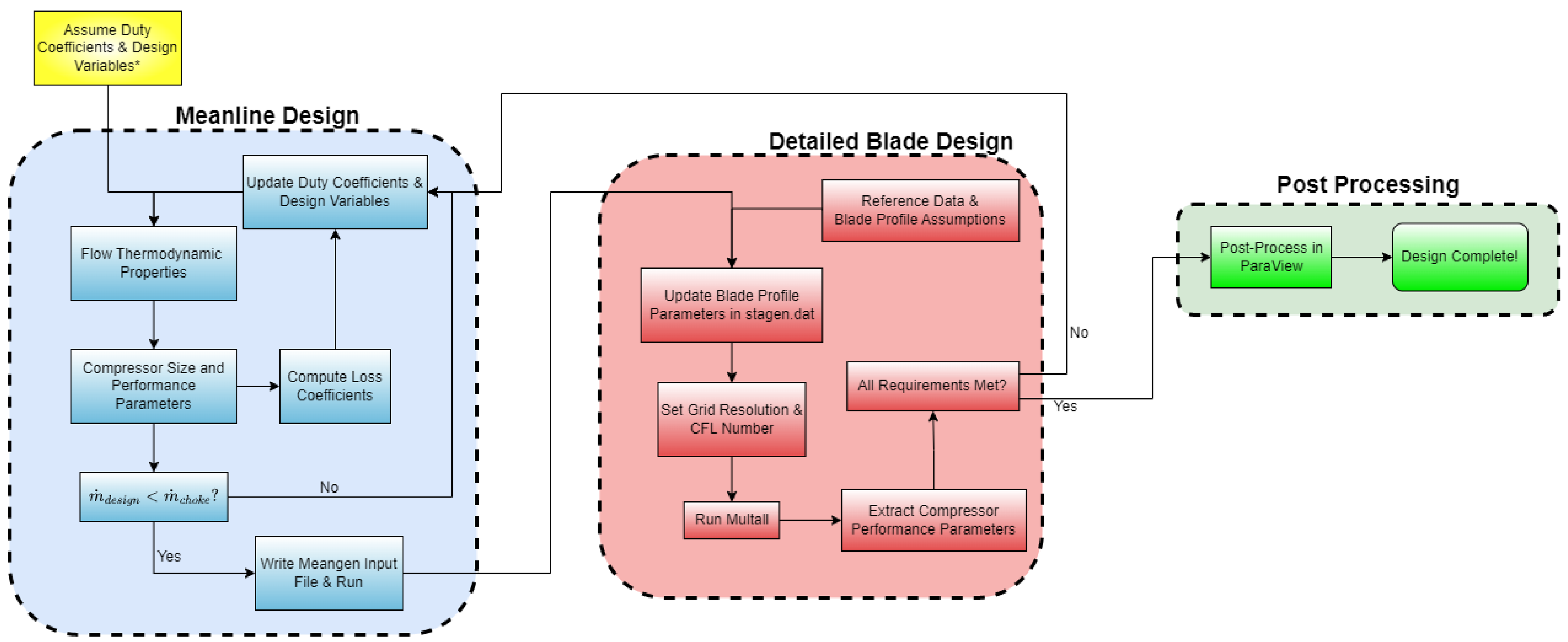

Figure 4: Design Procedure.

CONCLUSIONS & RECOMMENDATIONS

Conclusions:

- Large variation in degree of reaction and work coefficient along bladespan - unexpected given twist parameter was set to 0.5.
- Good agreement in rotor loss coefficient between CFD and meanline design. Y_s in CFD and meanline differ slightly due to the difference in thickness between the 2 design phases.
- Flow separation near rotor blade trailing edge suggests that the initially chosen double circular arc blade shape is not the optimal blade shape choice.
- Efficiency measurements in meanline design did not account for variability in twist of blades and only modelled boundary layer losses, hence CFD loss coefficients are larger than in meanline design.

Recommendations:

- Set iteration loop between CFD output and meanline design to determine the effect of and optimal twist parameter for minimal sum of rotor and stator loss coefficients.
- Shift rotor blade maximum thickness aftward to delay flow separation by having a smaller region of unfavourable pressure gradient over rotor blade.
- During iterations between detailed blade and meanline design, increasing row gap dramatically increased stage efficiency but only 2 row gap updates were made - an optimal value for minimal efficiency is to be investigated.

CFD ANALYSIS

The CFD analysis through Multall was used to verify the satisfaction of design criteria and operation ranges. The following performance parameters were found from the CFD analysis: $\dot{m} = 270.96\ \mathrm{kg\,s^{-1}}$, a stage power of 8.54 MW and $\eta_{poly} = 0.845$, $\eta_{tt} = 0.837$, $\eta_{ts} = 0.363$. The duty coefficients and loss coefficients for the rotor and stator (Y_r, Y_s) at different spanwise locations are as follows:

- Hub: R = 0.02717, $\psi = 1.960$, $\phi = 1.7$, $Y_r = 0.3341$, $Y_s = 0.1922$
- Mid: R = 0.875, $\psi = 0.455$, $\phi = 0.6$, $Y_r = 0.0897$, $Y_s = 0.0269$
- Tip: R = 0.447, $\psi = 0.199$, $\phi = 0.364$, $Y_r = 0.0597$, $Y_s = 0.0557$

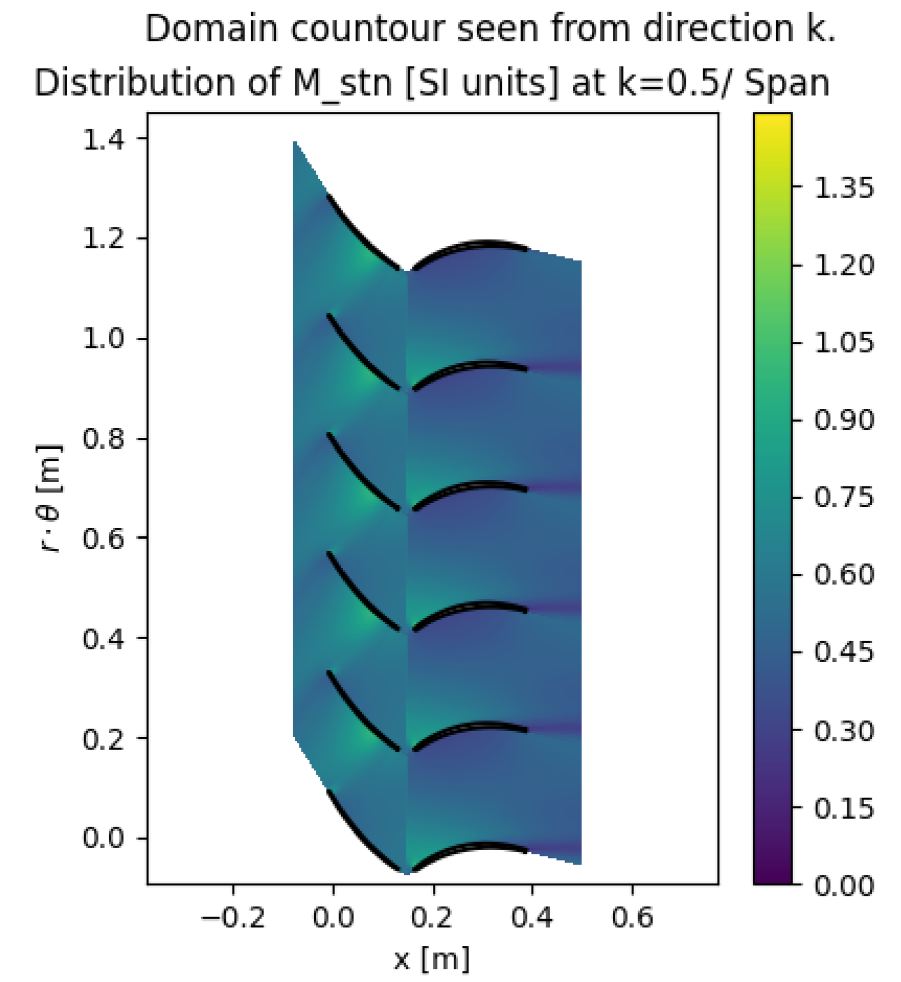

Figure 1: Mach number in stationary frame at midspan.

The stage performs within the specified operation conditions in meanline design, as indicated by Figure 1 but the sudden deceleration due to shock suggests flow separation near the trailing edge of rotor blades. This is confirmed by Figure 2. This figure also shows a laminar separation bubble over the stator blade at, indicated by the constant C_p section near the leading edge.

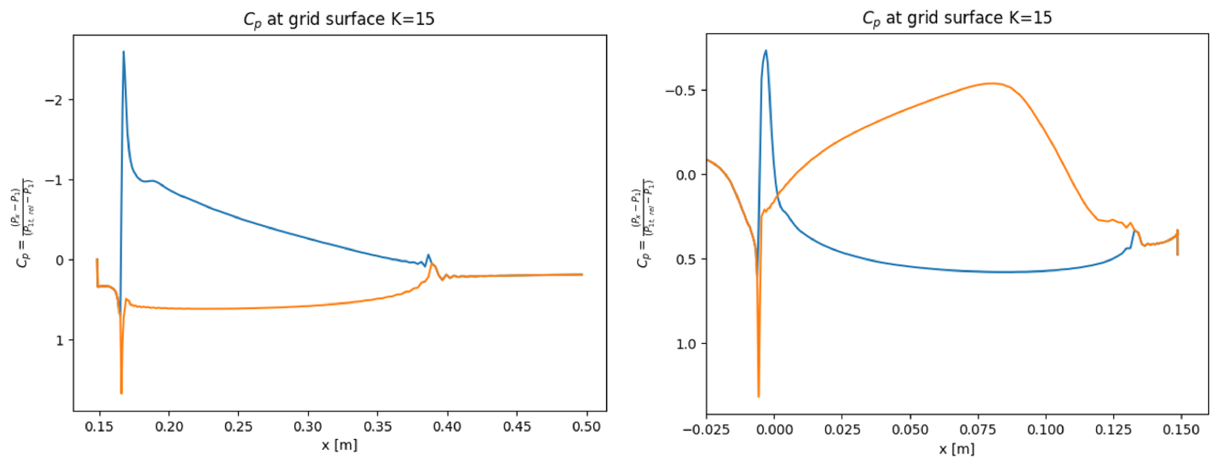

Figure 2: Pressure coefficient distribution over stator (left) and rotor (right) blade at midspan; Blue = upper surface, Orange = lower surface.

MEANLINE DESIGN

The meanline design is based on the choice of 3 meanline design parameters: inflow angle, flow coefficient and degree of reaction. The degree of reaction is chosen for the best compromise between compressor weight and efficiency. The result of choosing $\phi = 0.6$ and an $R = 0.8$, assuming axial inflow is a $\psi = 0.3789$. Compressor sizing from meanline design results in: $r_{hub} = 0.256$ m, $r_{tip} = 1.2$ m, $c_{rotor} = 0.1319$ m, $r_{stator} = 0.2205$ m and a power of 6.641 MW. Profile loss modelling estimates that the rotor and stator loss coefficients are 0.07498 and 0.06792 respectively. Based on the enthalpy change, $\eta_{tt} = 0.935$ and $\eta_{ts} = 0.634$ respectively.

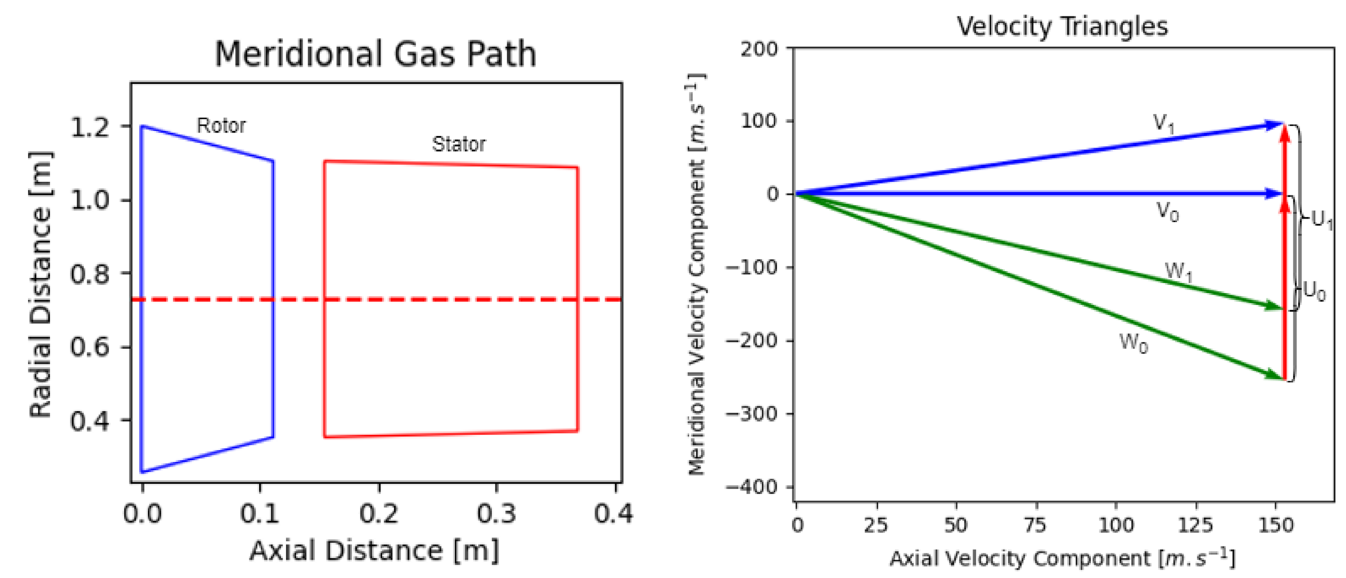

Figure 3: Meridional gas path (left) and meanline velocity triangles (right).

Table 1: Flow Properties.

Station	P_t [Pa]	T_t [K]	P [Pa]	T [K]
0	32700	240.82	27590.0	229.2
1	45780.0	265.12	36678.15	248.85
2	45780.0	265.12	39126.99	253.49

Table 2: Blade forces per unit blade span.

	Net Aerodynamic Force [$\mathrm{kN\,m^{-1}}$]	Centrifugal Force [$\mathrm{kN\,m^{-1}}$]
Rotor	7.38	89.17
Stator	3.36	89.17