

CRANFIELD UNIVERSITY

TALA EL SAMAD

INVESTIGATION OF THE FEASIBILITY OF
EMPLOYING A RADIAL TURBINE FOR A UTILITY-
SCALE SUPERCRITICAL CO₂ POWER CYCLE

SCHOOL OF WATER, ENERGY AND ENVIRONMENT
Energy and Power

Doctor of Philosophy
Academic Year: 2016 – 2020

Supervisor: Prof. John Oakey & Dr. Joao Amaral Teixeira
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To the most caring family

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Abstract

With the current growth rate of world population, comes a high demand on electric energy generation. Technological advancements in metallurgy, fuel conversion, heat transfer, and turbomachinery design allow the continuous dominating rule of fossil fuel energy resources and the expansion of non-renewable based power cycles. The challenges arising due to greenhouse gas emissions from fossil fuel combustion necessitate regulated policies to limit CO₂ emissions in particular. Design considerations and issues in power plants revolve around increasing efficiencies, reducing noise pollution and land footprint, while mitigating emissions. An example of advanced power configurations include the supercritical CO₂ cycle that has gained much interest in recent years due to its compact-size turbomachinery, increased efficiencies compared to other conventional steam or gas cycles, and its ability to be coupled with a wide range of heat sources. Another avant-garde technology is an oxy-combustion cycle that proves more favourable than the other carbon capture and storage routes of gasification and absorption. A proposed NET Power (Allam) cycle, which combines both technologies of using supercritical CO₂ conditions in an oxy-combustion gas-fired power plant, seems promising with claims of cycle efficiencies reaching 55%. However, the Allam cycle is still in a pre-mature phase due to the barriers that inhibit its full-scale development; challenges include the design of a high-pressure, high-temperature turbine which dictates cooling requirements, material considerations, and number of stages to name a few. A radial turbine, which generally has a simpler construction and fewer stages when compared to an axial turbine, is suggested as a superior candidate configuration for cycles of high fluid density. A thermodynamic analysis of a mid-range cycle similar to that proposed by NET Power is established while lowering the turbine inlet temperature to 900°C in order to remove cooling complications within the radial turbine passages. The cycle conditions are then considered for the design of a 100 MW_{th} power scale turbine by using preliminary and higher fidelity methods. Two radial turbine designs are illustrated; the first, whilst not satisfying the cycle operating conditions, is used to showcase the detailed 1D design as well as the 2D and 3D analyses

procedures followed during this work. A second 510 mm diameter turbine, running at 21,409 rpm, capable of operating within a 5% range of the required cycle conditions, is designed and presented. Results from computational fluid dynamics simulations indicate the loss mechanisms responsible for the low-end value of the turbine total-to-total efficiency which is 69.87%. Mechanical stress calculations show that the aerodynamic flow path of the rotor blades experience tolerable stress values, however a more detailed disc design is required to meet actual material constraints.

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Nomenclature

Roman Symbols

A	Area	$[\text{m}^2]$ unless otherwise stated
A_{SA}	Axial semi-axis	$[\text{m}]$
b	Passage width	$[\text{m}]$ unless otherwise stated
C	Absolute velocity	$[\text{m/s}]$
c	Chord	$[\text{m}]$ unless otherwise stated
C_{0s}	Spouting velocity	$[\text{m/s}]$
d	Diameter	$[\text{m}]$ unless otherwise stated
F	Force	$[\text{N}]$
g	Gravitational acceleration	$[\text{m/s}^2]$ unless otherwise stated
h	Enthalpy	$[\text{J/kg}]$ unless otherwise stated
l	Length	$[\text{m}]$ unless otherwise stated
M	Mach number	$[-]$
m	Mass	$[\text{kg}]$
M_b	Bending moment	$[\text{N m}]$
$\dot{m}$	Mass flow rate	$[\text{kg/s}]$
M_t	Twisting moment/ torque	$[\text{N m}]$
N	Number of blades	$[-]$
$\dot{n}$	Mole flow rate	$[\text{mol/s}]$
n_p	Polytropic coefficient	$[-]$

n_s	Specific speed	[-]
o	Throat	[m] unless otherwise stated
P	Pressure	[Pa] unless otherwise stated
q	Pitch	[m] unless otherwise stated
$\dot{Q}$	Volumetric flow rate	[m ³ /s]
R	Degree of reaction	[-]
r	Radius	[m] unless otherwise stated
R_{SA}	Radial semi-axis	[m]
S	Stress	[Pa] unless otherwise stated
s	Entropy	[J/kg K] unless otherwise stated
S_y	Yield strength	[Pa] unless otherwise stated
T	Temperature	[K] unless otherwise stated
t	Thickness	[m] unless otherwise stated
U	Blade velocity	[m/s]
V	Volume	[m ³]
W	Relative velocity	[m/s]
$\dot{W}$	Power	[W] unless otherwise stated
$\dot{w}$	Specific work	[m ² /s ²]
Z	Axial length	[m] unless otherwise stated
Z_c	Compressibility factor	[-]

Greek Symbols

α	Absolute flow angle	[°]
β	Relative flow angle	[°]
Δ	Difference	
ε	Displacement/ deformation	[mm]
η	Efficiency	[%]
γ	Heat capacity ratio	[-]
v_s	Velocity ratio	[-]
v_s'	AxStream velocity ratio	[-]
ω	Rotational speed	[rad/s] unless otherwise stated

Nomenclature

xxv

ϕ	Flow coefficient	[-]
Π	Pressure ratio	[-]
ψ	Stage loading coefficient	[-]
ρ	Density	[kg/m ³]
σ'	von Mises stress	[Pa] unless otherwise stated
σ'_{all}	Allowable von Mises stress	[Pa] unless otherwise stated
τ	Shear stress	[Pa] unless otherwise stated
θ	Metal angle	[°]

Subscripts

0	Total value
1	Volute inlet station
2	Volute outlet/ Stator inlet station
3	Stator outlet station
4	Rotor inlet station
5	Rotor mean outlet station
5 _h	Rotor hub outlet station
5 _m	Rotor meanline outlet station
5 _s	Rotor shroud outlet station
abs	Absolute reference frame
av	Average
ax	Axial
e	Electric
g	Gauging metal angle
i	Inner
id	Ideal
in	Inlet
isen	Isentropic
LE	Leading edge
M	Meridional
m	Meanline

<i>max</i>	Maximum
<i>NE</i>	Net electric
<i>o</i>	Outer
<i>out</i>	Outlet
<i>R</i>	Rotor
<i>rel</i>	Relative reference frame
<i>S</i>	Stator
<i>TE</i>	Trailing edge
<i>th</i>	Thermal
<i>ts</i>	Total-to-static
<i>tt</i>	Total-to-total
<i>U</i>	Tangential/ circumferential

Acronyms / Abbreviations

<i>AM</i>	Additive Manufacturing
<i>ASU</i>	Air Separation Unit
<i>AZEP</i>	Advanced Zero-Emission Power Plant
<i>CCGT</i>	Combined Cycle Gas Turbine
<i>CCS</i>	Carbon Capture and Storage
<i>CCUS</i>	Carbon Capture, Utilisation and Storage
<i>CES</i>	Clean Energy Systems
<i>CFD</i>	Computational Fluid Dynamics
<i>CLC</i>	Chemical Looping Combustion
<i>CPU</i>	Compression and Purification Unit
<i>EoS</i>	Equation of State
<i>Eq.</i>	Equation
<i>ETN</i>	European Turbine Network
<i>FEA</i>	Finite Element Analysis
<i>GE</i>	General Electric
<i>GHG</i>	Greenhouse Gas
<i>GHGT</i>	Greenhouse Gas Control Technologies

<i>GT</i>	Gas Turbine
<i>HE</i>	Heat Exchanger
<i>HP</i>	High Pressure
<i>HRSG</i>	Heat Recovery Steam Generator
<i>HT</i>	High Temperature
<i>IEAGHG</i>	International Energy Agency GreenHouse Gas
<i>IPCC</i>	Intergovernmental Panel on Climate Change
<i>IRR</i>	Internal Rate of Return
<i>LCOE</i>	Levelised Cost of Electricity
<i>LHV</i>	Lower Heating Value
<i>MCFC</i>	Molten Carbonate Fuel Cell
<i>NG</i>	Natural Gas
<i>NPV</i>	Net Present Value
<i>OEM</i>	Original Equipment Manufacturer
<i>OTM</i>	Oxygen Transport Membrane
<i>PD</i>	Preliminary Design
<i>PESTEL</i>	Political Economic Social Technological Environmental Legal
<i>PS</i>	Pressure Side
<i>PV</i>	Photovoltaic
<i>RMS</i>	Root Mean Square
<i>SCOC – CC</i>	Semi-Closed Oxy-Combustion Combined Cycle
<i>SF</i>	Safety Factor
<i>SFR</i>	Sodium Fast-cooled Reactor
<i>SS</i>	Suction Side
<i>TBC</i>	Thermal Barrier Coating
<i>TIP</i>	Turbine Inlet Pressure
<i>TIT</i>	Turbine Inlet Temperature
<i>TOP</i>	Turbine Outlet Pressure
<i>TOT</i>	Turbine Outlet Temperature
<i>TRL</i>	Technology Readiness Level

<i>WF</i>	Working Fluid
<i>WHR</i>	Waste Heat Recovery
<i>ZEITMOP</i>	Zero Emission Ion Transport Membrane Oxygen Power
<i>ZE – SOLRGT</i>	Zero-Emission Solar Gas Turbine

CHAPTER 1

Introduction

The first chapter of the thesis gives some background information related to the topic which is the focus of this piece of work; it provides an overview on the shift towards cleaner energy generation systems through novel technologies. The present and expected future use of energy is the underlying reason to pursue developments in natural gas power cycles. A section on the current advancements in conventional cycles includes an explanation of key technologies and enablers, such as higher operating temperatures and pressures, blade cooling and manufacturing techniques for more efficient units. A high-level comparison between different clean or more efficient power systems is discussed to allow the selection of the adopted cycle for this research. The motivation behind the PhD research along with the objectives set to achieve the main aim are also stated.

1.1 Energy Trends

Climate change remains one of the major concerns of our modern day, and non-polluting power production is the prime focus of governments, investors and companies to respond to the crisis. With the current growth rate of world population by around 1.05% per year [1], comes a huge energy surge in response to rapidly growing economies demanding more energy for their highly-industrial sectors, and to developing nations trying to harvest basic energy needs. If the aim to keep emissions, particularly greenhouse gases (GHGs), at or below environmentally sustainable levels, power production methods need to undergo a drastic change from current generation modes. Low-carbon energy sources, such as biomass, definitely come to the forefront of this movement to ensure zero-emissions; however, it is inherent that methods to diversify the intermittent energy mix supplied by renewables are required to fully address the ever-increasing power demand.

There exists many speculations on future use of energy sources with different trends, however for a reference case, a study performed by IEA is examined further here. Current target policies, with a projection of 1% yearly increase in energy needs until 2040, give an expectation (shown in Figure 1.1) that renewables will experience a considerable growth (especially solar photovoltaics -PVs), and that the use of energy from oil-derived sources will not rise, whilst gas-fuelled power generation will continue a steady increase [2]. The continuous use of natural gas requires measures to mitigate GHG emissions through increased energy efficiency and carbon capture methods.

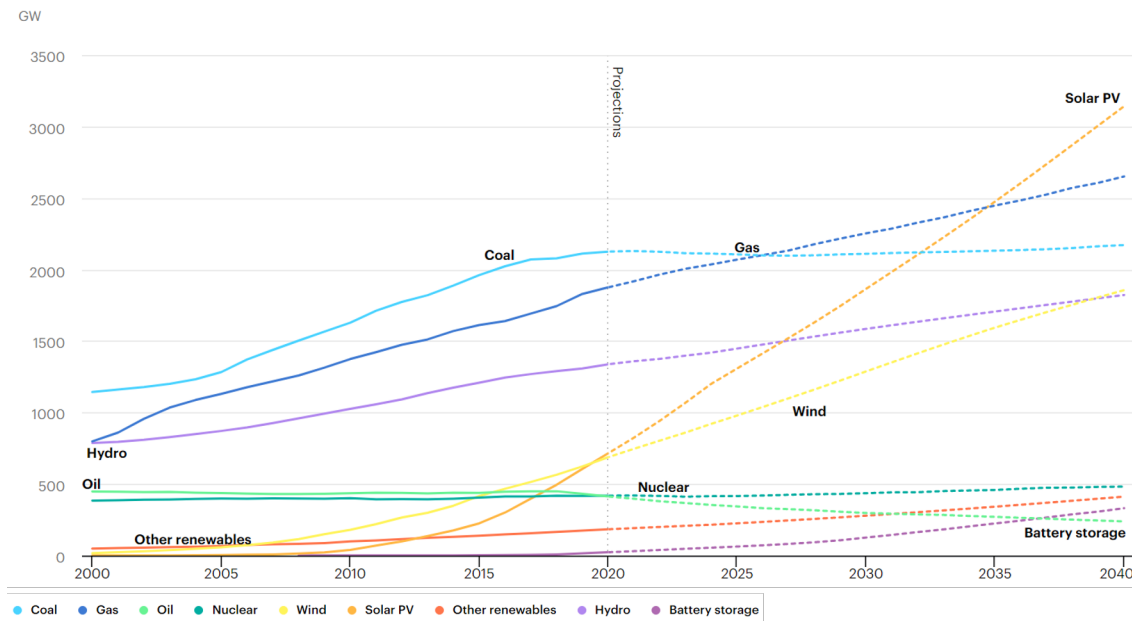


Fig. 1.1 Power generation by source, 2000-2040 [2]

1.2 Background

Natural gas is seen as the preferred fossil fuel to be used in the transition to a more renewable power generation system due to its relatively lower carbon impact compared to oil- and coal-derived fuels when carbon capture and storage (CCS) technologies are not implemented. Even when CCS is possible, the higher efficiency, lower capital costs and lower levels of other polluting impurities give natural gas (or similar methane-based fuel gases, e.g. shale gas) an advantage over higher carbon- content alternatives. For many years, the market for gas-fired power generation has been dominated by the major equipment suppliers who have been engaged in a battle to increase operating efficiency, reducing both the levelised cost of electricity and CO₂ emissions. However, with the substantial growth in solar PV (and the more steady growth in intermittent wind renewable) in many markets, gas-based power generation is being progressively relegated to operating in a complimentary mode to meet demand peaks and to fill gaps in supplies from

clean energy. Electricity supplied from nuclear power is also given priority for dispatch compared to that supplied from renewable sources (i.e. it is higher in the merit order of possible electricity sources), adding further complexity to the grid operators' challenge in meeting electricity demand. As nuclear plants can only supply base load power in their current configurations, the space for electricity supply from fossil-fuelled plants is further squeezed, leaving them to be the providers of flexible power to meet demand peaks and thus balance power grids.

In relation to current trends of utility-scale energy storage capacities across the world (80 GW between 2016 – 2025 [3]), there is a growing expectation that large scale energy storage, able to deliver flexibly, combined with more distributed generation from small-scale power/ storage and demand-side management, will lower demand peaks and so gradually abate the need for flexible fossil-fuelled generation thus reducing emissions. The actual timescale for the implementation of large-scale energy storage, distributed generation/ storage or demand-side measures remains unclear. In addition, other electricity market drivers, such as the support for efforts of transport and heat decarbonisation through electrification (amongst other options), is increasing electricity demand, therefore opening up the opportunity for flexible gas-based power generation to persist at significant levels in the coming decades.

However, the global power markets are giving conflicting messages at present, driven largely by varying responses to addressing climate change in different parts of the world. With European countries broadly moving rapidly away from all forms of fossil-fuelled power generation (in order to meet the challenging net-zero 2050 targets recently agreed [4]), others including the US and the major coal-using countries are still installing coal and gas power plants. As a result, it is not surprising that the major global original equipment manufacturers (OEMs) supplying gas-based power systems largely in the form of combined cycle gas turbine plants, are finding it difficult to find a way forward for their businesses. The market for these technologies has shrunk substantially in recent years [5] (as will be seen in Figure 1.2), mainly due to the support provided for renewables and the updated requirements from gas turbine plants making them less economically viable and putting significant pressure on OEMs to rationalise the sector. This has left the three major global power generation OEMs: Siemens, General Electric (GE) and Mitsubishi with many smaller suppliers looking for niche opportunities. While diversifying their energy portfolios, and watching other energy companies like the oil/gas majors moving into their operating space, all of these leading gas turbine OEMs are pursuing a number of parallel options, such as delivering upgrades to support current customer needs for the use of low-cost/ bio-derived fuels and improving operational flexibility (better start-up/ shutdown times, ramp rates). Increasing power generation efficiency through the implementation of incremental developments in combustion, materials, and blade cooling is still being pursued simultaneously with the development of new and retrofit hydrogen firing options and advanced cycles using supercritical CO₂ as a major component of the working fluid (based on advantages listed in Section 2.2).

The technologies discussed in the following parts of the introductory chapter reflect

the pathways being considered for advanced natural gas-fired power generation, either through improved efficiency combined with the implementation of carbon capture, utilisation and storage (CCUS), or through switching to unconventional working fluids.

1.2.1 Advancements in CCGT Cycles

Increasing firing temperatures and pressures in combined cycle gas turbine (CCGT) power plants has been the traditional approach to increasing cycle efficiencies for base-load operation, hence reducing the levelised cost of electricity (LCOE) along with CO₂ emissions, but at the cost of higher NO_x pollution. This approach, which was dominant for many years through the 1990s and early 2000s, has meant that operational flexibility was compromised in the larger systems and gas composition requirements narrowed. Nonetheless, market drivers in more recent times have become focussed more on operational flexibility in order to provide balancing power to meet demand peaks and to offset low generation periods from wind and solar renewables. In addition, with gas being more widely traded, maintaining gas compositions within narrowed specifications has led to problems in supplying high quality-grade fuel; operation with bio- and waste-derived gases, whether or not methane-based, has also become more significant. In summary, gas turbines having become less operationally and fuel adaptable, have had to be modified to meet the modern stipulations for higher ramp rates, start-up and shutdown times and part-load performance.

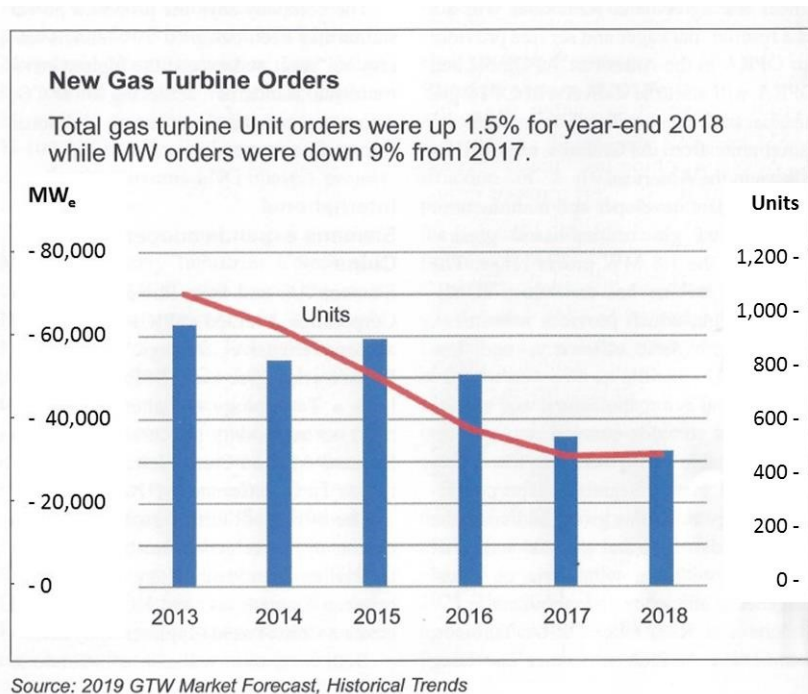


Fig. 1.2 Gas turbine order trends [5]

While gas turbine OEMs have long claimed that their technologies are fuel flexible, this has only been possible by changing multiple components (combustor, fuel systems, etc.) and by de-rating plants, as the turbomachinery would only have been optimised for the original design fuel, each time a significantly different fuel is used. Although the global market for gas turbines has shrunk by at least one third (Figure 1.2 [5]) with the rise in renewable technologies, there remains an ongoing need for gas turbines to provide flexible power alongside intermittent renewables, and in some parts of the world to provide base-load power where renewables are not yet a priority. Both the renewables and gas markets benefit from higher operating efficiencies as long as operational flexibility and maintainability are not compromised. The three global OEMs are all continuing to develop machines to achieve higher combined cycle efficiencies (e.g. GE9HA > 64%, Siemens 8000HL/9000HL > 63%), [6] with higher firing temperatures contributing to this advance alongside other incremental improvements.

The trend towards higher efficiency has continued to be evolutionary, with few revolutionary elements; the application of additive manufacturing being the notable exception [7]. Key areas which have received attention are compressor performance, combustion chamber design, cooling optimisation, thermal barrier coatings and sometimes changes in turbine blading. Avoiding increased capital and operations costs and reducing complexity to improve reliability have been indispensable priorities. The application of advanced modelling and manufacturing approaches to achieve these progressive developments, while managing costs, and their potential for future incremental developments has been a clear direction for all OEMs.

A review of some current technologies that enable improved CCGT performance is presented hereafter.

Higher Firing Temperatures

Increasing firing temperatures and pressures leads to improved thermodynamic cycle efficiency by allowing the cycle to approach a higher maximum attainable Carnot efficiency. Such improvements can only be delivered if combustor components can be upgraded through development of better materials and cooling designs to mitigate the induced thermal stresses and to ensure emissions, particularly of NO_x , are not increased beyond restrictions. For example, GE claims that the GT26 HE has the highest firing temperature of any machine on the market [8], and although the exact turbine inlet temperature (TIT) is not publicly disclosed it probably exceeds 1600°C (to surpass Mitsubishi's J-Series TIT [9]). Surging cooling mass flows to help control increased combustor component temperatures cannot be viewed in isolation: increased firing temperatures impact the cooling requirements for most hot gas path components, such as transition ducts, blades and vanes, and so lead to a significantly higher overall cooling air demand, thus reducing air availability for combustion. This leads to changes in other components, increased costs, and can significantly reduce the scale of the benefit expected from hotter temperatures.

As a result, improved materials, in particular thermal barrier coatings and the application of additive manufacturing to provide optimised cooling structures are key to achieving increased efficiency through raising firing temperatures.

Higher Compression Ratios

Operating pressures and turbine pressure ratios (Π) in combined cycle systems are lower than in simple cycle systems using the same turbine due to the need for high exhaust temperatures to drive the bottoming steam cycle. Also, in combined cycle systems, the maximum specific power (kW for a given mass flow of air) is an important parameter as increasing it leads to smaller equipment sizes and lower costs. Advanced modelling techniques are allowing OEMs to increase compression ratios without resulting in costly impacts on other parts of the overall system. For example, through improved compressor blade design Siemens has reported raised compression ratios from 21:1 to 24:1 while simultaneously reducing the number of stages [10]. Improved seals to limit leakage offer a further incremental means of achieving improved efficiency when combined with other approaches, with this option not leading to significant impacts on other system components.

Compression Intercooling

This approach is not being widely explored to achieve improved performance, largely due to the increased costs and complexity involved, and so its implementation is limited to individual opportunities where it can offer an economic benefit. For instance, water injection is a method of compressor intercooling that has been applied to over 1000 gas turbines world-wide with a boost in power of up to 20% [11].

Advanced Blade Cooling Options

A range of blade cooling options has been explored over the past decades driven by the desire to increase firing temperatures. However, increased blade and vane cooling increases the turbomachinery complexity and adds cost and efficiency penalties which can significantly offset the benefits expected from the increased temperatures. Many problems have been experienced with some of the novel approaches, e.g. the use of steam or closed cooling loops, thus the conventional use of compressor air for cooling hot gas path components remains the preferred approach. Cooling efficiency has been improved through advanced modelling, without the need to increase the demand for cooling air from the compressor, by optimising the blade and vane internal cooling structures. Current blade and vane manufacturing methods have not been capable of delivering the intricately optimised structures as designed using computational fluid dynamics (CFD) techniques, therefore additive manufacturing (AM) is seen as offering a pathway to the manufacture of improved structures.

Improved Materials and Coatings

As indicated above, improved materials are essential if efficiency improvements from higher TITs are to be achieved. Better blade and vane materials with improved creep and/or fatigue resistance take many years to implement so it seems unlikely that this pathway will contribute significantly to efficiency improvements by allowing components to perform reliably at higher operating temperatures, and so require less cooling. Moreover, the focus on the alloys used for hot gas path components is on resistance to fatigue for plants being operated intermittently and on the repair of these components to help minimise maintenance costs.

Advanced coatings, in particular thermal barrier coatings (TBCs), are the major area of development to achieve efficiency improvements. TBCs, traditionally made with yttria-stabilised zirconia, provide the interface between the hot combustion gas and the underlying high temperature (HT) components, combustor cans, transition ducts and early-stage blades and vanes [12]. The challenge for the external surface of the TBC is to survive in the hotter gas path environment, resisting sintering and other interactions. Surface sintering and interactions with gas path contaminants degrade TBC thermal performance, change its microstructure, and lead to its premature failure and spallation, with serious consequences for the underlying materials. Novel adjustments to TBC compositions are being investigated to increase the thermal resistance (i.e. temperature drop across a given thickness of TBC) as well as to resist sintering. OEMs and coating providers are also pursuing improvements to resist spalling due to thermal cycling stresses through additional processing steps (e.g. surface treatments) and the introduction of controlled defects (e.g. laser-cut slots) to reduce risks of spallation due to microstructure changes and thermal cycles. TBCs are also being deployed on more turbine stages, and cooling introduced to previously uncooled stages, to further improve performance [13].

Additive Manufacturing

Additive manufacturing is being widely explored and implemented by gas turbine OEMs and some end-users, with the major OEMs now owning their own production facilities. This generic technology is available using a variety of approaches, but for metallic components most are largely based on the technologies developed for the application of coatings. As such, it has mostly been limited to relatively small components using the material compositions developed for advanced alloys and coatings, though the AM approach is not limited to these routes; AM components have been made using advanced titanium, stainless steel and nickel based alloys [14].

Research within OEM companies has proven the value of AM in making special components, in particular fuel injectors, and for blade/vane repair. A key issue here is the difference in microstructure derived from AM and so the manufactured components will have different mechanical properties compared to the same components made using con-

ventional (subtractive) methods [14]. As a result, AM is mainly being focussed on components where the mechanical failures are not common and where it offers a low cost route to manufacture. For this reason, most developments have focussed on stationary parts such as fuel injectors for which AM offers a route to more effective designs at reduced costs; many such components are now in service. Other parts, including compressor and turbine blades/ vanes have also been manufactured and tested in operating machines, although AM-made blades may be expected to offer lower reliability and so cannot yet be seen as standard replacements for conventional parts [15]. Efforts are pushed to improve the properties of AM parts in order to bring them more into mainstream production. Table 1.1 lists the parts currently being manufactured using AM methods, but these are not all necessarily yet in service [16].

Table 1.1 Components currently being manufactured [16]

Critical static components	GT	Critical rotating components	GT	Other components
Burners		Compressor blades		Heat exchangers
Fuel nozzles		Turbine blades		Centrifugal compressor impellers
Fuel injectors		Turbine disks		
Turbine shrouds		Compressor blisks (blades and disks as a single part)		

1.2.2 Impact of Post-Combustion CO₂ Capture

A CCGT emits approximately 400 gCO₂/kWh [17]. To meet the UK and the global climate change targets, total emissions from all power generation sources must fall to < 150 gCO₂/kWh by 2030 and were aiming for < 50 gCO₂/kWh by 2050, prior to the introduction of the net-zero by 2050 target. By these points in time either most CCGTs must be decommissioned or some form of carbon capture and storage (CCS) must be installed to allow them to continue operating.

Most work on CCS has focussed on emissions from coal fired power stations and industrial emissions, and it is only recently (< 10 years) that significant attention has turned to emissions from gas-fired power generation. Flue gas CO₂ concentrations are ≈ 3 – 5 vol% from conventional CCGTs (compared to coal-fired plants where CO₂ levels in exhaust gases are around 14%) and thus require much larger CO₂ capture plants to achieve the same levels of reduction, making the application of CCS on CCGTs more expensive [18].

There exists three main paths for capturing carbon from emissions of power cycles classified into either decarbonising the input fuel to produce hydrogen for use in integrated gasification combined cycles (IGCCs), or extracting CO₂ from the combustion products,

or manipulating the combustion to produce a stream of CO_2 which can be easily removed (discussed in Section 2.1) [19]. For the applicability of CCS technologies to the consumption of natural gas, hydrogen production via methane steam reforming or partial oxidation can be considered. However, the processes themselves require carbon capture because of the water-gas shift reaction to convert the produced carbon monoxide to more hydrogen and thus more carbon dioxide. It is generally considered that the most economical way of capturing CO_2 from methane/ NG is to first reform the gas to H_2 and then capture the CO_2 before combusting the H_2 . However, because of the intricacies associated with the pre-combustion method which involve the syngas production method and the issues due to the high reactivity of the hydrogen fuel which incur safety considerations, this method is not explored further for its application to conventional CCGTs. Oxy-combustion cycles are modified CCGTs and are discussed in detail in the following Subsection (1.3.1) and in Section 2.1.

Post-combustion capture requires the use of chemicals to remove the carbon from the flue emissions. Several technologies exist which include using solid sorbents or liquid ammonia for absorption separation, utilising adsorption separation such as calcium/ chemical looping, employing membrane separation, and implementing cryogenic separation via CO_2 liquefaction [19]. The following two technologies have been assessed as having the greatest potential for rapid decarbonisation of CCGTs: 1) amine scrubbing and 2) calcium looping. The clear winner of the commercially available CCS post-combustion technologies is amine scrubbing.

Amine Scrubbing

Traditional amine scrubbing utilised monoethanolamine (MEA) solutions to capture CO_2 from flue gases. However, these days all commercial amine scrubbing systems, and those at a high technology readiness level (TRL), utilise more advanced proprietary blended amine solvents (Shell's CANSOLV, CESAR-1, MDEA methyldiethanolamine, DEA diethanolamine, piperazine, AMP 2-amino-2-methyl-1-propanol) [20, 21]. Some work has been undertaken into the use of solid amine based capture systems, though these have not progressed beyond the laboratory yet. The focus of today's research efforts into the solvents themselves is to minimise the volatility of the solvent, reduce the energy penalty for desorption, and minimise its potential for degradation or side reactions upon contact with SO_2 , NO_x , and other impurities [22].

Current state-of-the-art efficiency penalties are in the order of 8% (efficiency points lost) for post-combustion capture by amine solvents on gas combustion power plants and are not expected to decrease significantly in the near future [23]. Heat integration within the gas combustion process is likely to be a primary option for decreasing the CO_2 capture efficiency penalty. Amine scrubbing for CO_2 capture remains an expensive technology to apply, capital costs can be up to \$1,500/plant kW [24]. It is currently utilised mainly on coal-fired plants at several sites, most notably at the Boundary Dam coal power station in Canada, which has been in operation since 2014 and captures $> 1 \text{ MtCO}_2/\text{y}$ [25].

Estimates of the cost of CO₂ avoided are in the range of €40-90/tonne, and would be significantly higher on gas-fired plants [26, 27].

Calcium Looping

Calcium looping is regarded as a second generation post-combustion carbon capture technology. It is the process of circulating solid CaO and CaCO₃ particles between two fluidised bed reactors such that CO₂ is captured in one and released in the other. Calcium looping has an estimated efficiency penalty of only 6 – 8% points lost which presents a slight economic advantage over amine scrubbing [28]. Furthermore, as this technology is still at a pilot demonstration scale, there is potential for the efficiency penalty to decrease further.

Some modelling work has been conducted on the integration of calcium looping in CCGTs showing that the high temperature heat generation in the carbonator (exothermic process) enables the provision of an additional steam cycle. The implementation of this would reduce the efficiency penalty by a quarter but increase the cost of electricity by a third [29] compared to the simple cycle without the additional heat recovery. Estimates of the cost of CO₂ avoided are in the range of €10-50/tonne [30]. The main limitation of this technology going forward is still the rapid degradation of the CO₂-carrying capacity of materials over multiple cycles, and the low CO₂ partial pressure from natural gas combustion.

Post-combustion carbon capture is presented here for background information to the alternative oxy-combustion method (discussed in 2.1) of the cycle adopted through this work.

1.3 Motivation

While improvements in CCGT cycles are possible, cycle efficiencies have not greatly changed over the years and have reached an asymptotic growth. The cost of capturing the CCGT-related emissions is high and the technology is limited. Other gas-fired cycles, including hydrogen combustion, are hitting the limelight for various reasons noticeably either being better cycle efficiencies or cleaner power cycles and in some cases a combination of both motives.

1.3.1 Overview of Potential Gas-Fired Cycles

This subsection provides a general, high-level overview of the current and novel future gas-fired cycles, some of which are under active development while others are at a low-level research with only process and techno-economic studies available. These cycles are

presented in Table 1.2 which summarises some details of the technologies in question. The purpose here is to compare the strengths and weaknesses of the advanced cycles, which are still in various stages of development, with a conventional combined cycle gas turbine plant, which itself is still capable of some incremental improvement as described in Subsection 1.2.1.

Table 1.2 Overview of current and future gas-fired power generation cycles (citations in text)

Cycle	CCS (rate)	Status	Efficiency [%]
CCGT	Post-combustion	Commercial	60 (54 with CCS)
sCO ₂ Closed	N/A	Pilot- scale	Cycle dependant
SCOC-CC	Integral	Research	45 – 48
MATANT	Integral	Research	40 – 49
NET Power	Integral	Demonstration	55 – 59
Graz	Integral	Ready for demon- stration	49 – 54
CES	Integral	Demonstration	45 – 50
AZEP	Integral	Aborted	49 – 53
ZEITMOP	Integral		46 – 51
SOLRGT	Integral	Proposed	52.9
CLC	Integral	Demonstration	50 – 52

Other than the conventional CCGT, the closed supercritical CO₂ (sCO₂) cycle, and the chemical looping combustion (CLC) cycles, most of the other cycles can be classified as oxy-combustion ones differentiated either by their recycled fluids to moderate combustion temperatures and/or to cool hot gas path components, or by their source of oxygen. CCGTs are characterised by their simple design and construction plus their low operating costs and fast start-up times [31].

The closed supercritical CO₂ cycle uses the special properties of that working fluid in a closed loop with a turbine, to convert heat (often waste heat in the exhaust of other processes) into electricity using a high efficiency cycle. The cycle has been developed to a high level for solar thermal applications and is under development for several other applications including as a retrofit in coal power plants and to improve the performance of marine gas turbines; the partial cycle is described in more detail in Section 2.2. Critical issues within the cycle include the design of new turbine to suit the working fluid properties, and the material degradation associated with the replenishment of low-grade CO₂ [32, 33].

The semi-closed, oxy-combustion combined cycle (SCOC-CC), the Matiant/ E-Matiant cycles and the NET Power cycles all use recycled CO₂, thus providing a high concentration CO₂ working fluid, which in the latter case is at supercritical conditions. In comparison, the Graz and CES (developed by Clean Energy Systems) cycles use water/ steam

to moderate combustion. The AZEP (Advanced Zero-Emission Power cycle developed by Alstom) and the ZEITMOP (Zero Emission Ion Transport Membrane Oxygen Power, University of Twente) cycles are both aimed at using an oxygen transport membrane to supply lower-cost oxygen than that provided by a conventional air separation unit (ASU) [34].

The SCOC-CC is similar to a conventional CCGT, although the gas turbine compressor is used to recompress some of the cooled CO_2 from the turbine exhaust. Combustion takes place at a higher pressure of 45 bar providing combustion gases which are expanded through the power turbine before entering a conventional heat recovery steam generator, from which the superheated steam drives a traditional steam cycle. The exhaust gases are then cooled further to recover most of the water in the exhaust before the relatively pure CO_2 stream can be then processed and compressed for transport and use or storage [35]. The SCOC-CC has not been fully assessed as it is only at a concept stage.

In comparison, while the NET Power or Allam cycle (after Rodney Allam who conceived it) is based on a similar concept, it operates at a much higher pressure of between 200 and 300 bar with a pressure ratio of around 9. This means that the working fluid, which is predominantly CO_2 due to a higher level of recycle, is sub-critical in the turbine exhaust gas. After condensation of water in the exhaust gas, the remaining CO_2 -rich gas requires recompression before it is split between export for use or storage and the recycled stream to the combustor, being heated in high pressure (HP) regenerative heat exchangers with the waste heat in the turbine exhaust before being used to moderate temperatures in the turbine combustor [36, 37]. The main benefit of this cycle is the integrated carbon capture with zero-emissions at high cycle efficiencies, however, it requires a new combustor design and turbine blade geometry designs with cooling paths to handle the operating temperatures of around 1500°C , plus it incurs high cost for the materials of the heat exchanger (HE). The NET Power cycle is described fully in Section 2.3.

The Graz and modified S-Graz cycles are again oxy-fired cycles, consisting of a high-temperature cycle with a gas turbine and a heat recovery steam generator (HRSG), combined with a high pressure steam turbine and a low temperature steam cycle. The combustion operates at around 45 bar, similar to the SCOC-CC, with steam from the HRSG and a recycled exhaust gas mixture of water and CO_2 being used to cool the combustor and the turbine blades [38, 39]. The Graz cycles also require new combustor and turbomachinery designs, and face some corrosion issues.

The Clean Energy Systems (CES) cycle is a further oxy-fired cycle, but using water steam to moderate the combustion temperature rather than CO_2 . The cycle, which could simplistically be considered as a directly fired steam turbine, has been developed over several decades. It comprises a high-pressure oxy-combustor where part of the fuel and oxidant are burned using supercritical steam to moderate the combustion temperature. The resulting hot gas is used to drive a steam cooled high-pressure (HP) turbine, the exhaust gas from which is reheated by a further oxy-combustor before expansion through medium- and low-pressure turbines to below atmospheric pressure [23, 40]. Identified issues involve material selection for the turbine ($\text{TIT} \approx 1300^\circ\text{C}$) and the rocket technology

gas generator. Further details on the benefits and specifications of oxy-combustion cycles are presented in Section 2.1.

As indicated above, the AZEP and ZEITMOP cycles do not use a conventional ASU to supply oxygen, but rather aim to integrate a high temperature transport membrane for oxygen production (OTM) in the power cycle. These membranes require a hot pressurised air stream, from an externally heated air cycle for this purpose, either within the main cycle or as a supplementary cycle [34], and thus are identified as aspects that require further development.

The SOLRGT/ZE-SOLRGT (Zero-emission Solar Gas Turbine) cycles, proposed by the Chinese Academy of Sciences, are based on a novel hybrid system using solar heat to reduce the use of natural gas. In addition, these complex cycles incorporate steam-methane reforming and CO₂ separation and compression, while using steam as the primary working fluid [41]. It is important to note that efficiency levels of the SOLRGT cycle are highly dependent on weather conditions and that the net solar-electricity efficiency is 30%.

The final cycle included in Table 1.2 uses chemical looping combustion (CLC) as an alternative oxy-combustion approach. As combustion of the natural gas is carried out using an oxide particle, rather than air, there is no nitrogen in the exhaust gas stream making CO₂ separation a simple process of condensing the steam to leave high purity CO₂. The high temperature exhaust gas can be used for a variety of cycles and offers good potential for decarbonisation of energy intensive industries [28]. Material degradation occurs over multiple cycles in CLC and if gaseous fuel is to be used then the system must be pressurised to achieve efficiencies similar to CCGT.

1.3.2 Problem Definition

Regardless of the advantages attributed to the cycles mentioned in Subsection 1.3.1, there are many critical issues associated to each one that need to be overcome before actual deployment of the technology. The NET Power cycle is definitely the most attractive of the listed cycles because of its ability to merge the benefits of oxy-combustion and the use of supercritical CO₂ to obtain the highest cycle efficiency while releasing zero emissions.

Although the combustion system has taken huge efforts in the design process of the NET Power cycle, another critical barrier that inhibits the full-scale development of the novel Allam cycle is the design of the high-pressure, high-temperature turbine which dictates cooling requirements, material considerations and number of stages to name a few. The turbine operates at an extreme combination of high temperatures comparable to those of gas turbines, and high pressures analogous to steam plants in the presence of unconventional working fluids. Cooling needs to be implemented within small-sized blades of a proposed and undisclosed axial turbine, developed by Toshiba, which poses hard constraints on the design that can limit aerodynamic performance and manufacturability.

A radial turbine, which generally has a simpler construction and fewer stages when compared to its axial contestant, is suggested as a superior candidate arrangement for such

cycles of high fluid density at smaller power scales (up to 100 MW). For these reasons, the NET Power theme is taken as the basis of a paradigm cycle configuration around which restrictions of a radial turbine design can be applied. A thermodynamic analysis of a mid-range cycle is established while lowering the turbine inlet temperature to remove cooling complications within the radial turbine passages. The cycle conditions are then considered for the design of a multi-MW scale turbine by using preliminary and higher fidelity methods.

1.4 Aim and Objectives

The goal of this PhD research, along with the target tasks, are set out on the rationale stated in the problem definition.

1.4.1 Aim

The aim of this work is to assess the suitability of a radial turbine in a utility-scale, oxy-combustion, supercritical CO₂ power cycle through an investigation of the thermodynamic cycle, aerodynamic design, mechanical analysis, and consideration of enabling technologies. The outcome of the investigation is a preliminary design of a radial turbine that fits within the identified requirements.

1.4.2 Research Objectives

The two main objectives of this piece of work are:

- RO1. Identification of appropriate power cycle conditions (power output scale and operating temperatures/ pressure ratios) suitable for a radial turbine**
- RO2. Characterisation of the flow behaviour within the turbine passages to design a befitting radial turbomachine for high pressure/ high temperature supercritical CO₂ cycle**

To fulfil the outlined research objectives, a set of assessable tasks is identified:

- Task 1.** To identify the gaps in knowledge and gather relevant information by conducting a thorough literature review and communicating with experts in the field
- Task 2.** To define an appropriate cycle model based on requirements and acquire the operating environment of the radial turbine through cycle process modelling
- Task 3.** To obtain a range of overall turbine dimensions and performance criteria by using validated design methods and recommendations found in literature

- Task 4.** To gather a selection of possible preliminary turbine geometries via a systematic investigation of a number of potential turbine designs with specific constraints
- Task 5.** To assess different working designs based on performance requirements by means of throughflow analysis
- Task 6.** To study turbine flow-field and evaluate turbine performance by employing numerical methods of advanced flow modelling
- Task 7.** To investigate the mechanical integrity of the turbine by performing computational stress calculations
- Task 8.** To determine areas for further work based on the availability of enabling technologies

1.5 Thesis Organisation

This thesis is comprised of eight chapters. Chapter 1 covers an overview of the importance of novel power generation and its global relevance, plus the stimulus behind this piece of work. Within the same informative framework, Chapter 2 offers a robust literature review that enables the identification of the knowledge gaps that are tackled within this study. The methodology of research along with software-specific details are explained in Chapter 3. Results chapters are covered between Chapters 4 – 6. The first of those (Chapter 4) presents results of the modified cycle model that are necessary for the turbine design, as well as the basis of preliminary turbine design. Chapter 5 deals with the whole turbine design procedure; performance outcomes of 2D turbine analysis and high fidelity CFD modelling results of an initially selected turbine design are showcased, plus the investigation of the respective flow physics are discussed. The last results Chapter 6 exhibits the redesign step based on conclusions drawn from the initial turbine selection to obtain a more successful design, and presents the examination of static stress analysis of the radial turbine in a supercritical CO₂ environment. A discussion in Chapter 7 is provided to link the whole thesis together and give an overall analysis of the work; it also provides comments on the required technologies for the adoption of a radial turbine in a utility-scale power cycle. The final Chapter 8 draws the conclusion of the thesis and opens the floor for future and additional exciting work.

Chapter Summary

The opening chapter presents some background information on CCGT and relates it to the advantages that the adopted cycle has in terms of emissions, size and efficiency. The chosen NET Power cycle combines the reasons that tackle the main efforts of more efficient and cleaner power production with smaller footprints. In relation to high density flows, the main aim of the project is then to analyse the use of a radial turbomachinery expander,

instead of a conventional axial turbine, by going through the process of designing one; the objectives required to achieve the goal are stated from cycle thermodynamics, passing through turbine design and analysis, and finishing with static stress examination.

CHAPTER 2

Literature Review

This chapter presents a literature review of the technologies used in the NET Power cycle (oxy-combustion and supercritical CO₂) to show what advantages they have and where their hindering issues lie. An explanation of the operation of the Allam cycle is also provided for reference and to allow the reader to note changes made to the cycle at later chapters of the thesis. A critical analysis of the design works produced by other researchers is given which enables both the verification and justification of used methods and obtained results, and the identification of current gaps in knowledge that are addressed in this work.

2.1 Oxy-Combustion Cycles

The challenges arising due to greenhouse gas emissions aggravate the ever-growing threat of global warming. In compliance with the warnings of the Intergovernmental Panel on Climate Change (IPCC) [42] to limit CO₂ discharges and keep the temperature rise below 1.5°C, techniques for mitigating power generation emissions are being widely employed; examples range from using renewable energy sources and increasing the efficiency of power plants, to implementing CCS [43]. As explained by Jordal et al. [44], there are three routes to CCS, which are depicted in Figure 2.1 [19], classified as pre-combustion capture (gasification), post-combustion capture or absorption (explained in Subsection 1.2.2) and oxy-fuel combustion, out of which oxy-combustion proves to be the highest favourable because of the simple separation of carbon dioxide from the steam present in the flue gases [45, 46, 47]. The main idea of this method is that a fuel is combusted with pure oxygen to produce a stream of exhaust gases consisting mainly of H₂O and CO₂ which can be separated downstream through condensation [48].

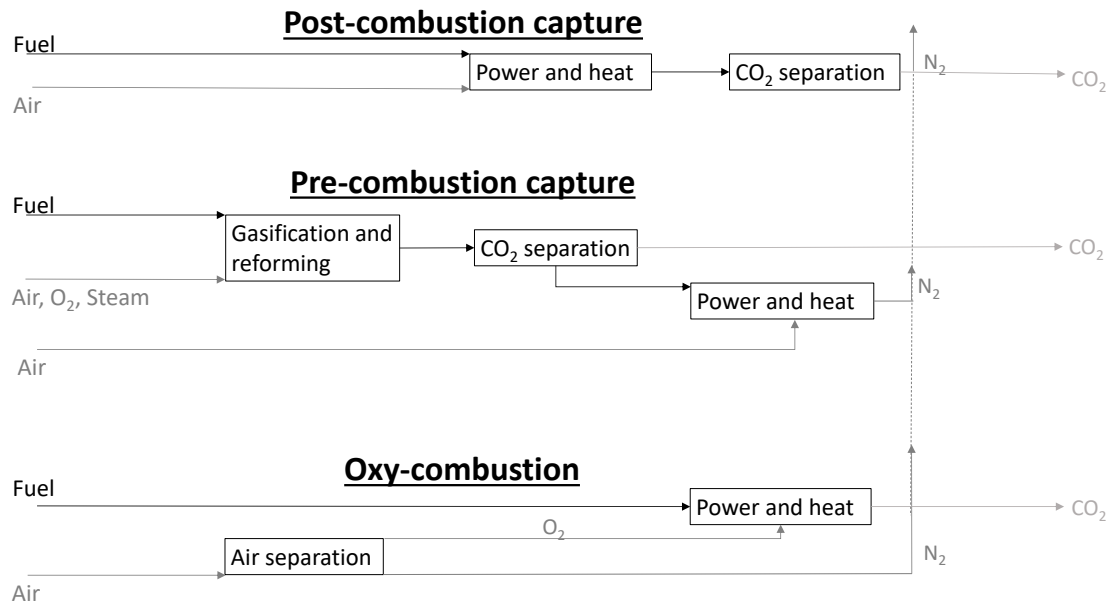


Fig. 2.1 Carbon capture techniques [19]

A review of oxy-combustion cycles has been reported by the International Energy Agency (IEA) identifying the main cycle configurations, requirements, benefits, complications and future developments by presenting cycle modelling and techno-economic study results [34]. A summary of the working environments, challenges and proposed efficiencies of the evaluated cycles (which are also present in Subsection 1.3.1) is tabulated in Table 2.1. The first three listed cycles have CO₂ as the recycle fluid while the Graz and CES use steam as the main working fluid. The cycle with the best theoretical efficiency is the NET Power cycle while the most developed cycle in terms of component readiness is the SCOC-CC. The power plants that fall on the lower scale of efficiencies below 50% are the SCOC-CC and the MATIANT variants. The ZEITMOP cycle is the one which requires the most developmental work, in the largest number of components, specifically in the OTM technology which is considered to need a technological "break-through" [34]. Although the NET Power has the most superior cycle efficiency score, it acquires a drawback of development penalty mainly from the turbine where a dedicated design with major modifications to current commercial turbines is required; the combustor and regenerator require less adjustments to conventional designs [34, 37]. Moreover, the CES cycle maintains a relatively high efficiency (around 50%), and is more mature than other cycles; its principle difficulties lie in the high temperature turbine and the gas generator which do not necessitate a quantum leap, but demand a more focused design based on process parameters [49].

Table 2.1 Summary of different oxy-combustion cycle characteristics [34]

Cycle	Working environment	Challenges	Efficiency [%]
SCOC-CC (1998)	TIT = 1400°C Π = 30 – 45 TOT = 580 – 620	High pressure ratio Minimum cycle pressure Flue gas cooling temperature Turbomachinery design	45 – 48
MATIAN (1997)	TIT = 1300°C	CO ₂ turbines Regenerator CO ₂ condenser	40 – 49
Allam (2010)	TIT = 1150°C TIP = 300 bar TOT = 775°C TOP = 30 bar	Turbine Combustor Economiser heat exchanger Condenser	49 – 54
Graz (1995)	TIT = 1500°C Π = 40	Turbomachines Combustor Condenser Evaporator	49 – 54
CES (2000)	HPT TIT = 900°C HTT TIT = 1328°C	Turbine Gas generator Reheater	45 – 50
AZEP (2001)	TIT = 1140 – 1275°C	HT HE OTM permeator	49 – 53
ZEITMOP (2004)		OTM HT HE HP CO ₂ heating CO ₂ turbine	51

Barba et al. [50] also evaluated and performed a political, economic, social, technological, environmental, and legal (PESTEL) analysis on the oxy-combustion power cycles reported in Table 2.1. From a political perspective, the oxy-combustion cycles require attractive legislative measures for investment. Socially, the survey carried out by the authors of the study [50] showed that the public is more inclined towards the widely known renewable technologies rather than CCS methods. Besides the CCGT, which is not a CCS oxy-combustion cycle, the NET Power cycle shows better marks technologically,

economically and socially. The runner-ups, in terms of the sum of the PESTEL factors given to each cycle, are the E-MATANT, followed by the CES. Both the CES and the NET Power cycles have been regarded with an "advanced-status" owing to the fact that they are being developed at a pilot/ demonstration scale [50]. Oxy-combustion cycles have underlying challenges in terms of oxygen production and purity, and the design of components such as turbines and boilers which affect the final cycle efficiency.

2.2 Supercritical CO₂ Cycles

Initial studies on supercritical CO₂ cycles date back to 1948 with Sulzer Bros [51]; since then, interest has been rising due to the advantages that the working fluid brings along. One of the early investigations was performed by Feher in 1968 [52], he developed a thermodynamic cycle that eliminates the problems associated with the Rankine plants (large footprint for high pressure ratios) and Brayton cycles (high compression work requirements), and incurs additional benefits. The proposed cycle operates above the critical point of carbon dioxide, which has a critical temperature and pressure of 30.98°C and 7.38 MPa (73.8 bar) respectively. The attractiveness of the use of sCO₂ cycles is based on the availability and inertness of the working fluid as well as the unanimously cited advantages by researchers in the field [53]:

- The nature of the high density sCO₂ fluid means that power conversion systems are very compact compared to conventional power cycles [54] (≈ 30 times smaller than steam cycles and ≈ 6 times smaller than air cycles)
- High cycle efficiencies are also obtained, attributed to lower pressure ratios and the large potential for internal heat recovery (i.e. high turbine exhaust temperature and low compressor/ pump delivery temperature), and the non-ideal gas properties of sCO₂ which translate to lower fuel consumption and lower capital and operating costs. Targeted efficiencies of large-scale, closed supercritical CO₂ Brayton cycles approach 50%
- The suitability of the cycle for waste heat recovery from a variety of heat sources (e.g. nuclear, solar, fossil, geothermal)
- Application of these cycles reduces greenhouse gas emissions due to their increased efficiency and the possibility of using an emission as a working fluid or as a recycle
- The suitability of sCO₂ cycle systems for use in arid climates because of their ability to use dry cooling (thus saving water) [55]
- Low purification requirements for air leakages because the cycle operates above the critical pressure of CO₂

Although supercritical CO₂ cycles provide unique benefits, they face downsides particularly with regards to the instability of the non-ideal gas specific heat near critical point which influences heat recuperation. The combined effects of pressure ratios and high TITs create a large amount of heat difference between the cold and hot sides of the heat exchange system that needs to be recuperated inducing a pinch-point problem in the recuperator [51]. This phenomenon affects the design of the recuperator which in turn has an immediate consequence on efficiency and size of the component, hence ultimately incurring a negative impact on the cycle economics.

2.2.1 Supercritical CO₂ Cycle Applications

One of the many appeals of the sCO₂ cycle is its adaptability to be coupled with different heat sources, and thus the various applications for its use. Direct-fired or semi-closed cycles are of the oxy-combustion type which adopt the technology discussed in Section 2.1, whereas indirect-fired or closed sCO₂ cycles are integrated to other heat systems (many of which fall under the renewable energy category) linked through a system of heat exchange. Potential applications are visited in the following headings.

Solar Thermal

Supercritical CO₂ cycles are considered for integration with large-scale concentrated solar power (CSP) plants because of their ability to reduce costs while increasing cycle efficiency. Economic integration to CSP applications is attributed to the high temperature difference between the single-phase hot and cold flows of the heat exchanger. Research at the National Renewable Energy Laboratory [56] agrees well with that undertaken by Coco-Enriquez et al. [57] on the benefits of integrating CSP with partial cooling recompression supercritical CO₂ cycles because of smaller recuperator sizes and high thermal efficiencies.

Geothermal

Supercritical CO₂ cycles are also being considered as a possibility for using geothermal heat sources [58, 59]. The ECO₂G system of GreenFire Energy [60] takes advantage of the properties of sCO₂ that enable easier accessibility to the plastic, hotter geothermal zones which steam is not capable of reaching when hydrothermal systems are used. The feasibility of using the supercritical fluid reduces parasitic and compression power losses, and drilling risks, thereby lowers the capital cost of the system. Sabau et al. [59] showed the interdependency between heat exchanger area (cost), pressure ratios, and TITs in geothermal sCO₂ cycles; efficiencies increase with higher turbine inlet temperatures but that leads to large evaporator areas which requires high pressures.

Nuclear

Supercritical CO₂ cycles also extend the realm of their application to encompass nuclear energy heat sources; research focus is given mainly to small modular (SMR) sodium fast-cooled reactors (SFR). Studies found by Ahn et al. [42] indicate that for SFR applications, sCO₂ Brayton cycles compete highly in terms of efficiency and simple single-stage turbomachinery design with other Brayton cycles (helium or nitrogen). Yoon et al. [61] also advocated the promising advantages of coupling sCO₂ cycles to SMRs which include better performance than water-cooled reactors, lower economic risks, easier operability and the capability of system transportation.

Waste Heat Recovery

Bella et al. [62] indicate that power plant thermal efficiency improvements can be achieved through waste heat recovery (WHR) from gas turbines through the utilisation of sCO₂ cycles instead of large steam cycles. An compact and economic 9 MW_e commercial WHR power generation system has been developed by Echogen which avoids the use of water [63].

Fuel Cell

Besides having the potential to compliment the previously mentioned heat sources, sCO₂ Brayton cycles can also be applied as bottoming cycles for molten carbonate fuel cells (MCFC). Both Ahn et al. [58] and Sanchez et al. [64] emphasise on the recompenses of the hybrid MCFC-sCO₂ system in terms of high efficiencies and simple integrations; up to a 10% increase in efficiency compared to orthodox systems using hot air turbines is reported.

2.2.2 Barriers to Progress for Supercritical CO₂ Cycles

Currently, sCO₂ Brayton cycles (with $T_{max} \leq 500^{\circ}\text{C}$) are at a technology readiness level of 4 – 5 while cycles operating at higher temperatures still require major developmental work (TRL $\approx$ 3) [65]. Technology improvements for target temperatures depend on component development and several crucial barriers that need to be overcome [66]:

Materials

- Material performance has not been tested for temperatures above 700°C (for high efficiency) and the high cost of possible required materials to ensure meaningful lifetimes could impact industry implementation.
- Corrosive degradation of materials throughout the cycle components presents significant hindrances to commercial adoption. The mixture of CO₂ with any trace

steam (or with oxy-combustion products in the case of direct-fired cycles) at high pressures poses severe corrosion issues as noted by Laumb et al. [67].

- Sandia National Laboratories [68] have identified erosion mechanisms in the primary heat exchanger, the turbine inlet piping and castings, and the high temperature recuperator, where impurities have been present in the sCO₂ fluids. These problems will affect component lifetimes as commercial scale plants will not be able to use high purity CO₂ gas.
- Joining dissimilar metals: there is limited experience with advanced nickel alloys for these applications to manufacture cycle components.

Heat Exchangers/ Recuperators

- A high capital cost is associated with the use of advanced materials (for example in printed circuit heat exchangers (PCHEs)) that can withstand oxidation and corrosion, thermal cycling and fatigue.
- Fouling, which reduces heat transfer, increases pressure drop, and increases coolant mass flows, has been deemed possible in sCO₂ heat exchangers.
- There is a lack of information proving the durability of PCHEs in creep, fatigue and oxidation/corrosion for temperatures greater than 700°C.
- There is a need to use thin wall components for high temperature heat exchangers to maximise performance and reduce fatigue damage, but at the risk of oxidation/corrosion failures.
- Compact heat exchangers with solid-state diffusion bonding, instead of conventional welding, (for size reduction and high pressure capabilities) lead to mechanical stiffness which requires thermal shock/ cycle and fatigue testing. Additive manufacturing is being sought for sCO₂ heat exchangers but the technology still requires a scale-up [69].

Heat Rejection

- Large component dimensions result in significant pressure losses; the development of smaller dry systems is needed to lower the cost and complication of wet-cooling.
- To meet compressor inlet temperature constraints, there is a need for advanced cooling mechanisms in the heat rejection heat exchangers.

Turbine

- There is limited data available for sCO₂ turbine performance.

- Alternative designs or new high temperature materials, that can sustain the erosion found in developmental turbine nozzles and blades, are required.
- Improved, tailor-made bearing designs are required to run at high temperatures and pressures and at relevant scales for sCO₂ cycles [70].
- Novel sealing technologies for high temperature and high pressure operation are needed to contain the pressurised highly-dense solvent-like fluid [71].
- Turbine experimental testing necessitates high-cost for complete assemblies.
- Understanding of all damage and performance loss mechanisms, and their magnitudes, is necessitated.

Compressor

- Proximity of compressor conditions to the critical point means that compressors would need to operate with sub-critical, transcritical and supercritical fluid environments which might induce two-phase flows and condensation [72].
- Control issues, to maintain the appropriate balance between both the main compressor and the recompressor for some high efficiency recompression Brayton cycles, arise through parallel compressor operation.

In addition to the above technical challenges, the cost of development and the optimization of the resulting LCOE versus cycle efficiency are crucial if the technology is to be implemented. Poor perception of sCO₂ cycles by potential operators and plant owners might require the need for policy intervention to encourage the adoption of these new technologies, and to avoid them from being neglected in favour of conventional systems. Several external factors, such as market demand and the competitiveness of alternative technologies, facilities planning and safety, and supportive/ restrictive policies also need to be considered.

2.3 NET Power Cycle

The NET Power cycle combines both originalities of oxy-combustion and supercritical CO₂, and because it is the basis of the cycle under study, this section provides a more thorough description of its operation and technological obstacles.

2.3.1 Overview of NET Power Cycle

This cycle is aimed at high power generation efficiencies coupled with high levels of CO₂ capture. An Allam cycle plant (named after its inventor) will provide a high purity CO₂ stream directly, requiring compression and minimal extra clean-up compared to that for

a conventional CCGT coupled to a post-combustion CO₂ capture plant, thus reducing the economic and efficiency penalties for introducing CCUS greatly. The extent to which these penalties are negated will depend on the ultimate efficiency and CO₂ export conditions achieved. In addition, an Allam cycle plant will be much smaller than an equivalent CCGT plant for the same output, due to its higher operating pressure (1/3 smaller than natural gas combined cycle). No steam elements (high, intermediate and low turbines, steam generator, piping) nor emission control systems are needed, thus offering major benefits of potential lower costs (\$/kW) and a smaller footprint.

A specific application of the Allam cycle is the NET Power cycle, which is a semi-closed, direct oxy-combustion, high pressure, low-pressure ratio, regenerative cycle that employs supercritical CO₂ as a working fluid. The flow diagram shown in Figure 2.2 together with the pressure-enthalpy (P-h) chart of Figure 2.3 illustrate the processes that occur within the cycle. Natural gas fuel or coal and oxygen from an air separation unit are combusted to high temperatures reaching 1200°C at high pressures (≈ 30 MPa) with the recirculated CO₂; due to the absence of N₂ in the combustion process, the flue gases are mostly composed of CO₂ and H₂O with no formation of NO_x (for a natural gas based cycle). The combustion products are then expanded in the CO₂ turbine with a pressure ratio between 6 and 12 before entering a multi-flow economizer heat exchanger where the turbine cooling flow and the recycled CO₂ and O₂ streams to the combustor are pre-heated using the residual flue gas heat [73]. The exhaust stream is also cooled for condensation and separation of water from the mixture. A fraction of the remaining (mostly pure) CO₂ stream is sent for carbon capture and storage while the rest is re-compressed and used as a temperature moderator in the combustor through mixing with the O₂ coming from the ASU [37].

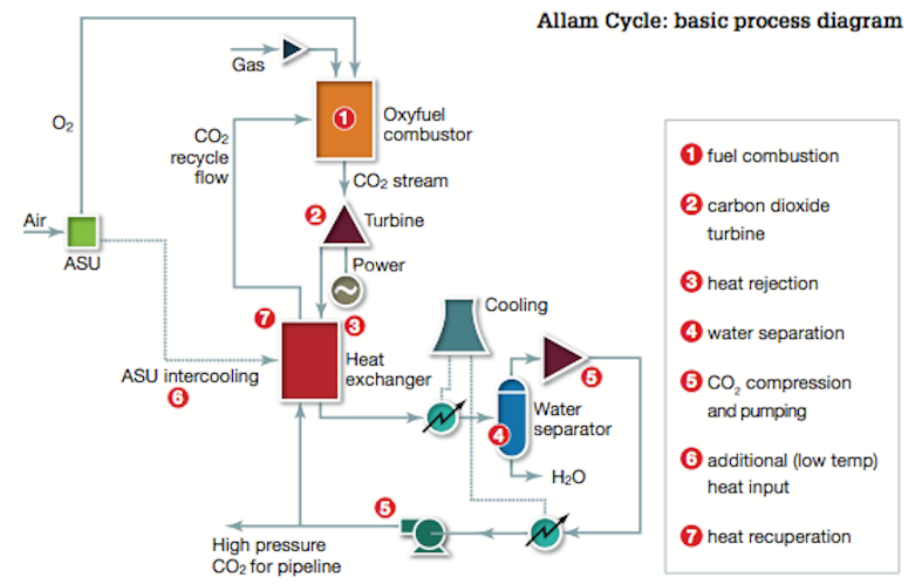


Fig. 2.2 Flow diagram of natural gas NET Power cycle [36, 37, 50]

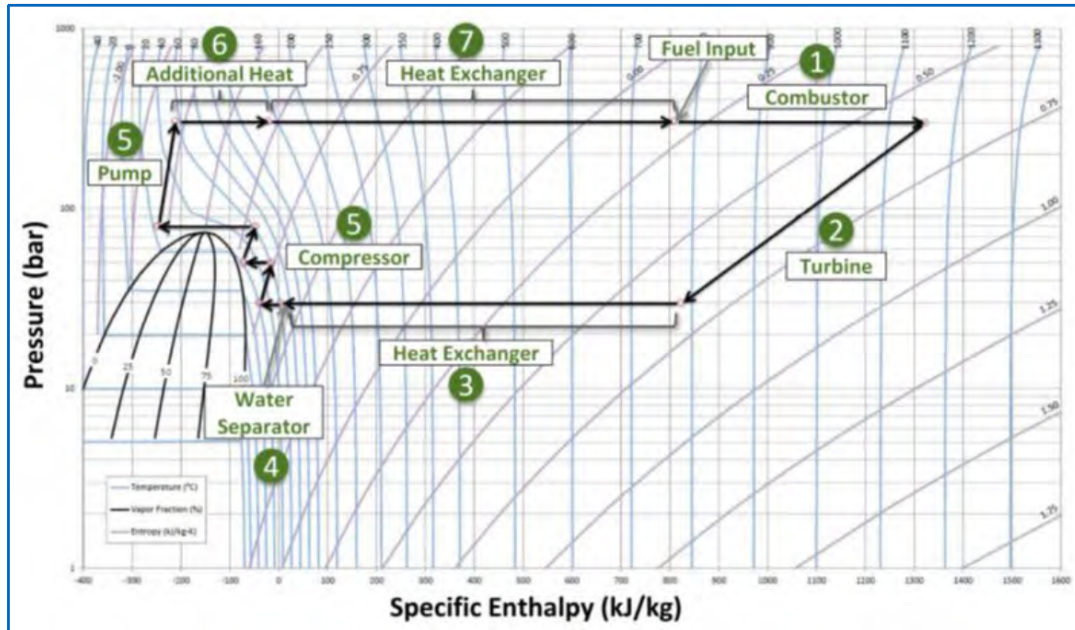


Fig. 2.3 P-h Diagram of NET Power Cycle [34, 36, 37]

Cooling of the combustor and turbine components to allow operation at the relatively high turbine inlet temperature of 1200°C is more complex than in a conventional CCGT where cooling air can be bled directly from the compressor section. For the Allam cycle, the available cooling fluid is the CO_2 stream unless a closed loop system is implemented using a different fluid, e.g. steam, although this would add cost as there is no steam available in the cycle. This constraint could limit the cycle from further efficiency improvements through increased turbine entry temperature due to the counterbalancing cooling penalty. The sensitivity to coolant flows is one of the reasons cooling is eliminated from the adopted cycle of this PhD work.

The purity level of the CO_2 capture stream depends on that of the O_2 ; the optimal performance of the plant is achieved with highly pure oxygen (99.5 %mol) from the ASU. A near-zero emission cycle with higher CO_2 capture rate (98%) than a base case with capture rate of 90% can be obtained with the addition of a membrane in the CO_2 compression and purification unit (CPU) but an efficiency percentage points penalty (-0.4%) is induced [37]. Due to the substantial temperature and pressure imbalance within the regenerator, Allam et al. [36] suggest using external heat sources from compressors or the air separation unit. Efficiency figures are around 55% on lower heating value of fuel (LHV) in open literature cycle data, although NET Power claims an efficiency of 59% LHV based on undisclosed information.

The expected performance characteristics of a full-scale NET Power plant are listed in Table 2.2; the initially projected year of operation was 2020 but with further testing required the date has been pushed to 2022 [74].

Table 2.2 Projected performance characteristics of full-scale NET Power plant [36, 75]

Parameter	Expectation
Net power output	303 MW _e commercial plant
Net target efficiency (LHV)	59%
Fuel	Natural gas
Thermal input	511 MW (LHV)
O ₂ consumption	3555 MT/day
Turbine flow rate	923 kg/s
Turbine inlet conditions	P= 300 bar & T= 1158°C
Turbine outlet conditions	P= 30 bar & T= 727°C
O ₂ plant power	56 MW
CO ₂ compression power	77 MW
Specific cost	1068 €/kW [34]
Levelised cost of electricity (LCOE)	73.8 €/MWh at 8 €/GJ (NG LHV) [76]

2.3.2 Areas for Development for NET Power Cycle

Considerations related to supercritical CO₂ cycles listed in Subsection 2.2.2 are applicable to the NET Power cycle as well; more specific issues associated with turbomachinery, heat exchangers, combustors and auxiliary components pertinent to the Allam cycle are described below [34, 66].

Combustion Requirements

Due to very high pressures (≥ 20 MPa) and temperatures, the Allam cycle oxy-combustor is closer to a rocket engine than a conventional combustor, hence it requires an innovative design [34]. Several issues have been flagged in relation to the design relating to injector design, heat transfer and combustion dynamics; there are various partner working with the US National Energy Technology Laboratory Fossil Energy programme, and focusing on oxy-combustion computational modelling which is crucial for the development of the technology for use in direct fired cycles [77].

Economizer Heat Exchanger/ Recuperator

The heat exchanger cost is a major limitation because of its large size, high operating temperatures, and the high pressure differentials within heat transfer fluids. The wide variation in specific heat capacities due to differences in pressures and temperatures allow multiple pinch-points within the heat exchanger which restrict heat transfer and thus require careful design [34].

Turbomachinery Considerations

There are various aspects to the issues related to turbomachinery components of the NET Power cycle, much of which stem from the supercritical CO₂ working environment. Some of these topics are discussed again later in the thesis (Section 7.2) to relate them to the radial turbine design at hand.

- **Requirement of materials to sustain the high turbine temperatures $\geq 1000^{\circ}\text{C}$:** While component life is affected by mechanical (creep, fatigue) and thermal properties (conductivity, thermal expansion coefficient), the selection of materials is based on cost, properties, fabricability and availability of the materials and thus a compromise might be necessitated. Oxidation, erosion and internal carburization are aspects that also need to be addressed; high temperature testing is not yet reported and is limited to temperatures lower than those present in the Allam cycle, so it is not clear that the materials challenges can yet be overcome. The use of high temperature materials for an increase in combustor outlet temperature (i.e. increasing metal temperature from 860°C in the base case to 950°C) raises the net electrical efficiency from 55.1% to 56.7%. Toshiba suggests the use of Nickel (Ni)-based materials for the rotor centre and CrMoV forgings for the blade ends; the final products are predicted to be expensive due to the cost of the materials and the difficulties associated with their manufacture [36].
- **Turbine rotor and blade thermo-mechanical design:** Consideration has to be given to the mechanical integrity of the turbine due to high blade loading for the required pressure ratios. Integration of cooling for operation at elevated temperatures induces high thermal stresses due to the higher heat flux when using sCO₂ compared to flue gases of conventional cycles. The turbine blades require a new design methodology compared to the approaches used for CCGTs due to the increased stresses associated with operation. The turbine design needs to accommodate the working fluid expansion from supercritical to sub-critical conditions and the relevant thermodynamic changes [66].
- **Power plant size, operational requirements and economics:** Economic metrics including LCOE, internal rate of return (IRR), and net present value (NPV) are closely associated with cycle efficiency, economic viability for heat source integration, and power production scale.
- **Working fluid contamination:** This is due to the nature of the oxy-combustion cycle which may have an effect on the degradation of materials and would need fluid clean-up. Acute cases of erosion have been reported in turbine blades in test loops using pure supercritical CO₂ [78], which means that these deteriorating effects are going to be more pronounced when the fluid is made up of a mixture of combustion components.

- **Ancillary technology (bearings/ shaft end seals):** The selection of bearings depends on cost, duty cycle, load, speed, weight/ size, efficiency and dynamic performance. Bearings are also affected by high surface speeds and high unit loading; for small power scale cycles, the turbomachinery would run at rather higher speeds than the proposed utility-scale ones which would run synchronously. The choice of sealing technology is necessary to maintain tolerable levels of primary flowpath escapement at elevated temperatures and pressures for fluid containment which differ from conventional technologies due to the solvent-based nature of the sCO₂ working fluid.
- **Starting:** For air-firing combustion start-up, additional compressors are needed to achieve pressure of recycle gas with part-load until oxygen is produced from the ASU at acceptable purity. Oxygen-firing start-up does not require extra components but a large CO₂ inventory is needed for re-circulation. Start-up times for the oxy-combustion cycle resemble the ASU time range of less than an hour for a few minutes of shutdown, to 36 hours in the case of ASU total shutdown for defrost.
- **Manufacturability:** The compact turbomachinery sizes of the Allam cycle plant pose additional challenges for the manufacturing of the critical turbomachinery components, due to their small dimensions and complex internal cooling designs.

Many of the described limitations have not been tackled thoroughly and one of the defining limitations is the design of the NET Power turbine. No public information is accessible for such design and thus the topic is selected for further investigation with a different turbomachinery proposition.

2.4 Modelling Tools

There are a series of methodical steps that are required for designing a turbine that is fit for a cycle similar to the NET Power cycle. This work covers a fully numerical methodology which needs verification from other works. An overview of the methods and tools used by other researchers to design a turbomachinery component is given here.

2.4.1 Cycle Modelling

Before attempting a turbine design, a cycle definition is needed to realise the working environment and the boundary conditions of the turbine, especially that the cycle used in this work is adapted from the original NET Power and has some modifications. Cycle performance modelling for the targeted power plant and components has been done by several authors. Banerjee et al. [79] simulated an oxy-combustion process in Aspen Plus software using two reactors to resemble fuel pulverisation and combustion. Researchers working on the IEA GHG report [34] used a gas turbine performance software developed

by Politecnico Di Milano, GS software, to model all oxy-combustion cycles listed in Section 2.1, besides the NET Power and supercritical CES cycles. The GS tool is aimed at performing design point, 1D aerodynamic, thermodynamic and geometric calculations for turbines with ideal gases. Therefore, because the mentioned code does not handle non-perfect gases and requires geometric design input (which is not available for the NET Power cycle), Aspen Plus was used for simulating the Allam and CES cycles [34, 73]. A continuous cooled-expansion model was developed to simulate the cooled Allam turbine, but since in this piece of work the turbine is assumed uncooled, then a single turbine model is implemented as explained in Section 3.1. Similarly, technical evaluations of other oxy-combustion cycles also employed Aspen Plus for modelling component blocks [48, 80]. Therefore, the same process simulating tool (Aspen Plus) is used in constructing the selected cycle for its relevancy and simplicity of use.

2.4.2 Equation of State

In oxy-combustion cycles, the working medium does not compare to neither that of steam cycles, where water is the main working fluid, nor to conventional natural gas-fired power plants where air or flue gases constitute the fluid environment. Due to the negligible presence of nitrogen in the oxy-combustion cycle, the main composition of the working fluid is CO_2 constituting at least 80% of the total gas composition [81]. This means that the characteristics of the gases in oxy-combustion plants are highly influenced by carbon dioxide at the specified states and are affected by the impurities that are present. Key parameters that influence the turbine design, such as isentropic exponents, density and speed of sound are dissimilar to fluids in conventional gas cycles, thus an orthodox turbine design belonging to the CCGT is not suitable for use in a cycle like the NET Power. The Allam cycle has a unique working fluid environment mixture around the turbine which is presented in Table 2.3 [34] where sCO_2 ($> 90\%$) is the main gas. The blend of all present chemical compounds, containing the combustion-derived flue gases and other traces of natural gas, affects and alters the thermodynamic properties responsible for heat transfer in the cycle components.

Table 2.3 Composition of NET Power turbine flue gases [34]

Fluid component	Unit	Value
CO_2	%	92.4
H_2O	%	5.80
N_2	%	1.10
Ar	%	0.50
O_2	%	0.20

Thermodynamic properties of the targeted working fluids at the operating temperatures and pressures of the NET Power cycle are not reported in literature; approximations

using property methods or assumptions considering pure substances are used instead to calculate thermo-physical properties of the fluids. The behaviour of supercritical carbon dioxide departs from that of an ideal gas (eg. air) specifically at high temperatures where its compressibility factor drifts from the ideal factor of 1 [81]. Consequently meaning that ideal gas properties are not applicable for the working fluid of the NET Power cycle, so equations of state (EOS) pertinent to real gases are incorporated for calculating the important parameters affecting component design both at the cycle modelling and turbine design stages.

Studies on the accuracy of various cubic equations of state, such as the Benedict-Webb-Rubin-Lee-Starling (BWRLS) or the Soave-Redlich-Kwong (SRK) or the Peng-Robinson (PR) EOS, in calculating real gas critical parameters, were performed by Mantovani et al. [82] who reported that the Peng-Robinson model has the lowest average deviation compared to other EOS parameter values. They also proclaimed that PR is the most accurate property method in approximating experimental data for CO₂ binary mixtures. Scaccabarozzi et al. [73] also evaluated three equations of state for the calculation of the dew point pressure and density of mixtures consisting of CO₂ with fluids from Table 2.3 and compared their data with experimental values. The PR EOS proved its superiority over the other models in attaining the targeted properties of mixtures in the sCO₂ temperature and pressure range. The authors hence used the Peng Robinson in the Aspen Plus simulation of the NET Power cycle.

Whereas literature information for the NET Power cycle modelling show the use of PR EOS, many researchers including Sauret et al. [83] and Lee et al. [84] have used the thermodynamic library REFERENCE Fluid PROPERTIES (REFPROP) developed by the National Institute of Standards and Technology (NIST), during their supercritical CO₂ turbine design processes to obtain real gas properties. The property methods used during each methodology step of this piece of work is based on the conducted literature review and is clarified in Chapter 3.

2.5 Turbine Design

Due to the unavailability of open source turbine design characteristics for the NET Power cycle in literature, reference to turbomachines that have some similarities to the one that could be used in the Allam cycle is reviewed for design basis. Oxy-fuel turbines with either steam or carbon dioxide as the recirculating fluid as well as sCO₂ turbines for different applications can be of use as a starting point for the project turbine design work. The features of these turbomachinery components account for the unique working fluid (combination of CO₂ and H₂O at different ratios) and the relevant operating conditions.

Most public domain pieces of information regarding oxy-turbines come from the SCOC-CC. Dahlquist [81] studied the influence of the oxy-combustion working fluid on the design of turbomachinery in his thesis; he found that for a 100 MW turbine power, the compressor is to be directly driven by the power turbine without the use of a gearbox

to minimise cost and associated gearing losses. For cycles with high pressure ratios, a single-spool configuration would not be sufficient to drive the compressor due to the high speeds, thus a gearbox would be necessary. To design his axial turbine, Dahlquist [43] followed a similar design process as adopted in Chapter 3, starting with a 1D design, followed by 2D throughflow analysis and ending with 3D steady-state CFD simulations for assessment.

Yang et al. [32] evaluated the design performance of a SCOC-CC turbine in comparison to a CCGT turbine. Their study focused on obtaining cooling effectiveness through appropriate cooling technology to achieve designated blade temperatures. Cooling is definitely required for oxy-combustion cycles because of the imposed high temperatures and is usually applied to axial turbine configurations similarly to aero-derivative engines.

Fleming et al. [85] have given an indication on turbomachinery type, bearing and seal technology plus motor/ alternator configurations for sCO₂ Brayton cycles based on specific power levels. The reference ranges, portrayed in Figure 2.4, are mainly intended for closed-loop cycles for nuclear applications operating at lower temperatures and pressures compared to the NET Power cycle (400 – 750°C and $\approx$ 200 bar). It shows that for higher power requirements, in use for industrial applications (above 30 MW_e), an axial multi-stage single-shaft turbine is the preferred configuration.

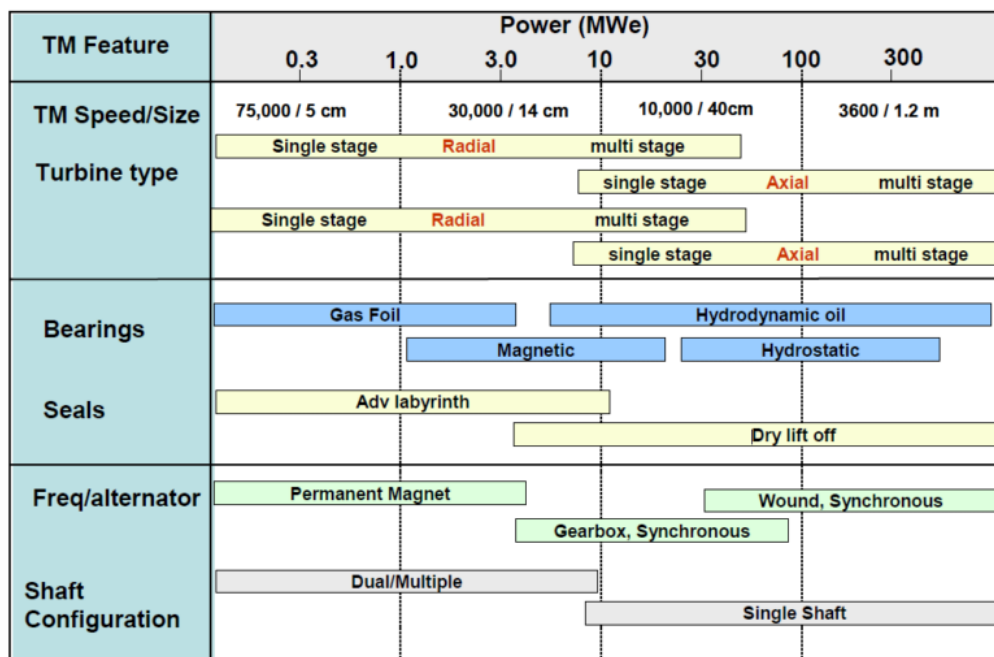


Fig. 2.4 Turbomachinery and component options for sCO₂ cycles [85]

The latest update by Allam et al. [86], on the development of the demonstration NET Power plant, has suggested that partial arc inlet admission is used in the axial demo-turbine to accommodate for the lower thermal input (50 MW_{th}). The high-pressure aspect

of the turbine is designed based on industrial steam turbine science for casings while the high-temperature facet is regarded with gas turbine technologies for cooling.

2.5.1 Radial Turbines

Although axial turbines are the more conventional architectures for expanders used in large-scale power generation, radial configurations are preferred in some cases for several reasons, given by turbomachinery designers and researchers [83, 87, 88, 89], including:

- Simplicity of construction with casting which lowers production costs
- More compact (i.e. smaller) design compared to axial turbines
- Single-stage designs can withstand high pressure ratios (up to 8) compared to an axial configuration which dictates multi-stages
- Can be designed with or without volutes, diffusers or stator nozzles making it more versatile depending on the application
- Efficient and robust for high-fluid density flows
- Better efficiencies for low capacities (small-scales) because of the lower sensitivity to blade shapes and clearances compared to axial turbines

A brief description of the radial turbine and its use in reported works for supercritical CO₂ are discussed next.

In contrast to an axial turbine, where the flow passing through the core remains almost at the same distance from the shaft, a radial turbine imposes a considerable path change for the fluid passing through the rotor. Where the flow stays mostly in the axial direction for the axial turbine, the fluid in the radial turbine enters radially and exits axially. The turning, expansion and radius change occur over a few components that compose the radial turbine which are an inlet volute or scroll, a nozzle row or stator, a rotor, and an exhaust diffuser; the use or absence of the volute, nozzle blades or diffuser depends on the application and design necessities [90]. An illustration of a radial stage sizing station including all the mentioned components and the respective cross-section is displayed Figure 2.5 [90].

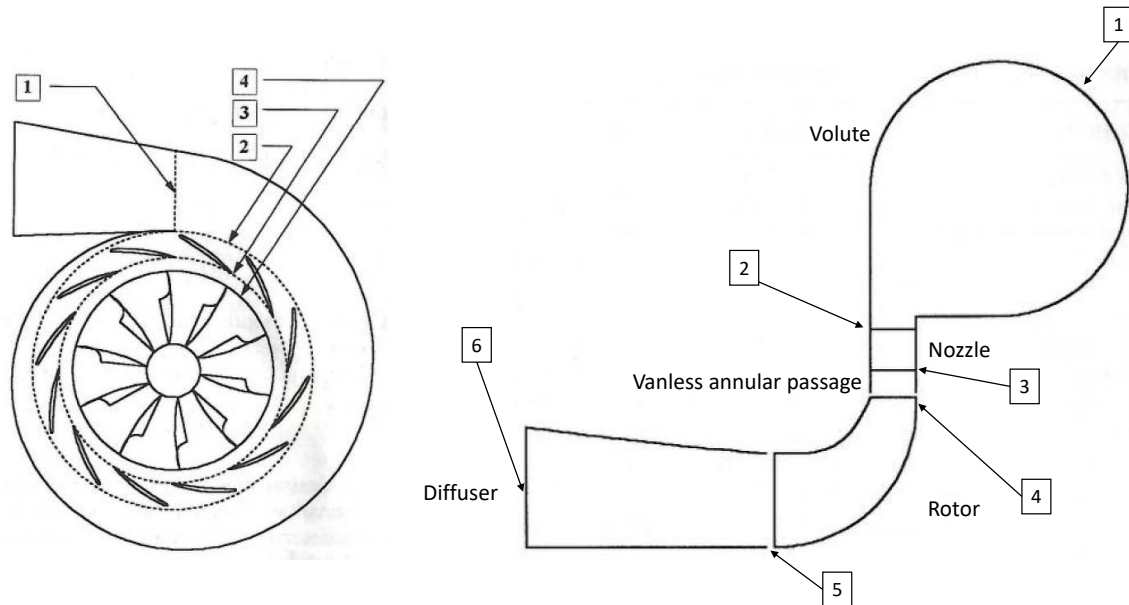


Fig. 2.5 Radial turbine stage sizing stations and cross-section [90]

If a volute (between stations 1 and 2 in Figure 2.5) is present, the flow enters it in a direction which is nearly tangential to the rotor; the role of the volute is to turn the flow towards the rotor periphery (meridional/ axial-radial plane) while adding a large swirl component to the relative velocity. Hence, it delivers a circumferentially uniformly distributed flow to the nozzle blades (if present) and rotor [91].

Another static component of the radial turbine is the nozzle ring (between stations 2 and 3 in Figure 2.5), consisting of a circular set of vane blades which impose the direction at which the flow reaches the rotor blades. The shape of the nozzle blades is dependent on whether a volute is an antecedent to the nozzle or not. In the case where a volute is used to supply the adequate swirl component, then the role of uncambered nozzles is to get rid of the non-uniformity of the flow. On the other hand, in the absence of a volute, the cambered nozzle vanes act in place of the volute to speed up the flow and correct the angle (radial at nozzle inlet) to the rotor [91]. The difference in camber between the two designs is portrayed in Figure 2.6. It is worth mentioning that although a nozzleless turbine is smaller and easier to manufacture, a volute on its own usually does not deliver the required circumferential uniformity to the rotor [91]. Also, according to Baines [91] for pressure/ expansion ratios greater than 3 – 4, nozzle blades are always incorporated within the design.

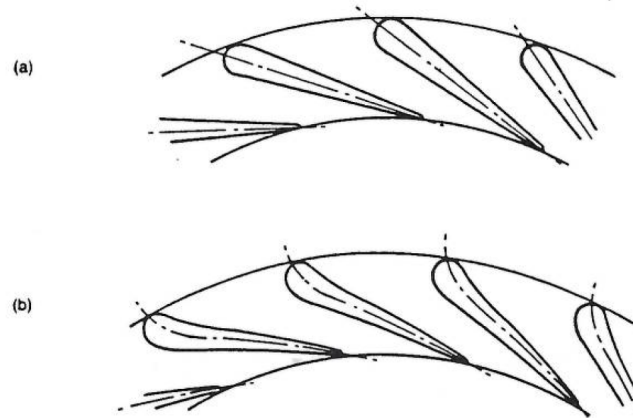


Fig. 2.6 Nozzle blade shapes (a) Uncambered (b) Cambered [91]

The next and most crucial component of a radial turbine (i.e. the element that extracts work) is the rotor blades (between stations 4 and 5 in Figure 2.5). In the case where nozzles are used, a minimum radial clearance between the trailing edge of the nozzle vanes and the rotor tip is necessary; this vaneless annular passage accelerates the flow towards the rotor. While the flow passes through the impeller, it is turned by the shape of the blades, which can be indicated in Figure 2.7, towards an axial direction when leaving the rotor with zero outlet swirl to minimise rotor exit losses [92].

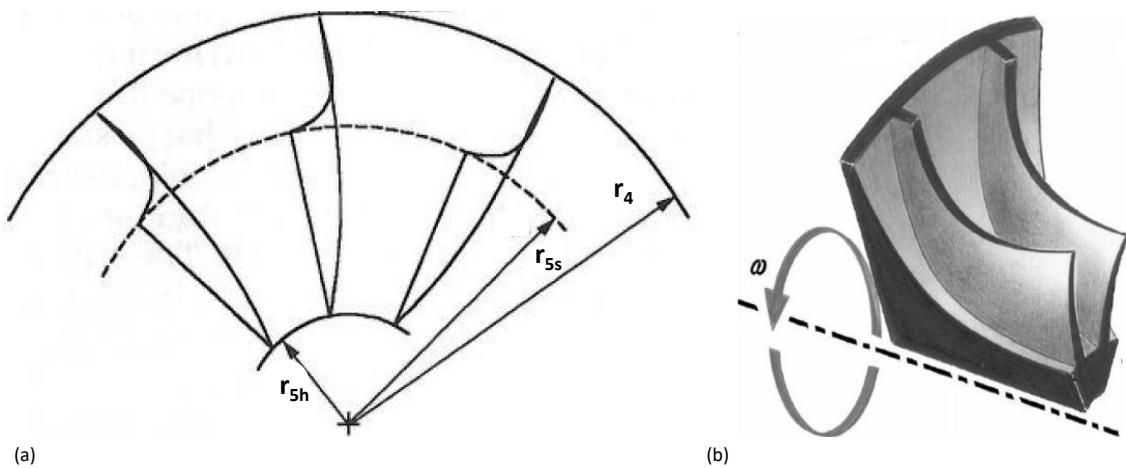


Fig. 2.7 Rotor blade geometry (a) Frontal view [90] (b) 3D section view [93]

2.5.2 High-Fluid Density Designs

A summarised collection of published radial turbine designs for different operating conditions is shown in Table 2.4. The variation of the dimensions and speed with respect to power range coincide with the data of Figure 2.4 where 30 MW is the highest power

output limit for a radial turbine. Table 2.5 lists the methods that researchers used for the design and analysis of sCO₂ radial turbines which range from 1D meanline design, to the higher fidelity 3D CFD modelling reaching in some cases expensive experimental testing. Radial turbines designed for use with other high-density fluids are also reviewed for information of flow behaviour and design methodologies.

Meijboom [87] developed a methodology for designing a supercritical CO₂ radial turbine for small concentrated solar power applications; the author also performed mechanical stress simulations on the turbine. Meijboom noted transonic speed flows at stator outlet and achieved a total-to-static efficiency of 74%. A description of entropy generation and losses through the turbine is made clear through the use of 3D CFD modelling which differs from performance results obtained in the 1D design code. Because of the moderate turbine temperature (550°C) and the low pressure ratio (≈ 2.22), the material of the designed turbine has enough strength to endure the imposed aerodynamic stresses. Recently, a radial turbine design was developed by Unglaube et al. [94] for a small sCO₂ Brayton cycle. The academics performed a cycle analysis study to obtain turbine operating parameters that are required for design. They adopted the design procedure of 1D meanline design followed by 3D CFD flow analysis to reach a total-to-static efficiency of up to 84%. The performed parametric study shows that there needs to be a compromise between expansion ratio and specific speed/ size to obtain good efficiency values. The turbine inlet pressure is half that of the NET Power cycle at a much lower temperature which means that their turbine design is less loaded.

Unlike other works where the turbine design was generated from initial cycle parameters, Ameli et al. [95] performed 1D numerical analysis, using different equations of state, and 3D CFD flow study for a radial turbine developed by Sandia National Laboratories in a sCO₂ test loop. Performance results showed significant differences between the two analysis stages (1D and 3D CFD). The paper lacked indications on the critical flow behaviours within the turbine passages in comparison with experimental findings. Wei [96] constructed a meanline design and performance analysis tool, employing well established methods and loss models, for a supercritical CO₂ radial inflow turbine for both design point and off-design conditions aimed at small closed Brayton cycles. The generated turbine rotor geometry was modelled using 3D CFD for flow characterisation and an observation is made on the effect of rotor incidence angles at different rotational speed on the extent of flow separation at the leading edge.

Researchers including Chen et al. [97] and Cho et al. [98] are some of the few that performed experimental testing for similarly sized sCO₂ turbines (≈ 80 kW power) proving that it is the least used method for flow analysis nowadays especially with the evolution of more sophisticated computational tools. From experiments, Chen et al. [99] suggest that sCO₂ radial turbines can be choked because of the high flow loading but recommends the avoidance of supersonic flows for better performance.

One of the largest scale radial turbines is designed by Persky et al. [100] for a 30 MW power output. An optimisation method was developed to assist in obtaining better performance of turbines using unconventional fluids especially at off-design conditions.

The power scopes seen in Table 2.4 conform with the scaling suggestions of radial turbines shown in Figure 2.4. However, all of the listed turbines are designed for closed loop cycles from medium to low temperature heat sources that do not fit with the semi-closed direct-fired NET Power conditions. Even the highest turbine inlet pressure (design by Strang [101]) is at least 25% lower than the Allam cycle pressure. White et al. [102] compared axial and radial turbine configurations for a 100 kW sCO₂ Rankine cycle and deduced that all design cases have comparable small dimensions which means that the clearance losses are high for such scales.

Useful information on radial turbine design and service can be obtained from literature on similar designs for non-ideal working fluids such as refrigerants or organic Rankine cycle (ORC) fluids. Sauret et al. [83] generated a preliminary design for a R134a radial inflow turbine, and from 3D CFD modelling, they recommend optimising the nozzle blade geometry to lower the supersonic Mach number to reduce associated losses. Alshammari [103] provided a coherent methodology of designing and testing a high pressure ratio (7) ORC radial turbine. Alshammari's results include off-design performance characteristics of the expander as well as descriptive flow analysis from CFD simulations. Findings include the effect of nozzle outlet supersonic flow, tip leakage flows, and rotor separation zones in reducing the total-to-static efficiency to 73.85%. Similarly, Fiaschi et al. [104] quantified relative loss contributions from different mechanisms with the highest being towards rotor secondary flows from sharp blade curvatures. The importance of computing flow parameters at initial design stages for performance analysis is described by Rahbar et al. [105]; this is referred back to in the design methodology of this work. Finally, CFD results of the radial turbine developed by Dong et al. [106] are in good agreement with findings from other researchers regarding the flow behaviour between stator and rotor, the stator wake interaction with the rotor bulk flow of the rotor, and the low momentum flow region at the rotor suction surface.

Table 2.4 Operating conditions of supercritical CO₂ radial turbine designs in literature

Author [Reference]	TIP [bar]	TIT [°C]	TOP [bar]	$\dot{m}$ [kg/s]	$\dot{W}$ [kW]	ω [rpm]	Diameter [mm]
Meijboom [87]	200	550	90	3	300	50,000	100
Unglaube et al.[94]	150	300	85	3	130	100,000	
Ameli et al. [95]	135	537	79	2.58	175	75,000	95
Wei [96]	107	670	78	1.8	84.1	80,000	88
Chen et al. [99]	120	100		0.5	1	3000	
Persky et al. [100]	200	560	91	422.3	30,000	20,000	370
Cho et al. [98]	130	180	81	3.04	82.4	70,000	48.6
Lee [107]	220	259	79	3.5		8,500	607
White et al. [102]	170	650	57	0.65	100	200,000	30
Strang [101]	233	358	98	170	14,720	21,000	340

Table 2.5 Methods for design and analysis of radial turbines in literature

Reference	Fluid	Meanline Design	3D CFD	Experiments
Meijboom [87]	sCO ₂	✓	✓	
Unglaube et al. [94]	sCO ₂	✓	✓	
Ameli et al. [95]	sCO ₂		✓	
Wei [96]	sCO ₂	✓	✓	
Chen et al. [99]	sCO ₂			✓
Persky et al. [100]	sCO ₂	✓		
Cho et al. [98]	sCO ₂	✓	✓	✓
Lee et al. [107]	sCO ₂	✓	✓	
White et al. [102]	sCO ₂	✓		
Strang [101]	sCO ₂	✓	✓	
Sauret et al. [83]	R134A	✓	✓	
Alshammari [103]	ORC	✓	✓	✓
Fiaschi et al. [104]	ORC	✓		
Rahbar et al. [105]	ORC	✓		
Dong et al. [106]	sORC	✓	✓	

2.5.3 Cooling in Radial Turbines

For the operating environment of the NET Power cycle, turbine cooling flows and blade thermal barrier coatings are deployed to reduce the temperatures handled by the turbine material. Even though cooling is used to allow for higher firing temperatures to improve cycle performance, it is at the same time associated with penalties on the overall thermodynamic cycle efficiency and with added system complexity [108]. In fact, cooling not only jeopardises the efficiency of the cycle on a whole by using a considerable amount of compressed flow to cool a hot gas component, but also inflicts losses in the turbine part itself due to the viscous mixing of coolant with the mainstream bulk flow as described by Denton [109]. And while the effectiveness of cooling techniques has been well optimised for axial turbine configurations via the use of different methods and material coatings, cooling in radial turbines is not common practice. NASA [110] and the US Army [111] were among the very few institutes doing research of cooled radial turbines during the last 20 years of the 1900s, after which funding and investigations were abandoned for not finding a suitable application of the technology.

More recently, a group of researchers from the University of Bath have designed and analysed an air-cooled radial turbine using CFD modelling [112] and experimental testing [113]. The concept is for a small turbine with an inlet temperature of 750°C and the cooling was effective in lowering the blade temperature by 117°C compared to an uncooled model [112]. Selective laser melting additive manufacturing proved feasible for the construction of the turbine with intricate cooling passages [113]. Although successful, this study is in its infancy and the temperature is much lower than that of the Allam

cycle. Nonetheless, it paves the way to future work for using cooling at higher operating temperatures that could be implemented in the passages of a designed radial turbine.

The absence of concrete work on cooling for extreme temperatures in radial turbines endorses the elimination of cooling from the cycle adopted by the author of this PhD thesis through lowering the turbine inlet temperature.

Chapter Summary

The robust literature review that is presented supports the main objectives of this work through recognising the research gaps in designing a radial turbine for a mid-scale supercritical CO₂ cycle and assessing its performance using different fidelity models. Firstly, it re-identifies the reasons behind choosing the NET Power cycle — similarly to the choices of Chapter 1 — for study based on the advantages of using oxy-combustion and supercritical CO₂ fluids. The methods used by authors to obtain thermo-physical properties of the working fluid, to perform thermodynamic cycle modelling, and to design a radial turbine are adapted for the methodology of this work. Reported findings on cycle parameters and high-fluid density flow behaviour in radial turbines are used in the results chapters for comparison with outcomes obtained in the PhD work. In addition, suggestions for future work are based on identified gaps and key issues found in literature. This chapter serves to fulfil the first identified research objective of this piece of work (**Task 1. To identify the gaps in knowledge and gather relevant information by conducting a thorough literature review and communicating with experts in the field**).

CHAPTER 3

Methodology

This chapter presents the methodology followed for each part of the work; assumptions and procedures are stated, some of which are based on previous works found in literature. It goes through the approach undertaken to model the selected cycle to obtain the operating conditions of the turbine which are required for design. The chapter covers all the steps carried out in the design procedure from meanline design, to acquire a range of the overall size and performance criteria of the turbine, to preliminary design, where turbine geometries are generated and analysed. The techniques adopted to assess the turbine performance, study the flow behaviour, and understand loss mechanisms are also described; those include 2D throughflow analysis, and using viscous computational fluid modelling. An explanation of the performed stress calculations is also presented. All the relevant tools and pieces of software used in this work are mentioned.

The design procedure of any turbomachine for a specific requirement follows an iterative interdisciplinary path involving thermodynamics, fluid mechanics, aerodynamics, stress and vibration analyses, manufacturing, and testing. This interdependency can be visualised in Figure 3.1, such that firstly the size and shape of a turbomachinery component are designed, then with the results obtained from different analysis methods the design would require modification. Once a satisfactory design is obtained, prototype testing follows which then calls for further examination that might well induce more changes [114].

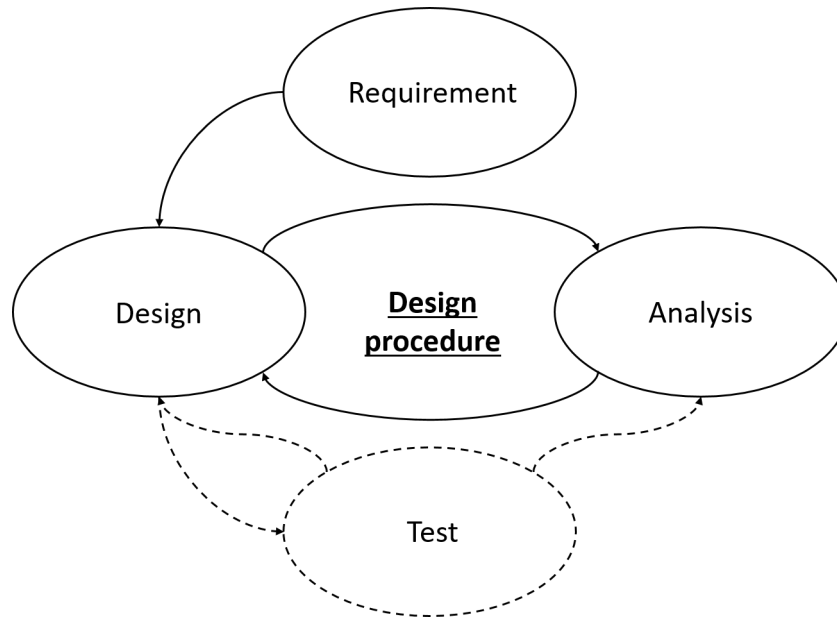


Fig. 3.1 Iterative design procedure [114]

Based on the main aim of this piece of work stated in Section 1.4, the adopted methodology does not cover the whole design procedure described above but follows the routes needed to understand the operating environment of the turbine and focus areas where challenges lie. Those routes are part of the non-dashed ovals of Figure 3.1 and sectioned into different thermodynamic, aerodynamic and mechanical design steps summarised in Table 3.1. The first column on the left itemises the design output from each step of the procedure with the second one giving the respective results section. The third column lists the means used while the right one relates the methodology sections to each design model that satisfy the research objectives stated in Section 1.4.

Table 3.1 Thesis design steps

Methodology	Section	Results	Section
Cycle requirements	3.1	Operating conditions definition	4.1
Machine type definition			
Meanline design without loss consideration	3.2	Overall dimensions and performance	4.2
Component design using 1D methods	3.3	Geometry definition	5.1, 6.1
2D streamline calculations	3.4	Performance analysis	5.2, 6.2
CFD analysis	3.5	Flowfield study	5.3, 6.3
FEA stress analysis	3.6	Structural requirements	6.4

3.1 Cycle Modelling

Task 2. To define an appropriate cycle model based on requirements and acquire the operating environment of the radial turbine through cycle process modelling

The cycle performance parameters are the foundation of turbine design. The operating conditions dictated by the cycle requirements are defined via a process simulation model.

3.1.1 Brief Cycle Description

The cycle considered in this work is a modification of the NET Power cycle introduced in Section 2.3. The changes include a reduced turbine inlet temperature and a lower pressure ratio; the uncooled turbine case is considered as a datum configuration for the following reasons:

- Simpler cycle modelling as an initial case taking into account that the aim of the project (Section 1.4) is not cycle optimization but more focused on turbine design considerations
- One of the benefits of a supercritical CO₂ cycle is small turbomachinery components which implies small blades that require intricate passages if cooling is employed. The design complexities associated with cooling are eliminated for the use of a smaller-scale radial turbine design
- Smaller power output scale so that a radial turbine can still be within acceptable efficiency ranges (power limit shown in Subsection 4.1.3)
- Reasons described in the literature review Subsection 2.5.3 regarding the lack of expertise in radial turbine cooling

The schematic drawing of the modified cycle under study is represented in Figure 3.2. Changes to the original cycle model of Figure 2.2 include the absence of cooling flows in the turbine and the regenerator, plus the use of two flow compressors instead of four. The remaining parts are left almost unchanged unless indicated in the methodology. The description of the components present in the cycle model of Figure 3.2 can be found in Table 3.2.

Table 3.2 Cycle model component description

Component name	Description
NG-CPR	Natural gas compressor
COMB	Combustor
TURB	Turbine
ASU	Air separation unit
HTHE	High temperature heat exchanger
LTHE	Low temperature heat exchanger
COND	Water condensor
SEP	Water separator
CPR-i	Flow compressors (-i)
IC-i	Intercoolers (-i)
STO	CO ₂ separator
STO-CPR	Storage compressor
ST-IC	Storage intercooler
O ₂ -S	Oxidant separator
OXI-CPR	Oxidant compressor

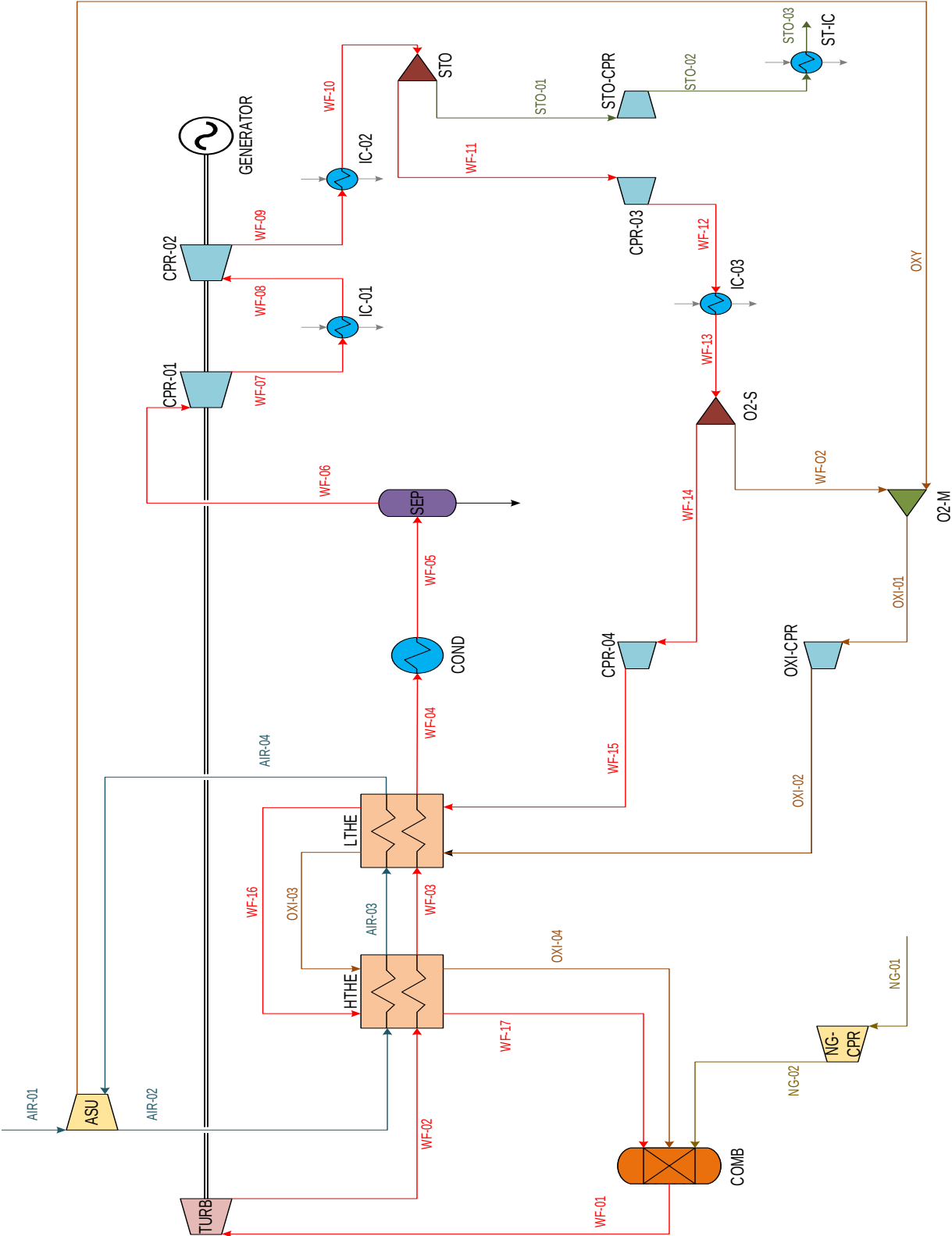


Fig. 3.2 Schematic drawing of modified cycle

In collaboration with a research group in Politecnico Di Milano, led by Prof. Emanuele Martelli, the licensed software Aspen Plus V9 [115] is used for cycle modelling to obtain the necessary data for turbine design which include working fluid ratios, power output, flow rates as well as temperatures and pressures. Cycle modelling and analysis is performed before detailed geometrical information or structural aspects of components are defined.

The initial steps in setting up a model using Aspen Plus take place in the properties environment where the following tasks are addressed:

1. Specifying the property and calculation methods used to obtain thermodynamic properties
2. Defining all the chemical components used in the simulation

The subsequent step consists of defining the main process flowsheet within the simulation environment of Aspen Plus as:

3. Choosing the appropriate component blocks and designating specifications (efficiency, operating conditions...) with connections to defined material, heat and/ or work streams
4. Assigning simulation calculation loops and regulators

3.1.2 Flow Definition

The parameters required for setting up the chemical simulation environment in Aspen Plus are introduced.

Equation of State

In order for Aspen Plus to estimate fluid properties, such as the effect of pressure on thermodynamic parameters, an equation of state, suitable for the provided criteria (real gases), needs to be identified [116]. As discussed in Subsection 2.4.2, the Peng-Robinson EOS — developed by Ding-Yu Peng and Donald B. Robinson — is chosen as the appropriate property method for the simulation due to other researchers [73, 117] showing its suitability for the temperature and pressure ranges applicable to the NET Power cycle.

Fluid Components

Natural gas (NG), with a lower heating value (LHV) of 46.502 MJ/kg [73], is the fuel supplied at 15°C and 70 bar to the cycle; the components that constitute NG are listed in Table 3.3.

Table 3.3 Natural gas composition [34]

Components	vol%
Methane	89.0
Ethane	7.0
Propane	1.0
Butane	0.1
Pentane	0.01
Carbon Dioxide	2.0
Nitrogen	0.89
Total	100

Besides the components that make up natural gas, other compounds present in the cycle are specified. Table 3.4 sums up all the chemical compounds selected; these include the isomers of the natural gas constituents and the gas composition of air.

Table 3.4 Chemical compounds of cycle model

Component	Chemical Formula
Methane	CH ₄
Ethane	C ₂ H ₆
Propane	C ₃ H ₈
N-Butane	C ₄ H ₁₀ -1
IsoButane	C ₄ H ₁₀ -2
N-Pentane	C ₅ H ₁₂ -1
2-Methyl-Butane (IsoPentane)	C ₅ H ₁₂ -2
Argon	Ar
Carbon Dioxide	CO ₂
Nitrogen	N ₂
Water	H ₂ O
Oxygen	O ₂

3.1.3 Component Models and Connections

The mass and energy balance equations are solved in Aspen Plus using the sequential modular simulator solver. The sequential flowsheet solver allows each unit operation block to be executed in succession; the computed output streams of every block makes the inlet/ feed stream of the following one [115]. The unit operation blocks assigned for the cycle components are summarised below.

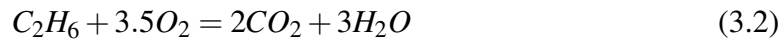
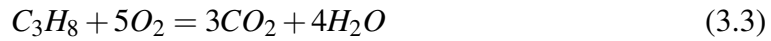
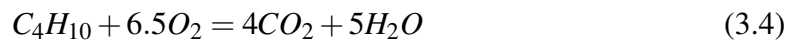
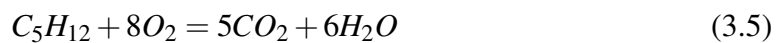
Compressors

The compressors of the cycle, which include the fuel compressor (NG-CPR), the ASU, the flow compressors (CPR-01, CPR-02, CPR-03, CPR-04), the storage compressor (STO-CPR) and the oxidant pressure changer (OXI-CPR) are specified as the readily available polytropic compressor block.

The ASU has particular specifications for supplying pressurized 99.5% pure oxygen at 120 bar and 15°C plus some heat (due to the compression of available 15°C atmospheric air to 7.5 bar) to the hot side of the heat exchanger [73]. Based on Scaccabarozzi, the power consumption of the ASU in this model is assumed to be 1,391 kJ/kg_{O₂}.

Combustor

The NET Power oxy-fuel combustor (COMB) is modelled as a *RStoic* reactor because the stoichiometric conversion of the reactions are defined and the reaction kinetics are unknown. The complete reactions occurring between natural gas components and pure oxygen are as follows:



The oxygen mole fraction in the combustor oxidant stream (OXI-01) is set to be 13.34% so that lean combustion takes places with an oxygen excess of 3% in comparison to stoichiometric conditions [73].

Turbine

Unlike the cooled turbine of the NET Power cycle modelled by Scaccabarozzi et al. [73], simulated using an improved continuous expansion model, this cycle assumes that an uncooled expander where the existing Aspen Plus turbine model (R-EX) is sufficient. A single stage isentropic turbine is simulated with the absence of the coolant stream by solving the main equation [115]:

$$\Delta h = \eta_{isen} \int_{P_{in}}^{P_{out}} V dP \quad (3.6)$$

where η_{isen} is assumed constant and represents isentropic efficiency, Δh is the change in enthalpy between the inlet and outlet states of the turbine, V is the volume and P is the pressure [116].

Heat Exchanger

For the same reasoning explained in Scaccabarozzi et al's work [73], one multi-flow heat exchanger is not suitable for the regenerator modelling because of its inability alone to capture the effects of the variation in the fluid specific heats and the condensation of water for conditions below the dew point of the turbine flue gases — both phenomena allowing pinch-points to occur. To be able to represent pinch-points and handle multiple flows with different exit temperatures, the regenerator of the uncooled cycle is modelled as two multi-flow heat exchangers *MHEATX*, which consider the high temperature (HTHE) and low temperature sides (LTHE). As shown in Figure 3.3, the flue gases from the turbine entering HTHE cool down to reach their dew point (vapour fraction set to 1) before entering LTHE. The temperatures of the hot flows (exhaust gases and air) exiting LTHE are calculated from energy balances of the heat exchangers; whereas the temperatures of the cold flows (oxidant and recycled working fluid) are regulated to acquire a minimum temperature difference of 5°C in LTHE so that to ensure optimal usage of the available heat present in the turbine flue gases entering the regenerator is transferred to the recycle and oxidant streams that are directed to the combustor.

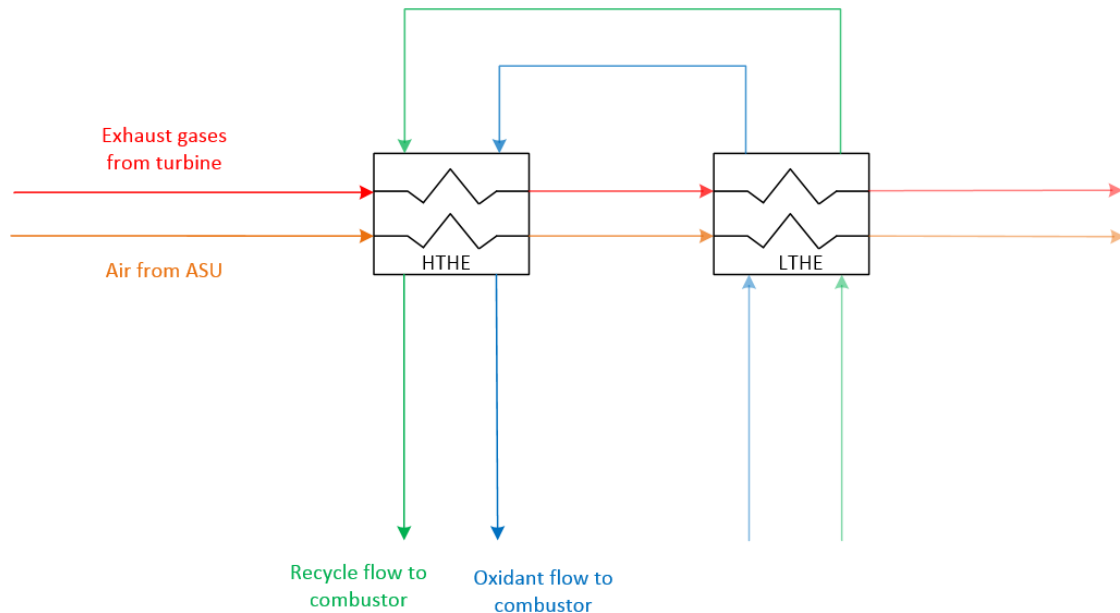


Fig. 3.3 Aspen model of regenerator

Quantitative Assumptions

Assumptions are required for the cycle model, most of them are adapted from Scacabarozzi et al. [73]. Table 3.5 lists the efficiencies applicable to each turbomachinery component within the cycle model. A higher mechanical efficiency of 99.6% is attributed to the main non-auxiliary cycle turbomachinery. A relatively low conservative turbine isentropic efficiency is started with to simulate a radial turbine that is still under study and development to suit the uncooled cycle requirements; an elaboration on turbine isentropic efficiency values is given in the results of Section 4.1.

Table 3.5 Turbomachinery efficiency assumptions

Efficiency	Value [%]
Polytropic efficiency of NG compressor & ASU	85
Polytropic efficiency of vapor phase compressors	90
Polytropic efficiency of dense phase compressors	85
Mechanical efficiency of auxiliary compressors	98
Mechanical efficiency of flow compressors and turbine	99.6
Generator efficiency	99

In addition to efficiency assumptions, assumed pressure drops are allocated for some components (mainly heat exchangers) and are found in Table 3.6.

Table 3.6 Pressure drops in components

Component	Pressure drop [bar]
High pressure flows in regenerator	2
Low pressure flows in regenerator	0.75
Combustor	3
Intercoolers	0.5 – 1

Stream Specifications

Table 3.7 shows the identified and unchangeable stream specifications for the cooling water, air, NG and O₂ from ASU. Other unmentioned initial guesses of stream temperatures, flow rates or pressures are also used as input for general convergence of the calculation.

Table 3.7 Stream specifications

Stream	T [°C]	P [bar]	Ar [%]	CO ₂ [%]	H ₂ O [%]	N ₂ [%]	O ₂ [%]
Cooling water (CW-i)	15	2	-	-	100	-	-
Air (AIR-01)	15	1.01	0.92	0.04	0.97	77.32	20.75
Natural gas (NG-01)	15	70	NG mole fractions				
O ₂ from ASU (OXY)	15	120	0.3	-	-	0.2	99.5

3.1.4 Simulation Definition

Design Specifications

Some design specifications are employed to fine tune the stream and block functions as per the studied cycle requirements; these include:

- Combustor duty: The mass flow of the recycle stream to the combustor (WF-17) is varied so that the reactor duty (i.e. net heat balance) is equal to 0 kW.
- Cooling water: The mass flow of the inlet cooling water to the condenser (COND) and inter-coolers (IC-01, IC-02, IC-03, ST-IC) is changed so that the outlet water temperature is 25°C.
- Minimum temperature difference: The hot temperature of the inlet working fluid (WF-03) to the low temperature side of the heat exchanger (LTHE) is iterated so that minimum temperature difference at that side is 5°C.
- Mole fraction of oxygen needed for combustion: the flow fraction of the oxygen splitter (O2-S) is varied for the oxygen stripped from the working fluid (WF-O2) so that the mole fraction of oxygen in the oxidant stream (OXI-01) before compression is 0.1334.
- Pressure ratio of first flow compressor: The pressure ratio of the first flow compressor (CPR-01) is amended to get equal work across the first two flow compressors.

Calculator Blocks

In addition to the defined design specifications, several calculation options need embedding to ensure convergence in accordance with the cycle design basis, these include the following:

- Mole flow rate of air:

$$\dot{n}_{air} = \frac{\dot{n}_{oxy}}{0.2075} \quad (3.7)$$

where $\dot{n}_{air}$ is the mole flow rate of the air needed to be compressed in the ASU to obtain the required oxygen content and $\dot{n}_{oxy}$ is the oxygen mole flow rate at the exit of the ASU.

- Mole flow rate of oxygen: The mole flow rate of oxygen – $\dot{n}_{oxy}$ – in the stream from the ASU is equal to the difference between the flow rate of oxygen required to combust the natural gas – $\dot{n}_{oxyNG}$ – and flow rate of oxygen in the working fluid (WF-13) – $\dot{n}_{oxyWF}$ – divided by the oxygen purity:

$$\dot{n}_{oxy} = \frac{\dot{n}_{oxyNG} - \dot{n}_{oxyWF}}{0.995} \quad (3.8)$$

The mole flow of oxygen required to burn the fuel (which is set to be 3% in excess) is based on the number of moles of oxygen for each component of natural gas as shown in Eq. 3.1-3.5.

$$\dot{n}_{oxyNG} = 1.03(2\dot{n}_{CH_4} + 3.5\dot{n}_{C_2H_6} + 5\dot{n}_{C_3H_8} + 6.5\dot{n}_{C_4H_{10-i}} + 8\dot{n}_{C_5H_{12-i}}) \quad (3.9)$$

- Storage mass flow fraction to keep the mass balance: The flow fraction of the mainly CO₂ storage flow in the storage splitter (STO) is equal to the summation of the mass flows of natural gas and oxygen supplied from the ASU minus the mass flow of water separated from the system.
- Temperature control in regenerator: Two other calculator blocks are added to maintain the temperature of the working fluid to be equal to that of the hot air within the regenerator.

3.2 Meanline Design

Task 3. To obtain a range of overall turbine dimensions and performance criteria by using validated design methods and recommendations found in literature

The turbine operating conditions obtained from the cycle model are used in this intermediate step to give an insight on the possible performance range and geometric dimensions of the radial expander. A meanline design tool is developed using MATLAB by implementing established turbomachinery equations.

3.2.1 Working Fluid

The programme NIST REFPROP has been used widely throughout literature for the attainment of thermodynamic and transport properties in sCO₂ turbomachinery design tools [96, 102, 118, 119, 120] (in reference to Subsection 2.4.2). It provides thermophysical properties of pure and compound fluids over a broad range of states for liquids, gases and supercritical phases based on validated data and calculation methods. Therefore, the free sample version mini REFPROP is incorporated within the meanline design MATLAB code as an expedient way to obtain fluid properties instead of using the Peng-Robinson equations which require further definition.

3.2.2 Meanline Design Procedure

The initial design procedure is carried out using a meanline approach, which assumes one-dimensional passage conditions at mean radius of the turbomachine. This method is used to determine the outline geometry of the turbine, given specific inputs, and to investigate the dependency of overall performance (size, efficiency, power) on key design

parameters and assumptions such as specific speed and operating conditions [121]. Within the MATLAB tool, the estimate of the turbomachinery efficiency is obtained without the integration of loss models as these get incorporated in the later stages of design. The main goal of this step is to obtain a range of turbine size parameters for use in the preliminary design tool of AxSTREAM in Section 3.3.

An aim of a meanline design is to obtain crucial geometric parameters at each stage section; these are illustrated in the nomenclature of Figure 3.4 which is adapted from Unglaube et al. [94] and Wei [96]. The parameters include inlet/ outlet radii and blade heights of each component, axial lengths, plus other data required to define the cross-section of the turbine. The mean rotor outlet radius (r_5), which gives an average of the blade span, is defined as the arithmetic average of the shroud (r_{5s}) and hub (r_{5h}) radii [96].

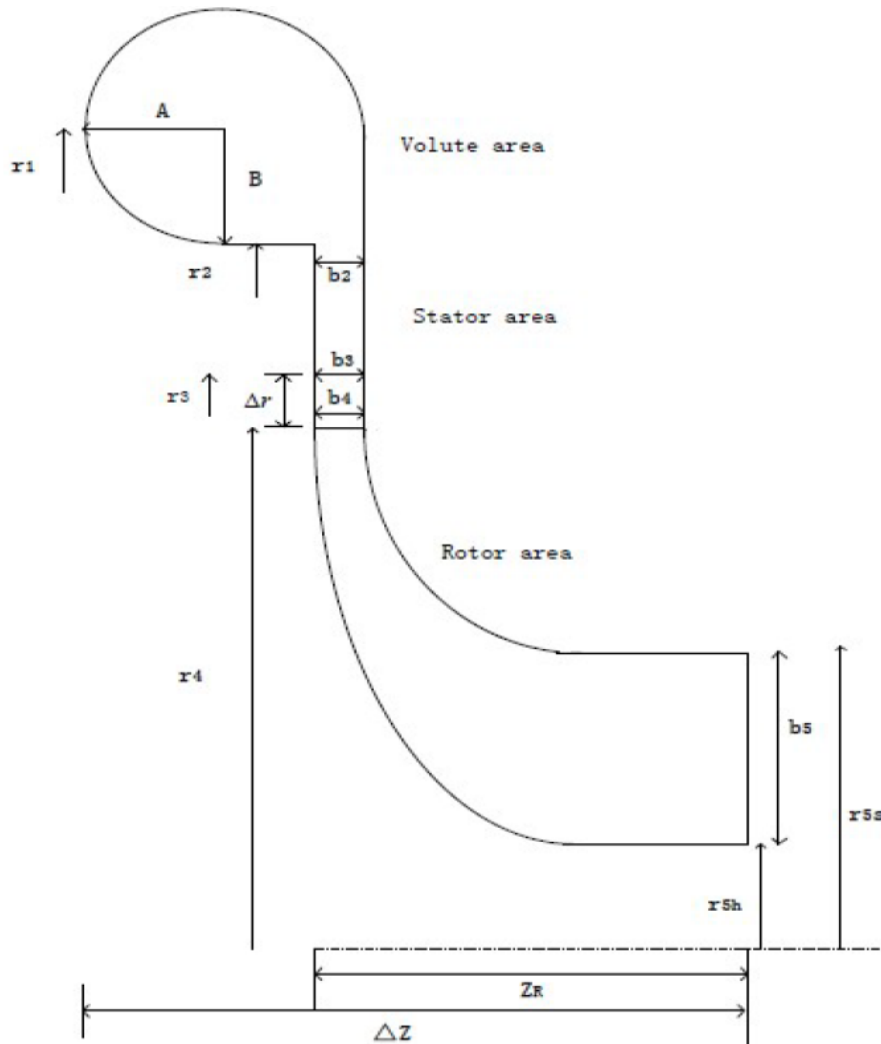


Fig. 3.4 Radial inflow turbine stage nomenclature [96]

Through the meanline analysis, an insight is also given on the velocity triangles of the

designed turbine. Velocity triangles give a visual vector representation of the kinematic principle that the absolute velocity in the stationary frame (C) is the sum of the relative velocity in the rotating frame (W) and the linear blade velocity (U) [93]. These values can enable the calculation of work extraction, radii and/ or rotational speed depending on the designated initial parameters. An example of rotor velocity diagrams of a radial inflow turbine is shown in Figure 3.5 which also illustrates the nomenclature used for the absolute and relative flow angles at rotor inlet and outlet [96].

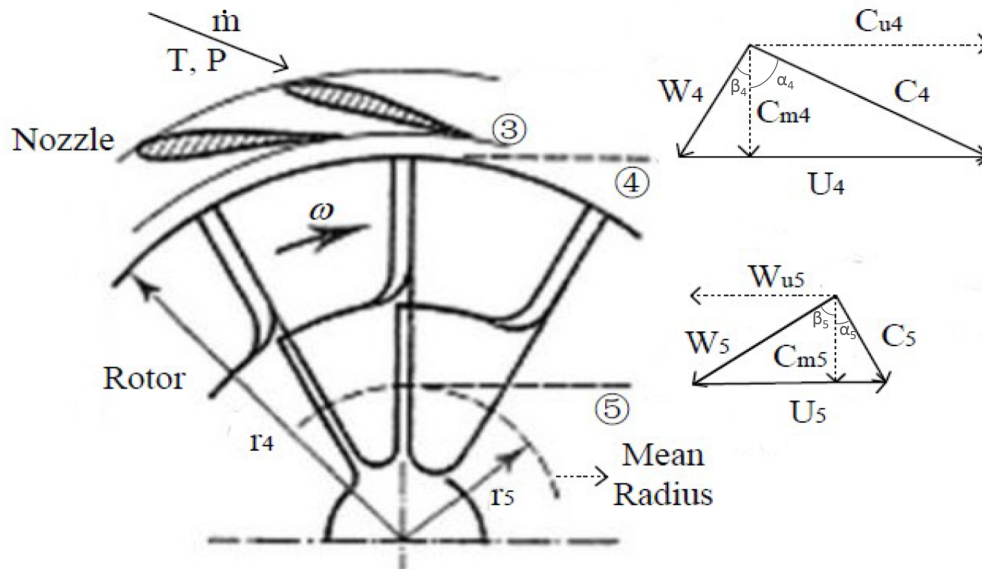


Fig. 3.5 Example of radial turbine mean radius and velocity triangles definition [96]

At rotor outlet (station 5), the aim is to minimise velocity exit losses. The flow needs to exit the rotor in the axial direction, this is achieved by seeking a zero exit swirl (i.e. $\alpha_5 = 0^\circ$), and by maintaining the direction of the exit blade angle (β_5) to be opposite to that of the shaft rotation [96]. In general, negative values are given to angles that are opposite to the direction of rotation.

There exists several methods of obtaining meanline design data of a radial turbine including propositions from [79, 90, 122]. A method introduced by Dixon [122] starts with initial estimates of non-dimensional parameters, such as flow coefficient (ϕ) and stage loading coefficient (ψ), to obtain the turbine nominal design followed by optimum efficiency design. Similarly, Aungier [90] developed an approach that can encompass the full operating range of a radial turbine. The meanline aerodynamic design procedure of this work follows that introduced by Aungier [90] and is explained in the following pages.

The turbine operating conditions are obtained from the cycle model implemented in Aspen Plus; those include the mass flow rate ($\dot{m}$), total inlet temperature (T_{01}) and pressure (P_{01}), thermal power output ($\dot{W}$), and total-to-total pressure ratio (Π_{tt}). These defined parameters are the input to the design code. The turbine inlet total conditions enable the

calculation of the inlet properties which are total enthalpy (h_{01}), density (ρ_{01}), and entropy (s_1).

The total enthalpy change within the turbine can be calculated by using the steady flow energy equation (Eq. 3.10) assuming no potential, kinetic or internal energy change and no heat transfer. This in turn enables the calculation of the total enthalpy at rotor exit (h_{05}) along with the total temperature (T_{05}), density (ρ_{05}), and entropy (s_5) since the total pressure (P_{05}) is derived from the total-to-total pressure ratio.

$$\Delta h_0 = \frac{\dot{W}}{\dot{m}} = h_{01} - h_{05} \quad (3.10)$$

The design strategy introduced by Aungier [90] is based on the definition of specific speed (n_s) and its correlation with total-to-static blade to gas velocity ratio (v_s) and total-to-static efficiency (η_{ts}). The definition of the dimensionless specific speed is given as:

$$n_s = \frac{\omega \sqrt{\dot{Q}_5}}{(\Delta h_{id})^{0.75}} \quad (3.11)$$

where ω is the rotational speed in rad/s and the Δh_{id} is the isentropic enthalpy change through expansion from stage inlet total conditions to rotor exit static conditions:

$$\Delta h_{id} = \frac{\Delta h_0}{\eta_{ts}} \quad (3.12)$$

and the volumetric flow rate ($\dot{Q}_5$) is calculated using:

$$\dot{Q}_5 = \frac{\dot{m}}{\rho_{05}} \quad (3.13)$$

The initial choice for n_s is usually based on generalized performance criteria available in literature such as that suggested by Figure 3.6 [90] which correlates expected values of the turbine efficiency and velocity ratio based on the selection of specific speed. The values from the graph have been validated using experimental data from two turbines [90]. As seen, the optimal range of specific speed, to obtain a relatively high total-to-static efficiency, usually lies between 0.45 – 0.75; however, this boundary is not fixed and can be extended beyond the limits to adapt to specific constraints in some parameters such as size, rotational speed, Mach numbers, and mechanical integrity.

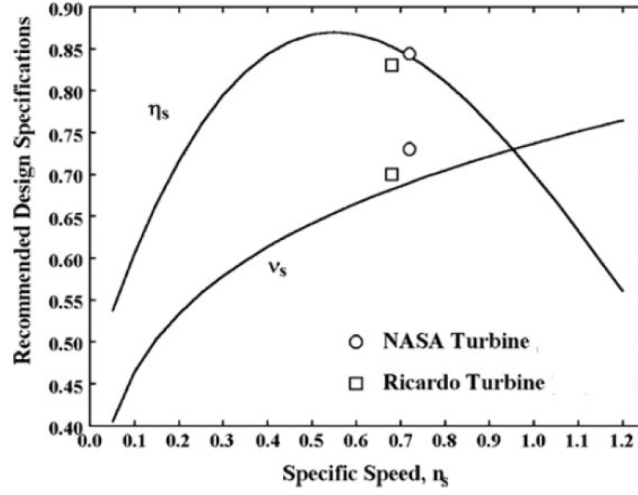


Fig. 3.6 Radial turbine stage performance chart [90]

The graphical correlation between the three basic design parameters can be given in the form of analytical equations in terms of specific speed [90]:

$$v_s = 0.737n_s^{0.2} \quad (3.14)$$

$$\eta_{ts} = 0.87 - 1.07(n_s - 0.55)^2 - 0.5(n_s - 0.55)^3 \quad (3.15)$$

The total-to-static efficiency does not get updated and is assumed constant throughout the meanline design procedure. Once the specific speed is initially guessed, the rotational speed can be calculated from Eq. 3.11.

3.2.3 Rotor Design

The turbine sizing procedure commences with the rotor design.

Rotor Inlet

The rotor tip sizing procedure begins after obtaining the stage design parameters (n_s , v_s , η_{ts}). The discharge spouting velocity (C_{0s}) which is associated to the kinetic energy corresponding to the enthalpy drop from total inlet to static exhaust conditions [122] is given by:

$$C_{0s} = \sqrt{(2\Delta h_{id})} \quad (3.16)$$

The rotor tip blade speed (U_4) is then calculated as:

$$U_4 = v_s C_{0s} \quad (3.17)$$

and the rotor inlet tip radius can be obtained from:

$$r_4 = \frac{U_4}{\omega} \quad (3.18)$$

To minimize high kinetic energy rotor exit losses, the rotor is designed with zero exit swirl (i.e. $C_{U5} = 0$) then the rotor inlet absolute tangential velocity (C_{U4}) is given by:

$$C_{U4} = \frac{U_4 \eta_{ts}}{2v_s^2} \quad (3.19)$$

The rotor inlet absolute flow angle (measured from the axial direction) is estimated using [90]:

$$\alpha_{4ax} = 10.8 + 14.2n_s^2 \quad (3.20)$$

thus α_4 (from tangential direction) as defined by Figure 3.5 is $90^\circ - \alpha_{4ax}$.

Now that the absolute velocity and tip speed are calculated, the rest of the rotor inlet velocity triangle can be derived starting with the rotor inlet absolute meridional velocity (C_{M4}):

$$C_{M4} = \frac{C_{U4}}{\tan \alpha_4} \quad (3.21)$$

and the rotor inlet absolute velocity (C_4) is:

$$C_4 = \sqrt{(C_{U4}^2 + C_{M4}^2)} \quad (3.22)$$

The calculation of the relative velocity components is carried out while having $W_{M4} = C_{M4}$, then the rotor inlet relative tangential velocity (W_{U4}) is

$$W_{U4} = C_{U4} - U_4 \quad (3.23)$$

thus the rotor inlet relative velocity (W_4) is

$$W_4 = \sqrt{(W_{M4}^2 + W_{U4}^2)} \quad (3.24)$$

the sign of the rotor inlet relative flow angle (β_4) is defined based on the direction of the relative velocity, which is usually opposite to the rotational speed to obtain optimum rotor incidence. Also, it is calculated as:

$$\beta_4 = \arctan \left(\frac{W_{U4}}{W_{M4}} \right) \quad (3.25)$$

The rotor inlet and outlet blade thicknesses (t_4 and t_5 respectively) and the exit hub radius are defined using suitable default values from Aungier [90]:

$$t_4 = 0.04r_4 \quad (3.26a)$$

$$t_5 = 0.02r_4 \quad (3.26b)$$

$$r_{5h} = 0.185r_4 \quad (3.26c)$$

The total pressure at rotor inlet is estimated based on the isentropic enthalpy loss indicated by the total-to-static efficiency and is given by

$$P_{04} = P_{01} - \frac{\rho_{01} \Delta h_{id} (1 - \eta_{ts})}{4} \quad (3.27)$$

the rest of the total conditions at rotor inlet (T_{04} , ρ_{04} , s_4) can be obtained by the equation of state knowing that $h_{04} = h_{01}$ (since the stagnation enthalpy across the volute and nozzle blades is constant). Moreover, the static enthalpy at rotor inlet (h_4) is obtained from:

$$h_4 = h_{04} - \frac{1}{2} C_4^2 \quad (3.28)$$

therefore, from the static enthalpy and the entropy, the remaining static conditions at rotor inlet (T_4 , ρ_4 , P_4) can be acquired.

Finally, the rotor inlet blade passage width (b_4) is calculated from:

$$b_4 = \frac{\dot{m}}{2\pi r_4 \rho_4 C_{m4}} \quad (3.29)$$

Rotor Outlet

The design of the rotor outlet comes next. Aungier [90] noted that the ratio of the absolute meridional velocities $\left(\frac{C_{M5}}{C_{M4}}\right)$ is a function of the ratio of the inlet passage width to the impeller inlet radius as follows:

$$\frac{C_{M5}}{C_{M4}} = 1 + 5 \left(\frac{b_4}{r_4}\right)^2 \quad (3.30)$$

with the rotor exit absolute velocity known from Eq.3.30 since the aim is for $C_{U5} = 0$ leading to $C_5 = C_{M5} = W_{M5}$. From that, the static enthalpy at rotor outlet (h_5) can be computed similarly to Eq. 3.28, along with all the static properties (T_5 , ρ_5 , P_5) using the equation of state.

The rotor exhaust area (A_5) is calculated from the continuity equation

$$A_5 = \frac{\dot{m}}{\rho_5 C_{M5}} \quad (3.31)$$

using Eq. 3.32, the rotor exit shroud radius can be computed:

$$A_5 = \pi(r_{5s}^2 - r_{5h}^2) \quad (3.32)$$

The rotor exit mean radius as defined by Figure 3.5 is then the average of the hub and shroud exit radii, while the rotor exit passage width (b_5) is the difference between the tip and shroud radii at outlet.

Having the rotor exit mean radius and the rotational speed, the blade tip speed at rotor outlet (U_5) is then:

$$U_5 = r_5 \omega \quad (3.33)$$

Having the relative tangential velocity equal in magnitude but opposite in direction to the blade tip speed ($W_{U5} = -U_5$), the relative tangential velocity at rotor outlet (W_5) with its respective relative flow angle (β_5) are obtained likewise to Eq. 3.24 – 3.25. This concludes the process of defining velocity triangles at rotor exit which enables the computation of the relative and absolute Mach numbers.

The rotor axial length (Z_R) and the rotor number of blades (N_R — rounded to the nearest whole integer) are estimated using the methods of Aungier [90] and Rohlik [123]:

$$Z_R = 1.5(r_{5s} - r_{5h}) \quad (3.34)$$

$$N_R = 12 + 0.03(33 - \alpha_{4ax})^2 \quad (3.35)$$

In addition, the rotor exit mean blade pitch (q_5) and the mean throat width (o_5) are determined using:

$$q_5 = \frac{2\pi r_5}{N_R} \quad (3.36)$$

$$o_5 = \frac{q_5 C_{m5}}{W_5} \quad (3.37)$$

3.2.4 Nozzle Design

The design procedure for the nozzle row sizing follows next.

Nozzle Outlet

Assuming straight vanes and considering the flow to be subsonic, the nozzle inlet flow angle is considered equal to that at nozzle outlet/ rotor inlet (i.e. $\alpha_2 = \alpha_3 = \alpha_4$) [96]; the passage width at nozzle entry and exit are taken equal to rotor inlet passage width (i.e. $b_2 = b_3 = b_4$).

The radial annulus gap between the nozzle vane outlet and the rotor blade inlet (Δr) is calculated according to Watanabe et al. [124]:

$$\Delta r = 2b_4 \cos \alpha_4 \quad (3.38)$$

Thereafter, the nozzle outlet radius (r_3) with the respective absolute velocity components are given as:

$$r_3 = r_4 + \Delta r \quad (3.39)$$

$$C_{U3} = \frac{r_4}{r_3} C_{U4} \quad (3.40)$$

$$C_{M3} = \frac{C_{U3}}{\tan \alpha_3} \quad (3.41)$$

$$C_3 = \sqrt{C_{U3}^2 + C_{M3}^2} \quad (3.42)$$

Relative velocity components cannot be defined for the stationary nozzle blade row. The stagnation enthalpy is constant across both the volute and nozzle row ($h_{01} = h_{02} = h_{03}$). The static conditions (T_3, ρ_3, P_3) can be calculated by obtaining the static enthalpy at nozzle exit from C_3 and assuming constant entropy in the annulus gap ($s_3 = s_4$); this also enables the calculation of total properties ($T_{03}, \rho_{03}, P_{03}$).

Nozzle Inlet

Due to the unknown geometry at nozzle inlet, an initial approximation of the ratio of nozzle inlet to outlet radius is assumed. According to Aungier [90] the recommended value is between 1.1 and 1.7 where an initial guess of $\frac{r_3}{r_2} = 1.2$ is used to match Wei's [96] procedure.

$$C_{M2} \approx \frac{r_3}{r_2} C_{M3} \quad (3.43)$$

the rest of the absolute velocity components are obtained in the fashion of Eq. 3.40 and Eq. 3.42.

In order to refine the approximation that was suggested at the beginning of the nozzle inlet sizing, an iterative process using the new static enthalpy (h_2) from the estimated absolute velocity (C_2) is used to obtain the static conditions (T_2, ρ_2, P_2) while assuming $s_2 = s_1$, and then the stagnation conditions ($T_{02}, \rho_{02}, P_{02}$) can be obtained as well. The reiterated meridional velocity at nozzle inlet is:

$$C_{M2} = \frac{\dot{m}}{2\pi r_2 \rho_2 b_2} \quad (3.44)$$

3.2.5 Volute Design

An external volute type, where the whole component lies completely above the outlet radius of the scroll, as shown in Figure 3.7 is assumed in this case [96]. A_{SA} and R_{SA} are the ellipse axial and radial semi-axis respectively.

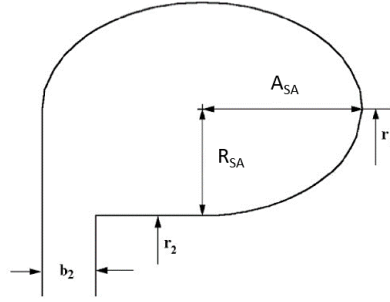


Fig. 3.7 Sketch of external elliptical volute [96]

Initially assuming that the volute inlet radius (r_1) is equal to the outlet radius (r_2), the inlet velocity of the volute is obtained from conservation of angular momentum as:

$$C_1 = \frac{r_2}{r_1} C_{U2} \quad (3.45)$$

from C_1 the static enthalpy at inlet (h_1) and the volute inlet passage area (A_1) are calculated from in the same way as Eq. 3.28 – 3.31.

The static conditions at inlet can be computed based on the static enthalpy and entropy (s_1) knowing the total conditions at turbine entry and the area of the volute along with the inlet radius are given as:

$$A_1 = \left(\frac{3}{4}\pi + 1 \right) A_{SA} B_{SA} \quad (3.46)$$

$$r_1 = r_2 + R_{SA} \quad (3.47)$$

For external volutes, the aspect ratio $\left(\frac{A}{B} \right)$ is generally equal to unity [96], so the values of the ellipse semi-axes can be obtained and new estimations of the inlet radius are calculated accordingly.

Finally, when all the meanline design data are obtained, the maximum radius (r_{max}) and axial length of the turbine stage (ΔZ) are expressed as:

$$r_{max} = r_1 + R_{SA} \quad (3.48)$$

$$\Delta Z = Z_R - b_3 + 2A_{SA} \quad (3.49)$$

3.2.6 Stage Performance Evaluation

After obtaining the stage design criteria, some non-dimensional performance parameters can be calculated to assess the aerodynamic design.

The stage loading or work coefficient (ψ) is a measure of the energy exchange or total stage enthalpy drop for a given blade speed (i.e. design radius); a high stage loading usually implies higher turning and a modest increase in relative Mach number which means there is more work extraction per stage allowing fewer stages (1 in this case of a radial turbine). Since the target design is zero exit swirl, then the loading coefficient depends only on rotor inlet conditions ($\Delta h_0 = U_4 C_{U4} - U_5 C_{U5} = U_4 C_{U4}$). Stage loading is expressed as:

$$\psi = \frac{\Delta h_0}{U_4^2} = \frac{C_{U4}}{U_4} \quad (3.50)$$

The flow coefficient (ϕ) is another parameter that relates to the shape of the velocity triangles; low values lead to large exit angles and increased radii. The flow coefficient is mathematically defined as:

$$\phi = \frac{C_{M5}}{U_4} \quad (3.51)$$

The stage reaction (R) is the a representation of the overall pressure split between the rotor and the nozzle (stator) and is defined as the ratio of static enthalpy drop across the rotor to the total enthalpy change across the stage:

$$R = \frac{(h_4 - h_5)}{(h_{01} - h_{05})} \quad (3.52)$$

Feasibility Limits

A few suggested constraints on geometry and flow are checked to try and evaluate the quality of the turbine design against other validated working designs. The rotor blade tip speed is limited to 450 m/s at any radius for material constraints [125]. The limits are based on best practice data found in literature; while trying to meet all the criteria, this step is not considered extremely crucial and values around the set limits are accepted to proceed with to the next design step as these will be better refined. Some of these constraints used in the meanline design procedure are listed in Table 3.8.

Table 3.8 Radial turbine design constraints [96, 102]

Limiting factors
$Z_R \geq 1.5b_4$
$0.2 \leq \frac{C_{m5}}{U_4} \leq 0.5$
$\frac{r_{5s}}{r_4} \leq 0.85$
$1 \leq \frac{C_{M5}}{C_{M4}} \leq 1.5$
$0.25 \leq R \leq 0.65$
$\alpha_4 \leq 82.5^\circ$

The proceeding methodology Sections 3.3 – 3.6 fall under the umbrella of the integrated turbomachinery design tool AXSTREAMTM licensed by SoftInWay. The flowchart of the AxSTREAM turbomachinery design tool is as shown in Figure 3.8.

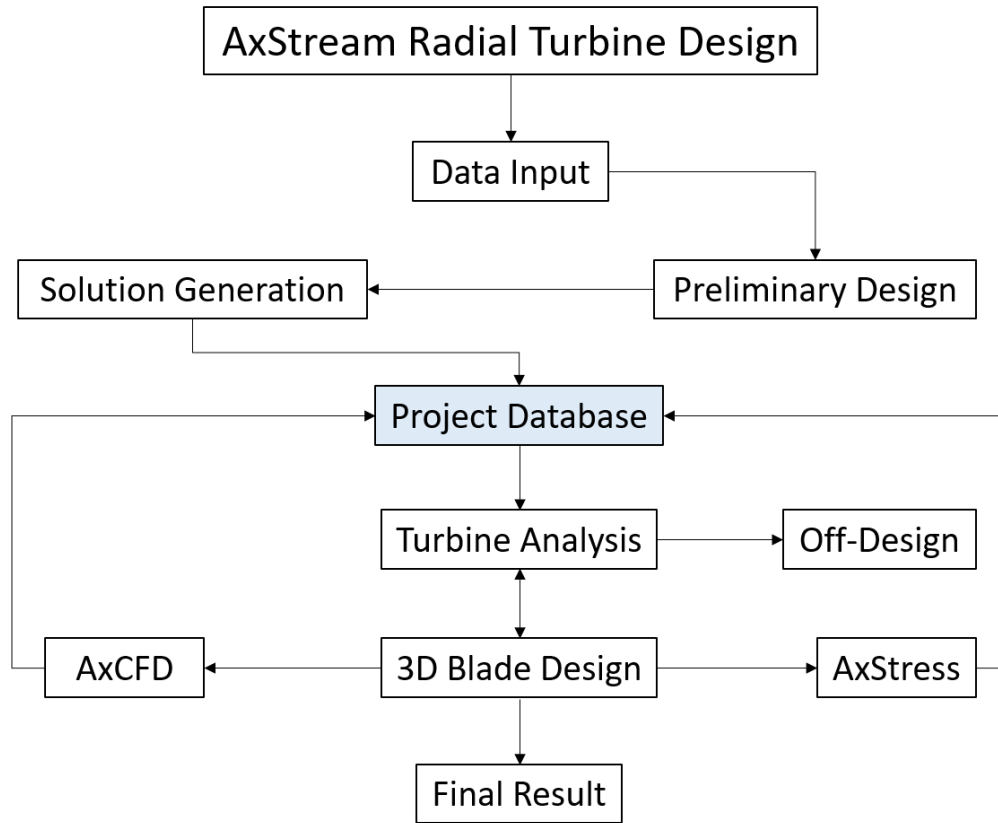


Fig. 3.8 AxSTREAM design flowchart

3.3 Preliminary Turbine Design

Task 4. To gather a selection of possible preliminary turbine geometries via systematic investigation of a number of potential turbine designs with specific constraints

The preliminary design (PD) tool from AxSTREAM is a fast and simple computation to generate several possible turbine flow path designs for a set of given boundary conditions (pressure ratio, mass flow rate, range of rotational speed, flow coefficient) taking into account specified geometric constraints. The model follows a similar procedure to the meanline preliminary design process explained in Section 3.2, relying on the assignment of stage pressure drop and degree of reaction as independent variables of energy, continuity, state and process one-dimensional, steady, equilibrium and adiabatic motion equations. Firstly, velocity coefficients are refined to meet the criteria of the selected empirical

design model, then losses are computed after obtaining a possible flow path, followed by the re-iteration of cascade angles for supersonic flows, and finally the arrangement of geometry profiles is fulfilled after obtaining flow angles.

3.3.1 Calculation Models

The inverse task calculation of preliminary design solves a set of equations, with the implementation of inlet and outlet boundary conditions, to search for the criteria of unknown flow angles as a function of generated design variables — flow coefficient and stage loading (both which relate to turbomachinery losses) — that are based on existing correlations. Details about the theoretical and mathematical background of the preliminary design tool can be found in the work of Moroz et al. [126]. Although there is a major three-dimensional aspect to the flow in a radial turbine, an approximate verification of 1D formulation is used in the PD step to provide an estimation of sizing data that depend heavily on the selected empirical loss methods [127].

The initial estimates obtained from the MATLAB meanline code (without loss models) are used as input data to AxSTREAM. The preliminary design tool generates a pool of many design solutions based on the given boundary data and geometric constraints; the required input data to the software is listed in Table 3.9. The supplied license of the software does not provide the capability of having a fluid mixture (such as that present in the cycle) so the working fluid is taken as pure supercritical CO₂ assigned by the AxS Carbon Dioxide property method which is a modified Redlich-Kwong equation of state that covers a wide range of pressures and enthalpies [128].

Table 3.9 AxSTREAM PD input requirements

Boundary conditions	Geometric parameters
Inlet total pressure	Stator outlet mean diameter
Inlet total temperature/ Inlet total enthalpy	Rotor blade height ratio
Outlet total pressure	Rotor diameter ratio
Mass flow rate	Rotor inlet height
Inlet flow angle	Hub reaction
Shaft rotational speed	Stator inlet diameter
Turbine type	Rotor outlet hub diameter

3.3.2 Loss Models

AxSTREAM employs the Craig and Cox system of loss calculation that relies on the combination of basic profile loss and secondary loss parameters. The profile loss factor depends on entry/ exit flow angles, plus blade length, pitch, and inlet and throat area; profile losses are corrected for Reynolds number, incidence angles, Mach number and

trailing edge thickness using previously validated graphical chart correlations with linear interpolation. Meanwhile, the secondary loss factor depends on inlet/ outlet flow angles, blade length and pitch with corrections for aspect ratio and Reynolds number. Default empirical methods and loss models for radial turbines embedded in AxSTREAM are used in the design tasks and are summarised in Table 3.10.

Table 3.10 AxSTREAM loss models calculation settings

Property	Model
Primary (profile) losses	AxS by Mitrohin-Stepanov
Secondary losses	AxS by Stepanov
Deviation angle	AxS Subsonic-Zero (real gas)
Blockage	Fixed loss factor
Nozzle leakage/	AxS default RT
Rotor tip leakage	

The kinetic energy that is not utilised in the corresponding component (i.e. loss) of the stator nozzle and rotor blade is a function of the loss ratio of each component which is assigned by the selected loss models. The loss in power due to the flow exit velocity also affects the specific work output of the turbine ($\dot{w}$).

$$\dot{w} = \Delta h_{tsisen} (1 - NLR - RLR - EELR) \quad (3.53)$$

where NLR, RLR and EELR are ratios of nozzle loss, rotor loss and exit energy loss respectively assigned by the calculation models.

Auxiliary losses (Δh_{loss}), which include disc friction and windage losses, can also be used to calculate the turbine specific work per unit mass generated by the rotor such as:

$$\dot{w} = C_{U4}U_4 - C_{U5}U_5 - \Delta h_{loss} \quad (3.54)$$

Thus the turbine efficiencies (η) and power output ($\dot{W}$) are affected by losses through:

$$\begin{aligned} \dot{W} &= \dot{m}\dot{w} \\ &= \dot{m}\Delta h_{tt} \\ &= \dot{m}\Delta h_{tsisen} \eta_{ts} = \dot{m}\Delta h_{ttisen} \eta_{tt} \end{aligned} \quad (3.55)$$

The generated power can be related to the rotor velocity triangles by using the Euler turbomachinery equation as such:

$$\dot{W} = 0.5\dot{m}[(U_4^2 - U_5^2) - (W_4^2 - W_5^2) + (C_4^2 - C_5^2)] \quad (3.56)$$

3.3.3 Design Space Explorer

All the generated turbine solutions can be explored in depth in the counterpart of the PD tool of AxSTREAM, the design space explorer, where designs are filtered through by applying constraints and limitations to allow trade-offs between power, efficiency, geometric design and velocities. Table 3.11 shows the filters applied to the design space to narrow the results selection process. The turbine solutions that do not fit within the limits are automatically discarded and the selected designs can be examined in terms of flow path geometry. This option also allows for a quick study of the influence of crucial parameters on performance and size at the early stages of preliminary design.

Table 3.11 Design space explorer constraints

Parameter	Unit	Constraint
$\dot{W}$	MW	$\geq \dot{W}$ from cycle Section 4.1.4
$\eta_{ts} \ \& \ \eta_{tt}$	%	≥ 80
ϕ	-	< 1
U_4	m/s	≈ 450
α_5	°	≈ 0

3.4 Turbine Analysis

Task 5. To assess different working designs based on performance requirements by means of throughflow analysis

The selected turbines from the preliminary design step are then analysed using the throughflow tool of AxSTREAM. Meanline analysis can be performed by using only the mean curvature section of the blade. Whereas a more refined streamline benchmarking calculation is done using the direct solver that considers the cascade geometry at hub, mean and tip for both stator and rotor. Kinematic and thermodynamic parameters are obtained along the blade height at three sections (hub, mean and tip). The aim is to get more precise performance parameters of the turbine before proceeding to blade profiling and design (if needed).

3.4.1 Problem Formulation

Two direct task formulations are present in the AxSTREAM analysis solver, either determining mass flow rate for the given outlet pressure, or vice versa. The setup for calculating the mass flow rate from the pressure drop is used for this project. Continuity, energy, process and state equations are solved with specified inlet conditions and downstream pressure for the fixed turbine geometry parameters obtained from the preliminary

design stage (rotational speed, mean component diameters and blade lengths, cascade metal angles, chord lengths, blade count, thicknesses), plus the estimation of losses from the inscribed loss model correlations for aerodynamic performance analysis. Loss models are kept the same as those in Subsection 3.3.2.

3.4.2 Convergence Criteria

Convergence control over the streamline calculations can be achieved by specifying the following:

- Number of iterations during loss calculation and equation solving is set to 50 because in this step solutions do not require a large number of iterations for convergence
- Number of failed iterations in sequence is defined as 10 so as to stop the calculations if accuracy is not achieved and discard the design
- Solution accuracy limits designated at 10^{-6} to set a threshold for admissible solutions
- Relaxation coefficient set as 0.2; higher values would decrease solution time but might incur inaccuracies within the solution calculation
- Default settings for all other parameters are based on the AxSTREAM software manual provided by SoftInWay

An example of a converged solution for a three-sectioned streamline calculation is shown in Figure 3.9; once the values of mass flow rate and power stabilise, convergence is reached. The equation and convergence accuracies are values plotted in AxSTREAM that indicate proper solution calculation and the quantities relate to the divergence from initial guesses.

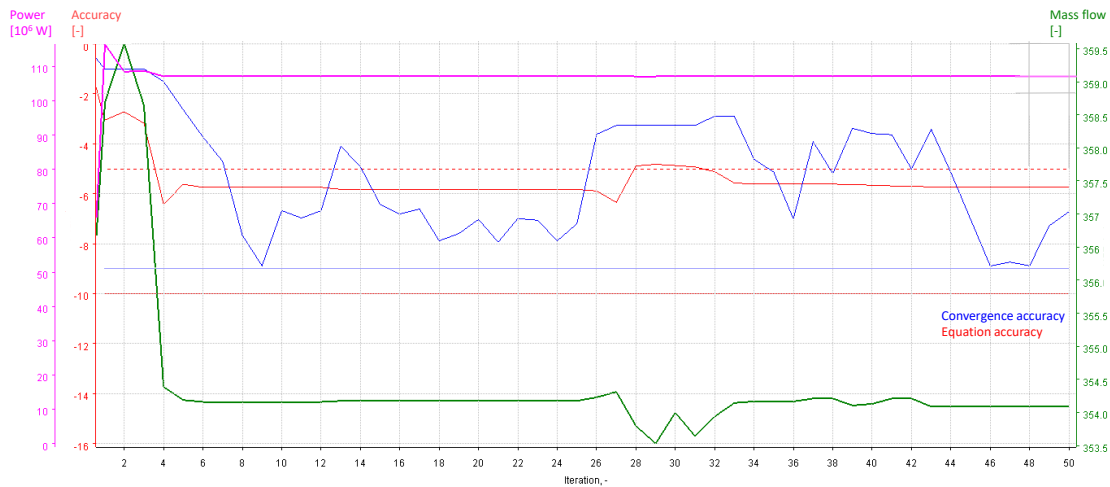


Fig. 3.9 Snapshot of an AxSTREAM converged throughflow calculation example

Results Analysis

Once some results are obtained for a select number of turbines, the user can start the post-design process of reviewing and editing the turbine flow path if needed; turbines that have not shown good performance or have not achieved solution convergence are discarded at this step instead of trying to optimise and edit the design. The outcome of the streamline calculations include loss and velocity charts, updated velocity triangles, meridional flow path and the enthalpy- entropy diagram, all which can be saved to the main project database to override performance results from the PD step. The project database will hold all relevant turbine information for each section (hub, mean, tip) as well as details on geometric and loss values for the design. Another check on the required constraints of Table 3.8 is performed in case any design fails to meet the criteria so as not carry it further to higher fidelity simulations.

3.5 Computational Fluid Dynamics

Task 6. To study turbine flow-field and evaluate turbine performance by employing numerical methods of advanced flow modelling

AxSTREAM provides tools that generate geometries based on 1D meanline design and allow analysis using higher fidelity models passing through streamline analysis, axisymmetric 2D computational fluid dynamics (CFD), and reaching 3D CFD. AxCFD is the tool used for pre-processing, mesh generation, calculation and results post-processing. This step allows for the evaluation of the designed flow path, the comparison of results

between the different calculation models (meanline, streamline, 3D CFD), and the understanding of flow mechanisms within the turbine.

The full turbine domain with a mixing plane halfway between stator trailing edge and rotor leading edge is simulated using AxCFD. Boundary conditions and working parameters are directly taken from the AxSTREAM project case saved after 2D analysis with default uniformity across hub, mean and tip sections taken as the spanwise average. Rotational speed and mass flow rate are also automatically inherited from the project database but are altered for off-design simulations. Like other CFD tools, AxCFD solves decomposed Reynolds-Stress-Averaged Navier Stokes (RANS) equations with time-averaged density, pressure and energy conservation, and mass-averaged velocity equations in discretised fluid domains.

3.5.1 AxCFD Mesh Generation

Passage mesh creation is required before any aerodynamic assessment of the turbine. The way of defining the size and number of the mesh elements in AxCFD is through mesh quality, a value ranging between 1 (coarsest) and 10 (finest). A mesh sensitivity study is performed, with results discussed in Section 5.3, to allow for a compromise between accuracy (mesh refinement) and computational time. Structured HO-type mesh elements with the same quality in hub-to-shroud and blade-to-blade directions are used in all cases with additional parameters for mesh adjustment as displayed in Table 3.12; values are taken based on recommendations by SoftInWay.

Table 3.12 AxCFD mesh adjustment parameters

Hub to shroud	
Multiplier for number of cells between hub and tip	1
Multiplier for number of cells from hub to mean	0.1
Multiplier for number of cells from mean to tip	0.1
Multiplier for number of cells of equal height in the middle part of the grid	1
Boundary layer	
Multiplier for number of cells along O-grid thickness	5
Multiplier for boundary layer thickness	0.25
Minimum thickness factor of near-wall cell	1
Multiplier for mesh density along blade suction side	1
Multiplier for mesh density along blade pressure side	1
Multiplier for number of cells for blocks at LE and TE	1

The rotating boundary zone of the domain is matched close to the rotor blade domain while non-rotating domains are extended beyond the rotor trailing edge and stator leading edge to capture flow physics beyond the blades. A mixing surface plane exists between

the stationary and rotating frames. The meshing zones defined by points on leading edge (LE) and trailing edge (TE) are modified manually over the blade profile contour for hub and tip sections. A mesh quality check is performed; good quality elements are described as those with no (or minimal) Jacobian determinant less than 0.3 and no (or minimal) angle parameter less than 12° .

3.5.2 AxCFD Solver Setup

The $k-\omega$ SST turbulence model is used for its suitability in viscous turbomachinery simulations based on previous experience [87, 106, 129, 130], with a medium turbulence intensity of 5%. The first order upwind scheme is implemented due to convergence restrictions when using second-order. Fluctuating and considerably high criteria for some residuals in the range of 10^{-5} are accepted for simulations of similar applications as shown by Wei [96] and Takagi et al. [131], but convergence is also checked against inlet/ outlet mass imbalance of within 0.2%. AxCFD also allows calculations over 2D spanwise fluid domain sections via axi-symmetric modelling which is elaborated on in Section 5.3.1.

3.6 Stress Analysis

Task 7. To investigate the mechanical integrity of the turbine by performing computational stress calculations

After analysing the turbine designs and selecting a suitable working model based on performance criteria, structural analysis can be performed to check if the structure is appropriate for the operating environment (aerodynamic and mechanical loads) and to suggest design or material improvements based on results. AxSTRESS is the finite element analysis (FEA) tool within AxSTREAM to address the structural mechanisms by computing relevant quantities of stresses and strains under aerodynamic and inertial loads. Static stress analysis is only carried out for the rotating part of the selected turbine where centrifugal stresses are concentrated.

3.6.1 Shaft Sizing

The design of the turbine wheel required for stress calculations on the rotating part of the machine can only be obtained once the shaft size is determined because the disc is placed on the outer diameter of the shaft. Thus, initial estimations of shaft sizing are computed based on works found in [132, 133]. The shaft is assumed to be hollow, to allow for cooling flow to lower the shaft metal temperature, with inner (d_i) and outer (d_o) diameters that define the shaft thickness $t_{shaft} = d_i/d_o$. The stresses on a surface point at the outer diameter of the shaft constitute normal stress (σ_x) which is the combination of

axial loading and bending moments, and twisting shear stress (τ_{xy}). The component of each imposed stress is given below:

$$AxialLoading = \frac{4F_{ax}}{\pi(d_o^2 - d_i^2)} \quad (3.57)$$

$$Bending = \frac{64M_b \left(\frac{d_o}{2} \right)}{\pi(d_o^4 - d_i^4)} \quad (3.58)$$

$$Twisting = \frac{32M_t \left(\frac{d_o}{2} \right)}{\pi(d_o^4 - d_i^4)} \quad (3.59)$$

The axial force (F_{ax}) is obtained from simulation results of AxSTREAM whereas the torque/ twisting moment is $M_t = \dot{W}/\omega$ and the bending moment is related to the mass of the turbine rotating part (disc + rotor – $m_{rotating}$) through $M_b = m_{rotating} \cdot g \cdot \Delta l_{shaft}$. The mass of the turbine is calculated by multiplying the volume of the solid domain by the density of the chosen material. The volume of the blade solid domain is obtained by subtracting the volume of the fluid domain from the volume of the whole passage domain for the total number of blades. The axial length of the turbine is taken from AxSTREAM and the length of the shaft is assumed to be 3.5 times the length of the turbine ($l_{shaft} = 3.5 \cdot Z$); the shaft is assumed to be fixed on its ends and the turbine is placed in the middle ($\Delta l_{shaft} = l_{shaft}/2$). Results of the shaft estimates are found in Section 6.4.

Using Mohr's circle for non-zero principal stresses, the von Mises stress (σ') is given as:

$$\sigma'^2 = (AxialLoading + Bending)^2 + 3(Twisting)^2 \quad (3.60)$$

The maximum allowable von Mises stress (σ'_{all}), based on the distortion-energy theory of failure [132], is related to the 0.2% yield strength (S_y) of the material at the operating temperature by a safety factor (SF):

$$\sigma'_{all} = \frac{S_y}{SF} \quad (3.61)$$

Thus, from Eq. 3.57 – 3.61 the outer diameter of the shaft can be calculated assuming the shaft is made up of a Nickel-based superalloy (IN 625) with a density $\rho_{shaft} = 8442$ kg/s [134] and a shaft safety factor SF= 1.6.

3.6.2 Mesh Generation

The 3D impeller model is created in AxSTRESS by designing the rotor hub shape (which will not affect the aero-thermodynamic flow path) while importing the rotor blade design

from AxSTREAM. The contour of the available blisk-type hub cap is adjusted in the meridional view of the blade; the spline base curve of the hub is extended beyond the blade leading and trailing edges with assigned fillet radii at each position. The impeller contour dimensions are based on the obtained shaft size and adjusted for stress reduction; the disc is assumed to be fixed on the whole surface of the rotation line formed between the impeller and the shaft. Then an auxiliary curve is tweaked to define the platform shape of the airfoil hub section position. The unshrouded turbine does not require cap tip definition.

Similarly to the mesh parameters of AxCFD (described in Section 3.5.1) the mesh quality from hub-to-shroud and blade-to-blade is varied in a mesh sensitivity study. Unstructured quadratic tetrahedral elements are used for discretised solving of the linear elastic stress-strain equations.

3.6.3 Problem Formulation

A generic Inconel Alloy 718 material is taken for the rotor of the turbine for its relatively high yield and good tensile/ creep-rupture performance under extreme operating temperatures [135]. The assigned material properties near the turbine operating temperature are listed in Table 3.13 [136, 137].

Table 3.13 IN718 material properties at turbine operating temperature [136, 137]

Parameter	Unit	Value
Young's modulus of elasticity	10^3 MPa	134.26
Material density	kg/m ³	8220
Coefficient of thermal expansion	10^{-6} °C	16.04
Poisson ratio	-	0.333

Boundary conditions and blade geometry are seamlessly imported from AxSTREAM and aerodynamic loads are taken from the AxCFD isolated rotor model (simulated without stator domain as described in Subsection 6.3.4). Static stress analysis enables the calculation of displacements and stresses induced by loads that do not cause significant inertia and damping consequences on the turbine wheel.

3.7 Methodology for other Objectives

The methods followed for the attainment of each set of research objectives have been described so far except for the first and last research objectives defined in Section 1.4. A short narrative is given below relating to Task 1 and Task 8.

Task 1. To identify the gaps in knowledge and gather relevant information by conducting a thorough literature review and communicating with experts in the field

The means to fulfil this aim is presented in Chapter 2 by performing a thorough literature review and going through information relating to various aspects of the work. The defined modelling and design methodology is adopted from works of other researchers and modified to fit the purpose and requirements of the set objectives of this research project while taking into account limitations of computational software.

Task 8. To determine areas for further work based on the availability of enabling technologies

Although this objective is not fulfilled via a specific method or tool, it is important to note the mechanism in which the areas and suggestions are identified. Based on the results of all the different parts of this work, a collection of questions regarding the actual applicability of the radial turbine with the current technologies come up. Even if some of the answers to the inquiries, which are presented in Section 7.2, are founded on guidance obtained from literature documentations, the author finds that that the suitable placement of such information belongs to the discussion chapter and not in the literature review chapter. The provided exploration of enabling technologies is also a summary of discussions with researchers from the US National Energy Technology Laboratory, Oak Ridge National Laboratories, and Sandia National Laboratories.

3.8 Assumptions and Limitations

As with any computer-based methodology, assumptions lead to inaccuracies of results and limitations to applicability and use. A list of the major suppositions that are recognised to influence the results is given below. This list applies to the various software tools employed for the design of the turbine.

- The use of an equation of state for fluid property calculations instead of real experimental data in all parts of methodology
- Quantitative assumptions in the cycle modelling especially that of the turbine efficiency because it is presumed before design and performance calculations
- Computational modelling of a non-proven physical cycle that does not take into account environmental conditions and geometries of components
- Assumption of pure CO₂ working fluid in turbine design steps while knowing that other fluid components are present
- The varying use of the fluid property method due to software restrictions

- The absence of loss models in the meanline turbine design (MATLAB code)
- Lack of quantitative validation cases
- The loss models in AxSTREAM may require recalibration for the radial turbine cases investigated as will be outlined throughout the thesis

Chapter Summary

The purpose of this chapter is to give the reader enough information to be able to replicate the study by answering the how questions of conducting research. Figure 3.10 summarises the methodology followed in this PhD work.

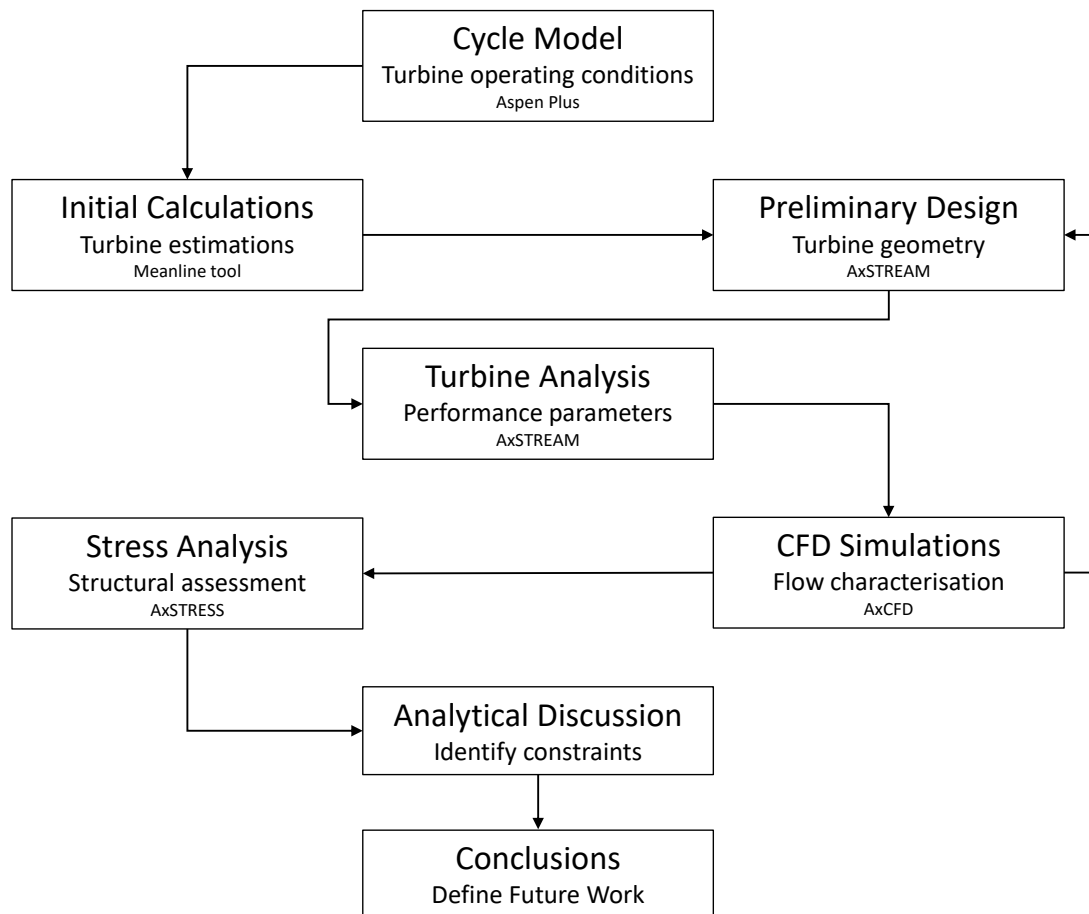


Fig. 3.10 Thesis methodology

CHAPTER 4

Cycle Modelling and Meanline Turbine Design

This chapter presents the cycle model simulations that are used to compare the performance of a cycle that employs turbine cooling flows to one that does not. In addition, it contains a study that examines the effect of turbine inlet temperature and turbine efficiency on the thermodynamic cycle output. The cycle modelling section concludes with the details of the cycle that is adopted in this work with its relevant turbine boundary conditions used for the component design. A parametric turbine performance study is also described using the meanline design code, and initial turbine design estimates follow.

4.1 Cycle Modelling Results

Before proceeding to process modelling of an uncooled cycle and using it as a basis for turbine design, it is reasonable to try and understand if there are any cycle benefits accredited to the removal of turbine cooling beyond that of reducing design complexities. From a practical point, a comparison between thermodynamically-identical cycles is conducted where cooling is the only limiting distinction.

4.1.1 Optimised Cycle Comparison

In an attempt to compare the performance of an uncooled cycle (the target cycle of this work) to a cooled one, like-for-like working conditions are applied. Conditions of the optimised cycle model from the work of Scaccabarozzi et al. are adopted [73] which was compared against the data from IEA GHG report [34]. Optimisation work, to achieve maximum cycle efficiency target, was carried out by Scaccabarozzi et al. [73] after performing a sensitivity analysis on the most relevant cycle-affecting parameters.

One of the parameters that allows the scalability of the power plant is the input mass flow of the fuel which in turn affects the flows of all other streams. As a means to keep the thermal energy of feedstock (LHV) comparable to that of Scaccabarozzi et al. [73] and to the results from the IEAGHG report[34], the inlet flow rate of natural gas is set to 16.52 kg/s which gives a thermal power input almost identical to the cycles of the mentioned authors. For the sake of comparison, the uncooled turbine is assumed to be capable of operating at a full-size power scale.

The almost identical thermodynamic conditions of pressures, turbine inlet temperature and turbine isentropic efficiency as the optimized cycle from Scaccabarozzi et al. [73] that are implemented in the current model are summarised in Table 4.1.

Table 4.1 Input parameters for uncooled optimised cycle

Input parameter	Unit	Value
Turbine inlet pressure	bar	283.62
Combustor outlet temperature	°C	1123.79
Turbine isentropic efficiency	%	89
Turbine pressure ratio	-	6.02
Second flow compressor (CPR-02) outlet pressure	bar	81

The comparison between some cycle performance results is viewed in Table 4.2. Generally there is good agreement in the results with some discrepancies that can be attributed to the differences in assumptions as mentioned in Subsection 3.1.3, and because of using two compression stages instead of four [73] which lowers the power required to drive the compressors.

Assuming adiabatic conditions, the absence of turbine cooling gives a higher turbine outlet temperature which implies a lower temperature drop across the turbine contributing to a lower turbine specific work. However, the higher combustion flow rate, required to reach the cycle desired temperature of the cycle, balances the drop in turbine specific work. Overall, the cycle with the uncooled turbine definitely gives a superior cycle efficiency compared to the cooled case, a 1.54% efficiency point gain. The heat transfer associated to preheating the turbine cooling flow within the regenerator (which also contributes to expansion losses) is removed and thus the detrimental efficiency penalty affiliated with this process is eliminated, causing a beneficial impact on the cycle efficiency.

In fact it is suggested that while turbine cooling allows an increase in TIT which enhances cycle performance, it also increases energy losses in turbines and thus leads to a lower aerodynamic efficiency [138] that in turn would have a great negative impact on the net cycle efficiency [139]. Scaccabarozzi et al. [73] noted that the maximum net cycle efficiency is achieved at a relatively low turbine inlet temperature (lower than the maximum simulated TIT value) because of the substantial cooling mass flows required if a high combustor outlet temperature is implemented.

Table 4.2 Comparison of optimised cycle between [73] model and uncooled turbine model

	Unit	Results of this work	Results of Scaccabarozzi et al. [73] optimised cycle
Thermal energy of feedstock (LHV)	MW_{th}	768.21	768.31
Turbine power output	MW_e	610.55	609.74
Recycle flow compressors	MW_e	88.00	97.81
Natural gas compressor	MW_e	3.97	-
Air separation unit	MW_e	85.51	-
Storage compressor	MW_e	0.25	-
Net electric power output	MW_e	432.82	421.06
Turbine outlet temperature	$^{\circ}C$	830.53	783.81
Turbine inlet flow rate	kg/s	1567.7	1513.7
Net electric efficiency (LHV)	%	56.34	54.80

4.1.2 Case Study

After demonstrating that the lack of turbine cooling (in a theoretical cycle where the material can handle the elevated temperature) is not deleterious and actually favourable to cycle efficiency, it is appropriate to consider more realistic scenarios of lower turbine inlet temperatures that could be withstood by current blade materials without the necessity of cooling. For the initial case, all flow conditions are set identical to the values of the optimised cycle (in Subsection 4.1.1) and only the turbine inlet temperature is changed to $950^{\circ}C$. The new outcome of the cycle efficiency is 51.46% which is around 8.7% lower than the uncooled optimised cycle.

A slightly higher cycle efficiency of 52.22% is obtained when the cycle is tuned to give turbine conditions corresponding to $TIT = 950^{\circ}C$, $TIP = 297$ bar and a lower pressure ratio of 4.95 compared to that of the optimised cycle. All other assumptions are kept the same including the relatively good isentropic efficiency of the turbine (89%).

This configuration of conditions ($TIP = 297$ bar, $\Pi = 4.95$) is considered for further examination regarding the effect of the turbine isentropic efficiency at two cooler but still considerably high temperatures of $900^{\circ}C$ and $950^{\circ}C$.

The previously mentioned $950^{\circ}C$ TIT cycle surmises a developed state of the art turbine within the layout; however due to the unconventionality of the cycle working fluid and the elevated temperatures and pressures at turbine inlet, either modifications on off-the-shelf turbine designs or new ad-hoc integrations need to be achieved to satisfy the requirements [34]. This immaturity of the turbine can manifest itself in a less efficient turbine within the Aspen Plus model.

The almost linear effect of changing the turbine efficiency (η_{isen}) on the net electric efficiency of the cycle (η_{NE}) is shown in Figure 4.1. For both temperature cases, with every 0.5% point decrease in expander efficiency, the cycle performance criteria drops by

almost 2% efficiency points. This variation is correlated to that done by Scaccabarozzi et al. [73] except that for the uncooled cycle, the reduction in cycle efficiency due to lower η_{isen} is more pronounced. As expected though, the higher temperature of 950°C allows the net cycle efficiency to be almost 1.5% points higher than the 900°C temperature over the range of considered turbine efficiencies. Thus, it is quite crucial for the designed turbine to have good performance and in future to be able to have a wide material selection for elevated temperatures.

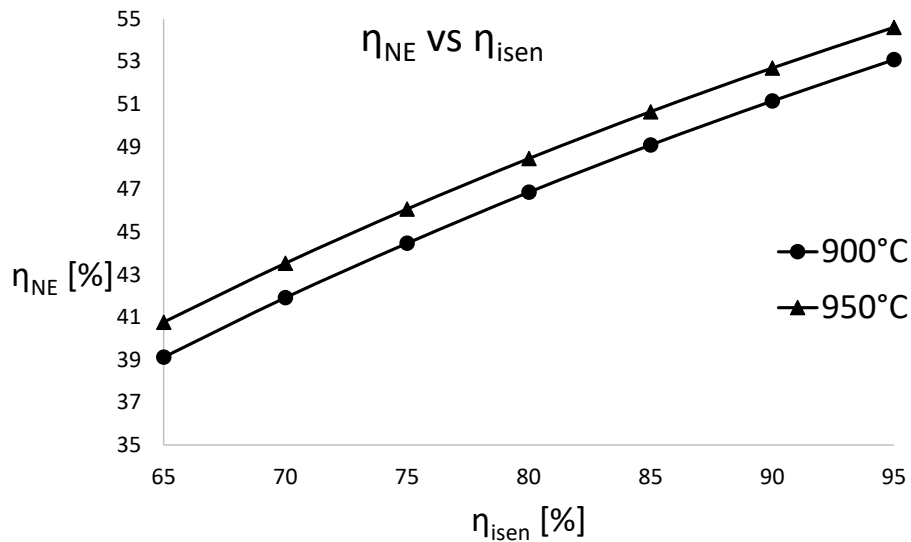


Fig. 4.1 Effect of turbine isentropic efficiency on cycle net electric efficiency for uncooled model

Although the cycle efficiency for lower TITs (with varying turbine isentropic efficiency) is not in the range of the NET Power projections of 55 – 59% [36], the values of cycles with moderately efficient turbines ($\eta_{isen} > 80\%$) still compete with other oxy-fired cycles (mentioned in Chapter 1) as illustrated in Table 4.3. In fact, the uncooled version, which still encompasses the zero-emissions and compact advantages of the Allam cycle, can achieve outstanding cycle efficiencies with less complexity in the turbine design stage.

Table 4.3 Cycle efficiency comparison of different oxy-fuel gas cycles [34]

Cycle	Cycle efficiency [%]
MATANT	40 – 49
SCOC-CC	45 – 48
CES	45 – 50
E-MATANT	46 – 47
ZEITMOP	46 – 51
Uncooled version of NET Power	46-55
AZEP	49 – 53
Graz	49 – 54
NET Power	55 – 59

4.1.3 Radial Turbine Selection

As discussed in Section 2.2, supercritical CO₂ cycles, regardless of whether they employ cooling or not, have the beneficial characteristics of small turbomachinery components due to the high fluid density. Suggestions from literature limit the applicability of radial turbines to cycles of up to 30 MW_e [85]. The criteria for that was proposed on the basis of closed sCO₂ cycles with lower operating pressures and temperature compared to conditions witnessed in cycles similar to the Allam cycle. This means that probably there could be a restatement of a new power output ceiling for cycles that operate at such extreme pressures and temperatures.

As introduced in Subsection 3.2.2, the numerical value of a turbine specific speed is used as an index of power output and passage size [122]. Figure 4.2 shows that for n_s in the narrow range of 0.3 – 1.0, where this span corresponds to small turbine passage areas, a 90° inward-flow radial (IFR) turbine can reach high efficiencies in contrast with the conventional axial turbines which have a much broader spectrum of peak performance [122].

From Eq. 3.11, one can see that a larger-scale turbine (higher volumetric or mass flow rate) corresponds to a higher specific speed value for fixed ideal heat drops and rotational speeds. Thus, taking a within-the-range recommended value of $n_s=0.7$ [90, 122] and knowing the ideal enthalpy drop from the cycle model ($\Delta h_{id}=284,370$ J/kg) plus having done some calculations on possible rotational speeds being in the range of 25,000 rpm, through initial sizing of Section 3.2 (i.e. $\omega=2,618$ rad/s) then from Eq. 3.11 and 3.13 the corresponding maximum allowable mass flow rate is:

$$\dot{m}_{max} = \dot{Q}_{max} \cdot \rho_{out} = \left(\frac{0.7 \Delta h_{id}^{0.75}}{\omega} \right)^2 \cdot \rho_{out} \approx 354 \text{ kg/s} \quad (4.1)$$

where ρ_{out} is around 33.2 kg/m³ from the cycle model.

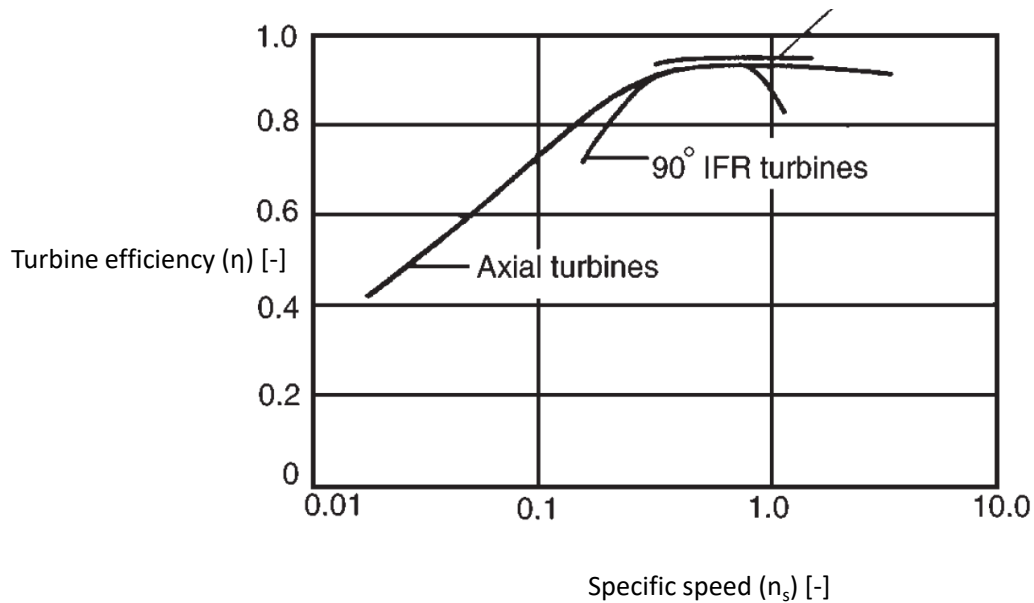


Fig. 4.2 Specific speed and efficiency characteristics for different turbine configurations [122]

4.1.4 Modified Cycle Results

The limit obtained for the largest size of a radial turbine operating under the conditions of a dense working environment, at $TIT = 900^\circ\text{C}$, $TIP = 297$ bar and $\Pi = 4.95$, is integral to assigning the scale of the cycle and assessing the applicability of such a size. The cycle model is altered with the implementation of $\dot{m}_{max}$ and an assumed medium-value turbine isentropic efficiency of 85%, and the final modified cycle performance parameters are listed in Table 4.4. Although the net electric power output of 67.33 MW_e is not within the reach of the suggested 700 MW NET Power plans, the net work of this cycle is in the scale of gas turbines used in small industrial and commercial applications by major OEMs. The market for turbines in the range of less than or equal to 70 MW_e is believed to grow considerably in the near future [140].

To summarise, Table 4.5 offers the turbine boundary conditions which are used to design, size and assess a radial expander that could be fit for use. The obtained turbine fluid components are comparable to the NET Power working environment reported in literature (reported in Table 2.3) with a slightly higher CO_2 mole fraction. This is because in the original NET Power cycle the combustion products are mixed with the turbine cooling stream which has a high CO_2 component. Although the major fluid component entering the turbine is CO_2 with 94.06% mole fraction, there are other fluid contaminants that could affect turbine design and performance. The effect of their presence is discussed in Subsection 4.2.3 and in Section 7.1.

Table 4.4 Modified cycle performance parameters

Data	Unit	Value
HTHE heat duty	MW	242.92
LTHE heat duty	MW	43.42
Thermal energy of feedstock (LHV)	MW _{th}	137.18
Turbine power output	MW _e	99.31
Vapor phase compressors	MW _e	3.96
Dense phase compressors	MW _e	8.86
Oxidant compressor	MW _e	3.09
Total flow compressors	MW _e	15.91
Natural gas compressor	MW _e	0.74
Air separation unit	MW _e	15.29
Storage compressor	MW _e	0.045
Net electric power output	MW _e	67.33
Net electric efficiency (LHV)	%	49.08

Table 4.5 Radial turbine boundary conditions from cycle model

Data	Unit	Value
Inlet temperature	°C	900
Inlet pressure	bar	297
Mass flow rate	kg/s	354.2
Outlet pressure	bar	60
Shaft power	MW _{th}	100.7
Fluid component mole fractions		
CO ₂	%	94.06
H ₂ O	%	4.13
N ₂	%	1.14
Ar	%	0.55
O ₂	%	0.12

4.2 Meanline Design

Following the procedure described in Section 3.2, a range of turbine sizing parameters can be obtained for different power scales. Similar to Figure 2.4, the speed and size dependency of a radial turbine on power scale for a supercritical CO₂ cycle with comparable conditions to the NET Power cycle is displayed in Table 4.6; the approximate values are an outcome of rough initial meanline design calculations as will be discussed. In contrast to a conventional sCO₂ closed Brayton cycle, the Allam cycle will have smaller turbomachinery elements and higher speeds due to its much higher pressures and temperatures.

Table 4.6 Scaling of turbine speed and size based on power range

Power range [MW]	0 – 1	1 – 10	10 – 50	50 – 100
Rotational speed [rpm]	450,000	100,000	50,000	25,000
Size (diameter) [mm]	20	80	200	400

4.2.1 Parametric Study

The turbine boundary conditions (of Table 4.5) from the cycle model are supplied together with a value of specific speed as input to the meanline code. Unlike other parameters, which are either dictated by the boundary conditions or are an outcome of the design (i.e. performance), the specific speed n_s is a non-arbitrary input parameter that is bound by limitations imposed by turbine size and speeds. A parametric study on the effect of specific speed values on the turbine design is conducted to find the range of working parameters.

Figure 4.3 displays the variation of rotational speed and rotor inlet radius with specific speed values. The plot gives an initial idea of the possible order of magnitude of rotational speeds and dimensions for the radial turbine operating under the cycle conditions. As expected, low values of n_s correspond to low speeds and large passages. Small rotors are harder to manufacture and limit the sizes of radial passages between stator and rotor which would affect clearances and increase the proportion of associated losses; on the other hand, large radii can incur extra mechanical stresses that could seriously inhibit operation. The driving of electrical equipment via gearing imposes constraints on allowable rotational speeds at the operating range of power output ($\approx 60 - 70 \text{ MW}_e$). This justifies the suggested range of specific speeds in literature [90] and thus indicates that for this turbine the preferred selection of n_s falls somewhere between 0.35 and 0.75.

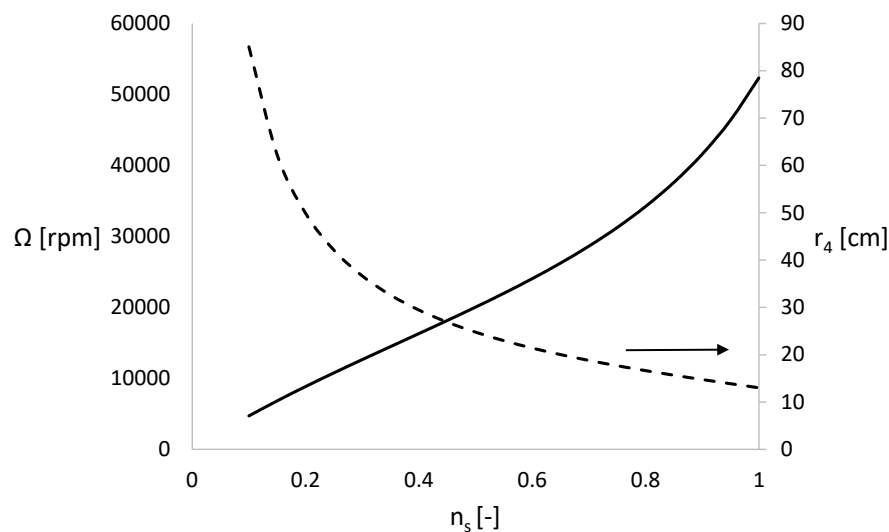


Fig. 4.3 Influence of specific speed variation on rotational speed and rotor inlet radius

The rotor inlet design, rotational speed (calculated from n_s), stage loading, degree of reaction and flow coefficient are all inter-dependent parameters. Results of the parametric study at three specific speeds of 0.37, 0.62 and 0.73 — to cover a range of speeds — (their corresponding rotational speeds are 15,000 25,000 and 30,000 rpm respectively) are shown in Figure 4.4. For $\psi < 1$, the 90° inward-flow radial tends to have a constant high degree of reaction at all speeds and a relatively low flow coefficient. This contributes to a negative relative flow velocity at rotor inlet ($\beta_4 < 0$ or W_4 opposite to rotation) compared to a radial inflow turbine with non-zero inlet blade angles. Higher blade loading coefficients correspond to non-zero inlet blade angles at lower R and increased ϕ ; at higher rpms, greater loading can lead to flow coefficients approaching unity which is not an adopted practice. A compromise between the parameters is required to meet all the design criteria set by experience of researchers in the field. The trade-off is done between blade loading, to allow for a degree of reaction that supplies a higher static enthalpy/ pressure drop across the stator ($R < 0.5$), and a moderate specific speed, that accommodates constraints on rotational speed and rotor sizing, while maintaining ϕ in the range of 0.2 – 0.5.

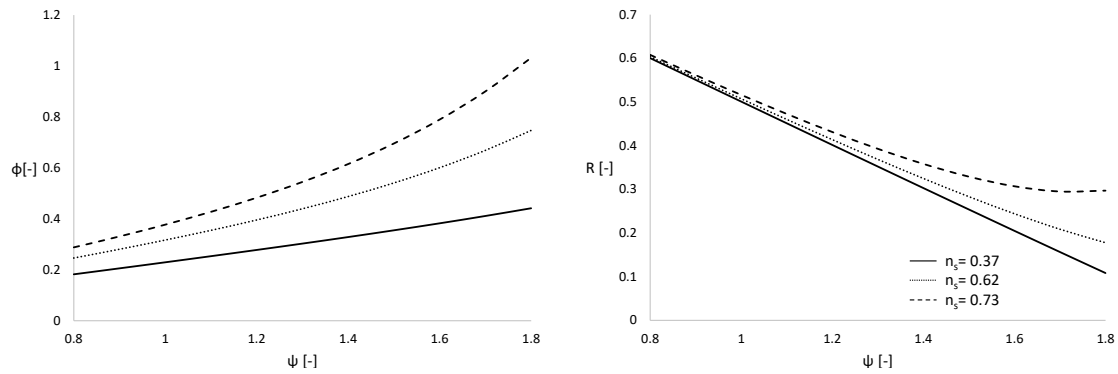


Fig. 4.4 Effect of blade loading coefficient on flow coefficient (left) and degree of reaction (right) at 3 different specific speeds

4.2.2 Operating Range

Having selected some design parameters based on the parametric study, a chosen case is used to obtain a range of operation for the radial turbine. It is not necessary at this point to investigate many possible designs as this step is used to populate initial geometric criteria ranges into AxSTREAM.

Relevant initial turbine performance variables are listed in Table 4.7. The estimated efficiency value does not account for any losses, and unless the turbine blade designs are properly optimised, the efficiency of the final design is anticipated to be lower than the calculated 86.39%. Similar quantitative estimates for stage loading, flow coefficient and degree of reaction are reported in other supercritical CO₂ radial turbines [96, 102]; these values are used as reference to the next steps of turbine design.

Table 4.7 Initial turbine design parameters

Data	Unit	Value
$\dot{Q}$	m^3/s	10.67
ω	rpm	25070
n_s	-	0.624
C_{0s}	m/s	811.4
v_s	-	0.671
η_{ts}	%	86.39
ϕ	-	0.488
ψ	-	1.40
R	-	0.325

A radial turbine cross-section with meridional dimensions of the outline design is shown in Figure 4.5. The maximum diameter and axial length of the turbine are in the range of 560 mm and 140 mm respectively which is indeed a very compact expander for a 100 MW (power to drive compressors as well) scale. Further geometric data of the rotor are listed in Table 4.8. The given design range is checked against the constraints set in Table 3.8.

Table 4.8 Initial turbine rotor geometric data

Data	Unit	Value
N_R	-	13
t_4	mm	6.867
q_4	mm	82.97
o_4	mm	59.40
A_5	mm^2	54290
t_5	mm	3.433
q_5	mm	46.56
o_5	mm	30.54

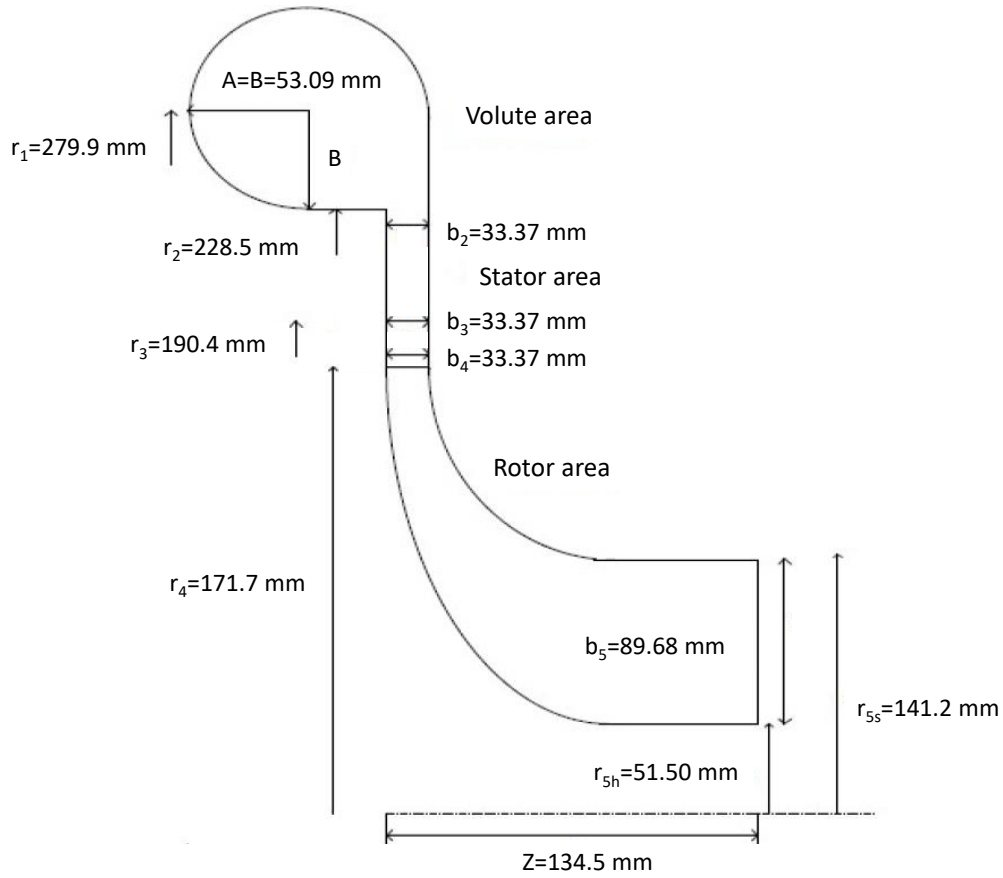


Fig. 4.5 Initial turbine meridional dimensions

The relevant rotor velocity triangles are visualised in Figure 4.6; the rotor tip speed is limited to $\approx 450 \text{ m/s}$ according to established design practice [125], and the design assumes zero swirl at exit. The absolute Mach number at rotor inlet being slightly > 1 indicates that supersonic flows occurring at the nozzle exit are expected because of the high expansion ratio and the thermal properties of sCO_2 ; the undesirable supersonic speed at nozzle outlet is a phenomenon that is reported in radial turbines using non-ideal gases (such as organic fluids and sCO_2) [102, 103].

The calculated thermodynamic quantities are direct outcomes of the chosen property method (REFPROP); for this case, a pure carbon dioxide fluid is used. The evolution of the main fluid properties across the turbine stage is recorded in Table 4.9. Total values are taken as constants in the stationary frame and in the vaneless passage between stator and rotor whereas in reality rapid and uncontrolled pressure changes will occur due to the shockwave formation between nozzle exit and rotor inlet. Entropy values, obtained from final design analysis are anticipated to be larger than the initial estimated quantities, accounting for loss generation across the turbine.

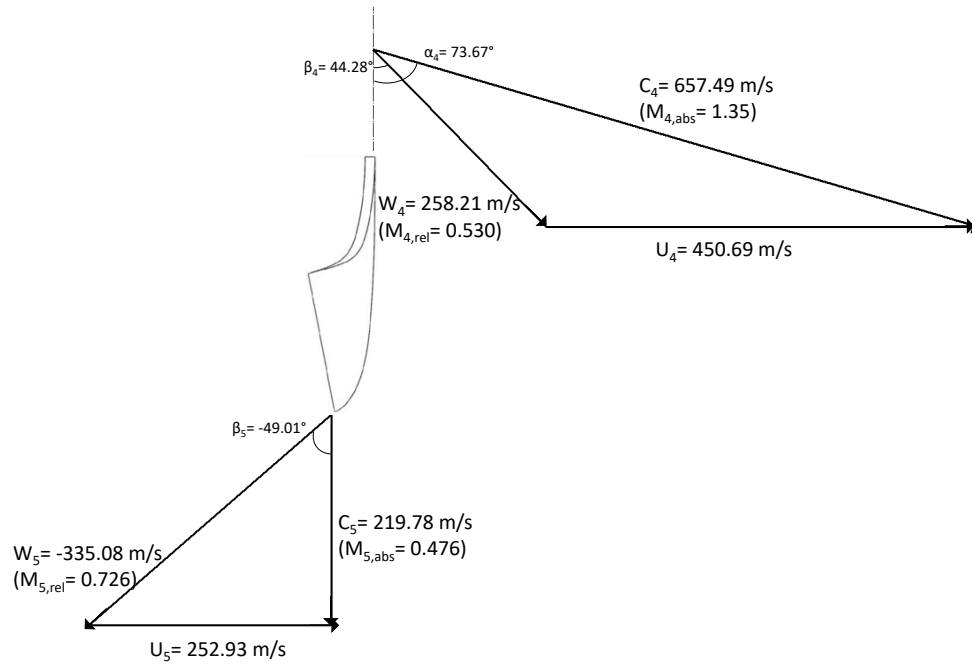


Fig. 4.6 Initial turbine velocity triangles

Table 4.9 Initial thermodynamic properties data at turbine stations

Variable	Unit	Volute inlet (1)	Volute outlet/ Nozzle inlet (2)	Nozzle outlet (3)	Rotor inlet (4)	Rotor outlet (5)
T_0	K	1173	1173	1173	1173	946
P_0	bar	297	297	283	283	60.0
h_0	kJ/kg	1480	1480	1480	1480	1196
ρ_0	kg/m ³	124.8	124.8	119.4	119.4	33.20
s	J/kg K	3112	3112	3121	3121	3155
T	K	1115	1078	1035	1002	926
P	bar	214	173	127	103	52.4
h	kJ/kg	1405	1357	1304	1263	1171
ρ	kg/m ³	96.73	81.38	62.91	53.23	29.68

4.2.3 Fluid Contamination Effect

The aforementioned design details in Subsection 4.2.2 are based on a fluid assumption of pure carbon dioxide, however, as revealed from the cycle model (results in Table 4.5), other components are present within the turbine working fluid notably water ($\approx 4\%$) and nitrogen ($\approx 1\%$). Assigning the same constraints and parameters to a design that encompasses the three main fluid components (CO_2 , H_2O , N_2) allows an assessment of the impact of the pure fluid assumption. The full meanline design results of the turbine with the fluid mix are found in Appendix B.

The differences between the turbine designs with pure CO₂ and with the fluid mix are negligible on the whole. The density of the fluid mix is around 6% lower than that of the pure fluid (117.5 kg/m³ compared to 124.8 kg/m³ at turbine inlet) because water and nitrogen behave as ideal gases (non-supercritical) at the operating temperatures and pressure so they are not quite as dense as the sCO₂. The lower density incurs slightly larger dimensions for the turbine with the three gases with a maximum diameter of ≈ 580 mm in contrast to the single-gas turbine design size of 560 mm. Although the generated design for the fluid mix does not fully encompass the required design considerations, it indicates that there should not be a significant change when assuming a pure carbon dioxide fluid. The performance estimate is subject to readjustment for the final design given that any fluid contamination will affect the efficiency of any turbomachinery component by a few percentage points. As mentioned in Section 3.8, the turbine design based on the assumption of the pure fluid is imposed by license constraints and will entail adjustments when designing the final turbine.

Chapter Summary

The comparative results of this chapter validate the notion that avoiding cooling in turbines will lead to fewer cycle heat losses and hence a more efficient system. A well-designed and efficient turbine with materials that can endure higher firing temperatures are integral aspects to achieve better cycle efficiencies. Finally, turbine operating conditions are obtained from a modified model established on the grounds of the NET Power cycle. The modified cycle still attains good cycle efficiency (49.08 %) through oxy-combustion and supercritical CO₂ as a working fluid, but is differentiated by its lower TIT= 900°C and $\Pi= 4.95$ at a smaller scale for the appropriateness of a radial turbine. Using the meanline design procedure, parametric studies of critical performance variables at the initial phase allow the establishment of an estimated range that is used to obtain possible geometries. For the required turbine scale of 67.33 MW_e (100.7 MW_{th}), the design has a diameter of near 560 mm and operates at rotational speeds in the range of 25,070 rpm.

CHAPTER 5

Initial Turbine Design

This chapter describes the stages of the turbine design and analysis which employ the AxSTREAM/ AxCFD tools that are part of the AxSTREAM wider suite. This phase involves the generation of preliminary turbine designs using initial guesses obtained with the meanline design, described in the previous chapter, and the selection of turbine geometries which match the design requirements of the formerly defined cycle. In order to illustrate the methodology, the steps leading to the selection of a design are presented along with all its relevant information, geometric parameters and velocity triangles, in addition to 2D/ 3D model analyses. The chosen turbine is analysed using throughflow formulations and its performance is compared to results obtained with the meanline design tool. Outcomes of 2D axi-symmetric CFD modelling and 3D turbine flow simulations are also discussed.

5.1 Preliminary Design Generation

The design phase that builds on the results of the cycle analysis and the meanline code consists of the definition of three-dimensional geometries which are then further investigated using higher fidelity tools. The tool employed in this phase is the preliminary design solution generator, which is part of the AxSTREAM design suite, which is initiated with defined settings (working fluid, loss models, number of stages), boundary conditions and geometry constraints. The boundary conditions required to obtain the preliminary design solutions are based on design specifications from the cycle requirements that are fixed, whereas all other parameters obtained from the meanline calculation are specified as a range of values. The particular range employed is defined based on the survey of literature together with outcomes of the sensitivity analysis described in the previous chapter.

The inlet total enthalpy and outlet total pressure are used as boundary conditions, and the criterion against which output designs are selected is based on the total-to-static efficiency. A vaned nozzle is used and cases for designs with and without volute are checked. The rotor inlet metal angle is set to 90° (i.e. straight radial blades at inlet) for manufacturability and structural reasons. Full input data is found in Table 5.1.

Table 5.1 Input data for preliminary turbine design

Variable	Unit	Value
Boundary conditions		
Inlet total pressure	bar	297
Inlet total enthalpy	kJ/kg	1480
Outlet total pressure	bar	60
Mass flow rate	kg/s	354.2
Inlet rotor metal angle	$^\circ$	90
Shaft rotational speed	rpm	25000 – 26000
Geometric parameters and constraints		
Stator outlet mean diameter	m	0.3 – 0.5
Rotor blade height ratio	-	2 – 3
Rotor diameter ratio	-	1.5 – 3
Rotor inlet height	m	0.01 – 0.04
Hub reaction	-	0.3 – 0.5
Stator inlet diameter	m	0.3 – 0.7
Inlet flow angle	$^\circ$	1 – 179
Rotor outlet hub diameter	m	0.05 – 0.3

5.1.1 Solution Generation

The preliminary design space allows the visualisation of the estimated set of turbine solutions for the given input data. An example of a generated design space with 193 possible solutions is shown in Figure 5.1 with each point corresponding to a particular turbine design. The total-to-static efficiency is represented by the colour of each point for both plots. Higher efficiency values coincide with the lower range of flow coefficients and higher velocity ratios (faster rotational speeds). v_s' is the parameter taken from the definition of the velocity ratio from AxStream, which coincide defined value v_s in the range between 0.4 – 0.7. In some cases, a qualification between efficiency and velocity ratio is needed to limit the induced rotor tip speed from high rotational speeds because of permissible material limits. Greater power is produced from the designs with higher efficiencies as expected.

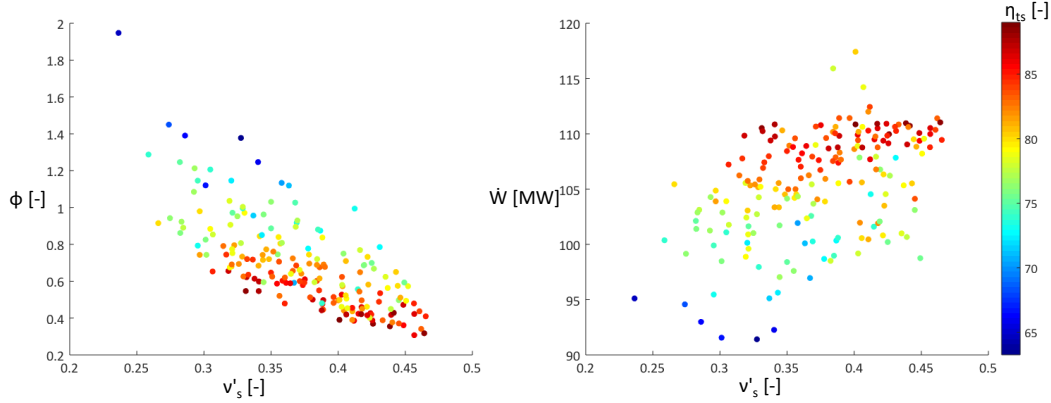


Fig. 5.1 Preliminary design space plots: flow coefficient vs velocity ratio (left), power vs velocity ratio (right)

From the preliminary design space, designs are filtered through firstly by imposing general constraints within the design space explorer that automatically discards points, followed by manually selecting the turbines and assessing geometries and velocity triangles. The design space explorer is used to set constraints to narrow down the selection process, and to compare designs that meet the turbine requirements. Filter limits are specified to automatically and rapidly remove unsatisfactory points; the imposed criteria is to discard any turbines that have $\dot{W} < 100$ MW and $\eta_{ts} < 80\%$. 90 out of the 193 solutions do not meet the imposed criteria and are removed from the selection process. The remaining solutions are individually selected for further assessment of the general geometric features and velocity triangles in order to ensure that the restrictions on tip speeds and flow angles are applied.

5.1.2 Preliminary Design Examples

For the sake of illustration of the process, two preliminary designs that meet the selected power and efficiency limits, but are rejected on the basis of other criteria are presented in this Subsection.

There are two main reasons behind choosing designs without a volute, the first being a simpler design choice and the second being that the nozzle blades of such configuration are narrow and long (example shown in Figure 5.2), meaning that there would likely be quite high blockage generation within the vane passages through the extensive thickening of boundary layers. The narrow shape of the vanes can be for reasons associated with reducing stress on the nozzle blades because of the combination of a highly accelerated flow in the volute with the very high inlet pressure (297 bar), thus smaller chord widths could reduce the load. Generally, a volute is used to deliver a uniformly distributed flow to the nozzle row and in this case, with the absence of an inlet scroll, an annular collector would be required to gather the inlet flow to the turbine. Any other designs (even without

a scroll) that have similar nozzle blade shapes are also discarded. As noted in Figure 2.6, radial turbines without a volute usually require a cambered nozzle blade design; this literature counsel is used as reference when choosing other appropriate designs without volutes.

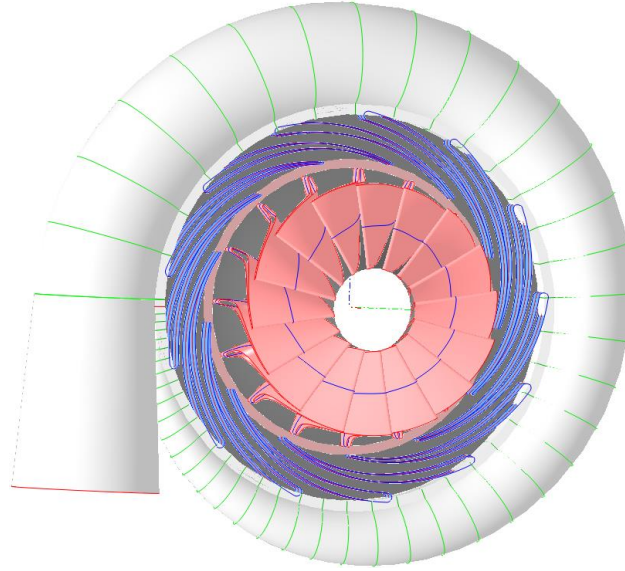


Fig. 5.2 3D model of discarded turbine design with volute

Some radial turbine solutions (without volute) which have meanline velocity triangles similar to that shown in Figure 5.3 are also omitted mainly because of the high circumferential velocity at rotor inlet ($U_4 > 450$ m/s) — which induces high stresses [125] — and a large outlet swirl component contributing to exit energy losses ($\alpha_5 \neq 0^\circ$).

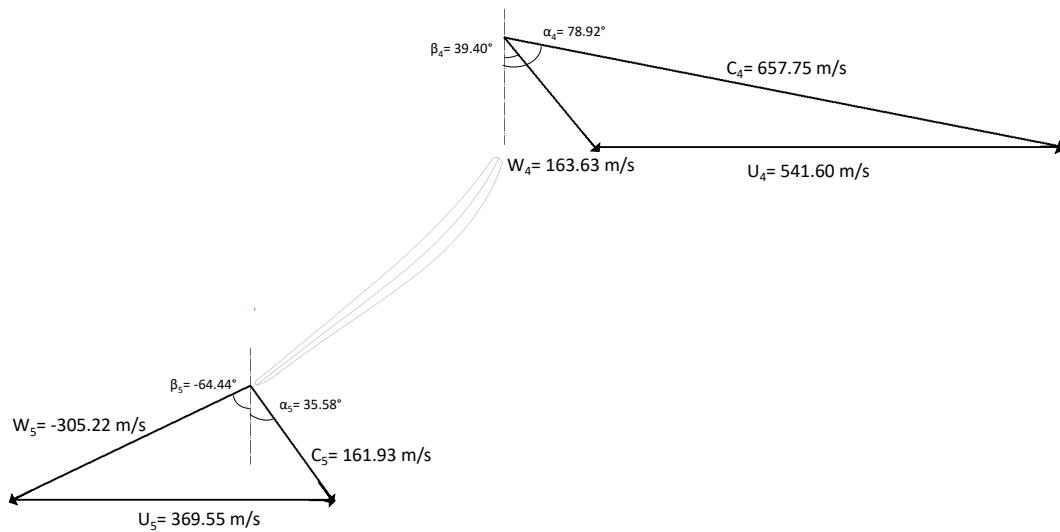


Fig. 5.3 Velocity triangle of a discarded turbine design

The preliminary design step is the simplest way to identify possible viable geometries at a very early stage before moving to higher fidelity analysis. A significant amount of time is spent filtering through results at this level but it is more time-efficient than iterating and discarding designs after further analysis. It offers a preliminary degree of scrutiny in order to enable the selection of more promising configurations.

5.2 Turbine Design and Analysis

This section outlines the comparison of performance parameters between the design intent as defined in the preliminary design phase and throughflow analysis results. Designs with satisfactory performance criteria, as defined by the throughflow analysis, are selected for further CFD study. Several designs that passed the preliminary design phase are dropped during this step for not meeting the performance requirements of mass flow, efficiency, pressure ratio or power. An example of the throughflow analysis of one of the design cases is presented below.

5.2.1 Case 1 Design

The 3D geometry of a suitable design case obtained from the PD step, labelled as Case 1, is shown in Figure 5.4 along with its dimensions displayed in Figure 5.5. The front-loaded profiles of the nozzle blades are adopted for better performance (in comparison with aft-loaded profiles [141]). The turbine dimensions are in good agreement with the estimated values obtained from the initial meanline design (Figure 4.5). The blade passages are slightly larger, by around 4 mm, but because of the absence of the volute, the overall size of the turbine is smaller compared to the estimated size from Subsection 4.2.2, with a maximum diameter of around 480 mm; the stator inlet radius though is larger to compensate for the missing flow acceleration from the volute.

Additional geometric parameters of the Case 1 turbine are found in Table 5.2; the blade metal angle (θ) is defined as the angle between the tangential reference plane and the blade camberline extension at leading edge for inlet, and trailing edge for outlet. In the case of no volute, the flow at vane inlet is assumed almost purely radial with straight radial blades at rotor inlet as well ($\theta_{in} \approx 90^\circ$). The rotational speed of the Case 1 turbine is 25,838 rpm and the rotor tip clearance is 1.046 mm.

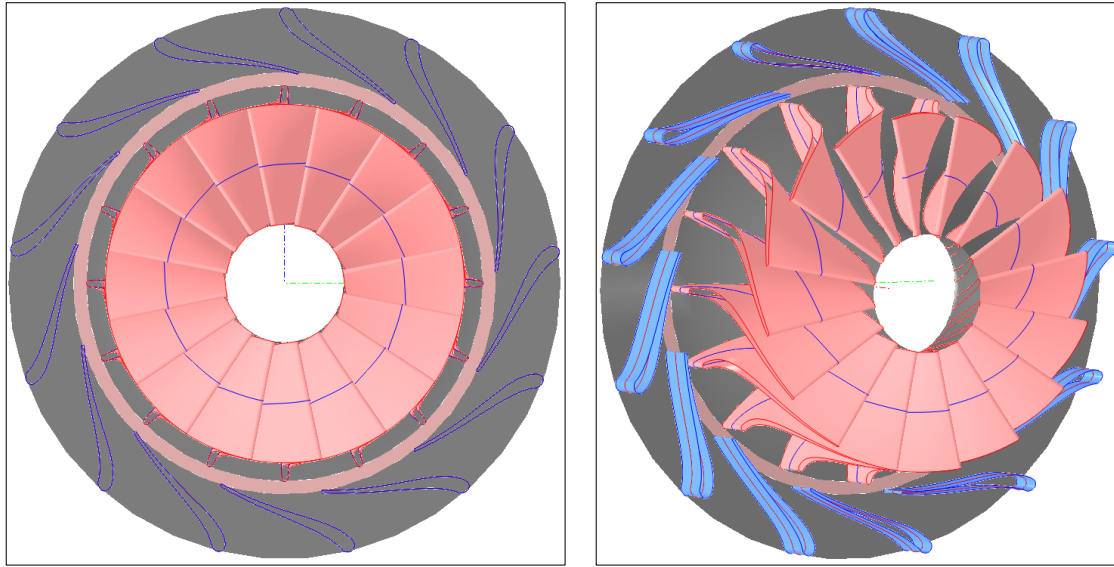


Fig. 5.4 3D model front view (left) and perspective view (right) of Case 1

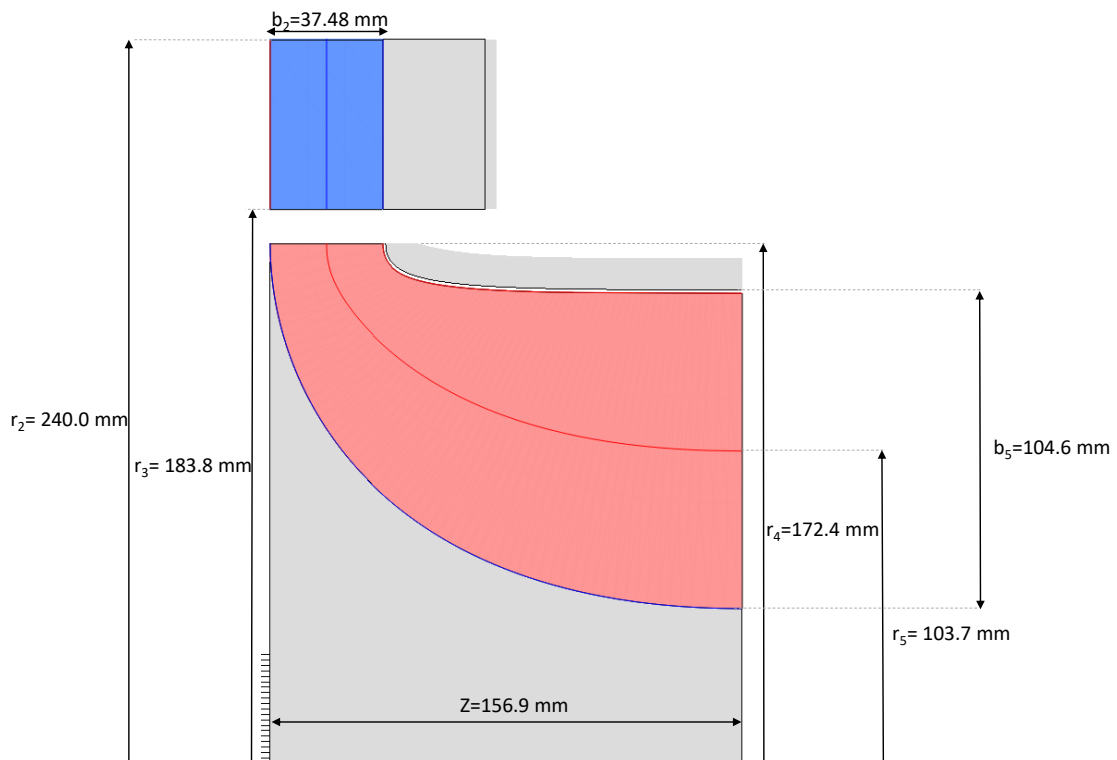


Fig. 5.5 Meridional dimensions of Case 1

Table 5.2 Geometric parameters of design case model

Stator			Rotor		
Variable	Unit	Value	Variable	Unit	Value
N_S	-	13	N_R	-	16
c_S	mm	112.3	c_R	mm	235.3
q_S	mm	88.83	q_{R_m}	mm	40.72
t_2	mm	11.37	t_4	mm	4.706
t_3	mm	2.274	t_5	mm	4.706
o_S	mm	18.94	o_R	mm	23.58
θ_2	°	98.69	θ_4	°	90
θ_3	°	12.31	θ_{5_m}	°	35.38

5.2.2 Case 1 Throughflow Analysis

Key performance results of the throughflow analysis using three levels of model refinement, 1 streamline (at mean span), 3 streamlines (at hub, mean and tip), and 49 streamlines (maximum amount of streamline curves in AxSTREAM), are listed in Table 5.3, against corresponding values obtained from the PD phase. There is little variation across all computed data irrespective of the level of refinement. The discrepancies between flow coefficient, degree of reaction and pressure ratio are attributed to small variations in velocity triangle results.

Table 5.3 Performance results comparison for different throughflow refinement levels for Case 1

Variable	Unit	PD	1 streamline	3 streamlines	49 streamlines
$\dot{m}$	[kg/s]	354.2	354.1	354.6	354.5
$\dot{W}$	[MW]	108.6	108.5	107.4	107.6
η_{tt}	[%]	89.54	89.56	88.51	88.63
η_{ts}	[%]	85.67	85.70	84.03	84.68
v_s	[-]	0.332	0.332	0.380	0.375
ψ	[-]	1.48	1.48	1.45	1.48
ϕ	[-]	0.376	0.376	0.418	0.380
R	[-]	0.231	0.224	0.252	0.221
Π_{ts}	[-]	5.39	5.39	5.47	5.40

The meridional distribution of entropy is visualised in Figure 5.6; although entropy generation across the turbine stage is captured in results from both the PD expectation and throughflow analysis using streamlines, the former presents a lower estimation of the entropy rise towards the rotor exit. Higher loss resolution from the throughflow calculations at increased sectioning is inherently a reason for the higher projected entropy especially

near the shroud curvature where deceleration occurs when the flow starts straightening in the meridional direction towards the rotor outlet; the flow accelerates in the axial plane and decelerates in the radial direction so there is a mixing of both velocity components.

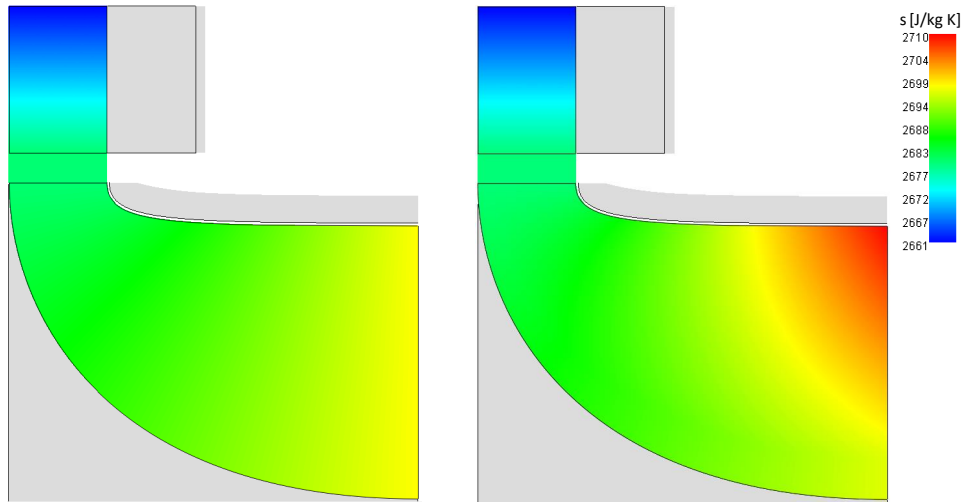


Fig. 5.6 Meridional entropy contours for Case 1 from: PD expectation (left) and 49 streamlines throughflow analysis (right)

Rotor velocity triangles and thermodynamic data for the Case 1 turbine, obtained after throughflow calculations with 49 streamlines, are found in Figure 5.7 and Table 5.4 respectively. There are some noticeable differences in the thermodynamic quantities at the turbine stations when using AxS Carbon Dioxide fluid (throughout the AxSTREAM software) compared to those from REFPROP (used in MATLAB meanline design code — refer to Table 4.9). Pressure and enthalpy values are very similar because they are the values used in defining the PD input but the temperature is between 50 – 60 degrees higher in AxSTREAM; a further examination is needed to ensure temperatures do not exceed material limits if the design is adopted. Although entropy values obtained from AxSTREAM are larger than REFPROP values, the entropy rise ($\Delta s = s_{out} - s_{in}$), which accounts for loss generation across the stage, is almost identical at approximately 40 J/kg K. The density variation is a consequence of the difference in temperature. An assumption used in the meanline design procedure is that the total thermodynamic quantities across stationary components remain constant. In AxSTREAM, the total pressure drop across the nozzle however, is accounted for due to the associated blade profile losses unlike the assumption of equal total pressure at stator inlet and outlet.

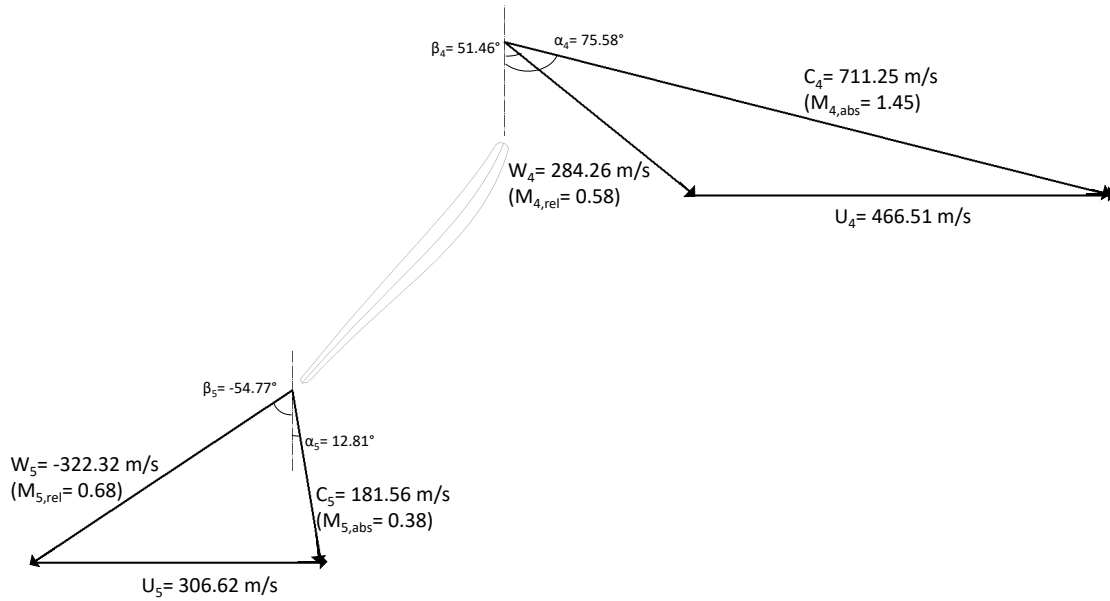


Fig. 5.7 Velocity triangles Case 1

Table 5.4 Thermodynamic properties data for Case 1

Variable	Unit	Nozzle inlet (2)	Nozzle outlet (3)	Rotor inlet (4)	Rotor outlet (5)
T_0	K	1234	1234	1234	996
P_0	bar	297	272	272	61.0
h_0	kJ/kg	1480	1480	1480	1179
s	J/kg K	2661	2679	2679	2701
T	K	1233	1065	1041	981
P	bar	295	105	90.5	55.1
h	kJ/kg	1479	1264	1234	1160
ρ	kg/m ³	118.1	50.79	45.10	29.44

The h-s/ Mollier diagram representing the expansion process across the turbine stage with irreversibility effects is shown in Figure 5.8. The process starts with station 2 representing the nozzle inlet because the design is without a volute (station 1 is reserved to scroll entry). Line 2 – 3 represents the actual expansion in the nozzle from P_2 to P_3 which is greater than the absolute pressure reduction taking place in the rotor from P_4 to P_5 correlating to the relatively low degree of reaction ($R = 0.221$). Nozzle and rotor component losses are equivalent to the outlet enthalpy difference between the isentropic/ theoretical enthalpies (denoted by the subscript s) and the static enthalpies whereas the exit energy loss is the difference between enthalpies at stations 5 and 05.

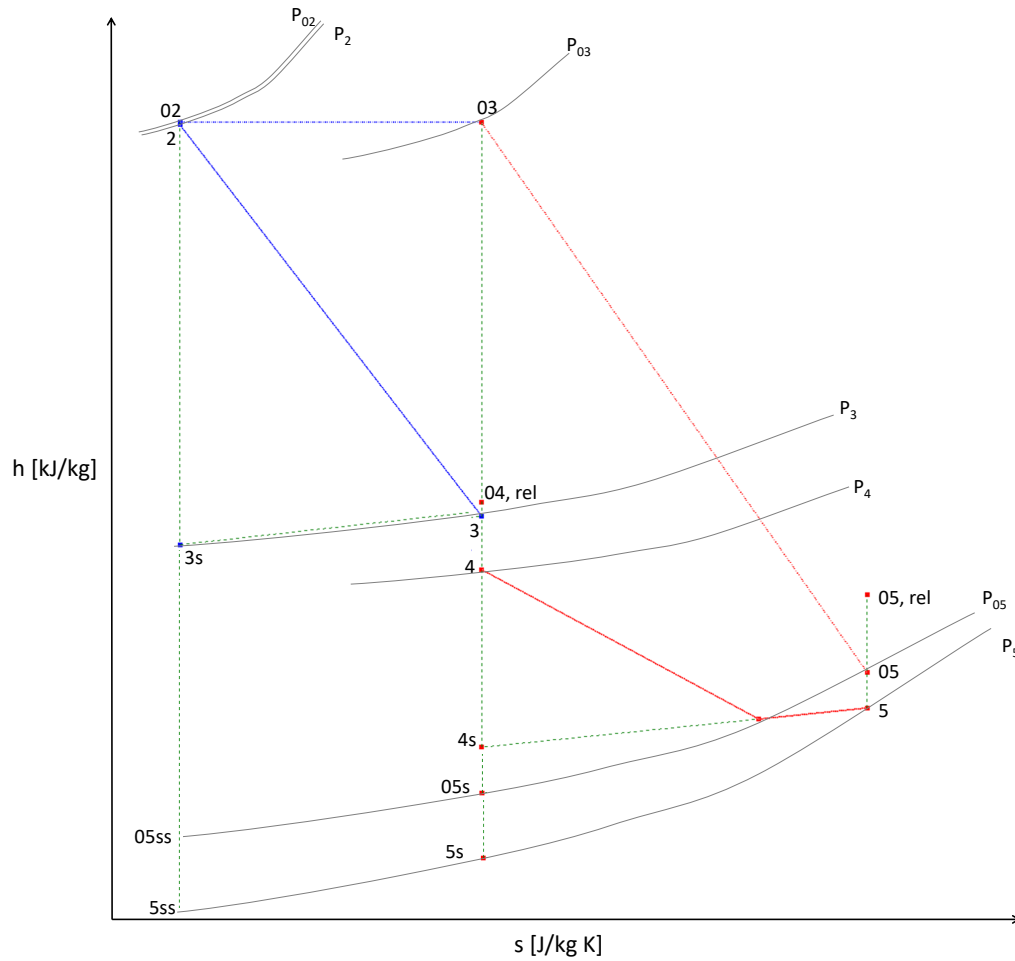


Fig. 5.8 Enthalpy- entropy diagram for Case 1

The axial and radial velocity distributions in the meridional plane of the Case 1 turbine are displayed in Figure 5.9. The visualisation shows the radial acceleration of the flow throughout the nozzle passage (negative radial velocity because the flow is opposite to the radius direction) up to the point of curvature turning of the rotor blade, where maximum radial velocity is reached, after which the flow starts losing its radial direction and approaches the rotor exit in the axial/ meridional plane with a small swirl component as represented in the outlet velocity triangle of Figure 5.7. The meridional flow distribution across the blade passages is representative of inward flow radial turbines in general.

The flow within the radial turbine, just like in any other turbomachinery component, has an unsteady and three-dimensional nature dominated by viscous effects, thus the use of 2D inviscid simplifications will neither fully account for the stator/ rotor interactions nor capture detailed zones of possible flow separation or shockwave effects. This is made clear in Figure 5.10 where the meridional distribution of relative Mach number (which is equal to the absolute Mach in the nozzle passage) is indicative of normal flow accelera-

tion/ deceleration across the stage with the change in radius and blade curvature. Within the same figure and context, the area between nozzle and rotor corresponding to Mach number greater than unity does not exhibit matching abrupt pressure fluctuations that are usually attributed to uncontrolled sonic compression followed by expansion. Pressure change in the vaneless annular passage is due to the abrupt area expansion loss applied to the meridional component of the velocity head. Better understanding of the flow field within the turbine is obtained through CFD techniques implemented in Section 5.3.

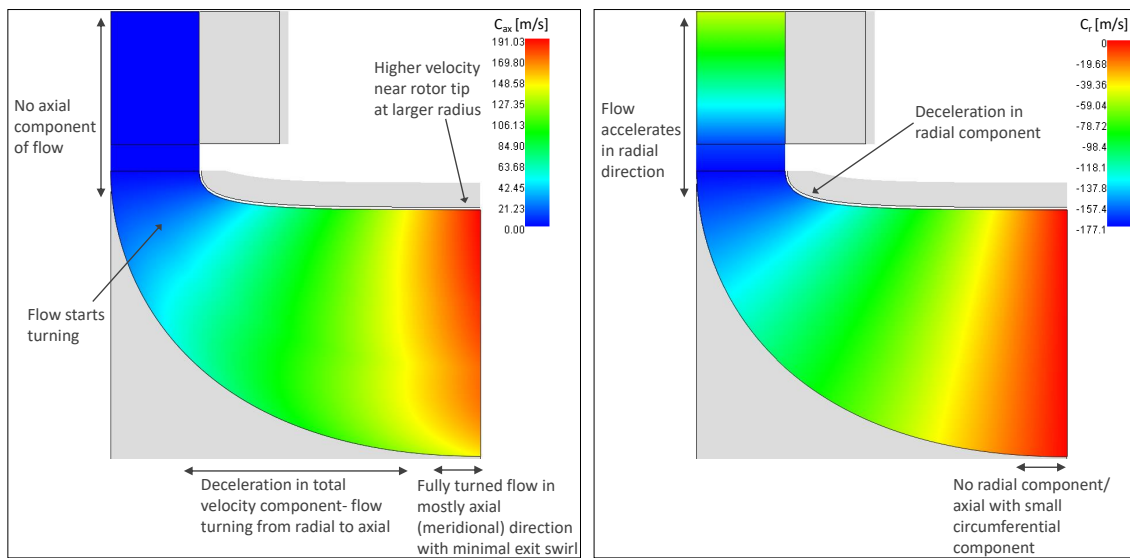


Fig. 5.9 Change in axial component of velocity (left) and radial component (right) in meridional plane for Case 1

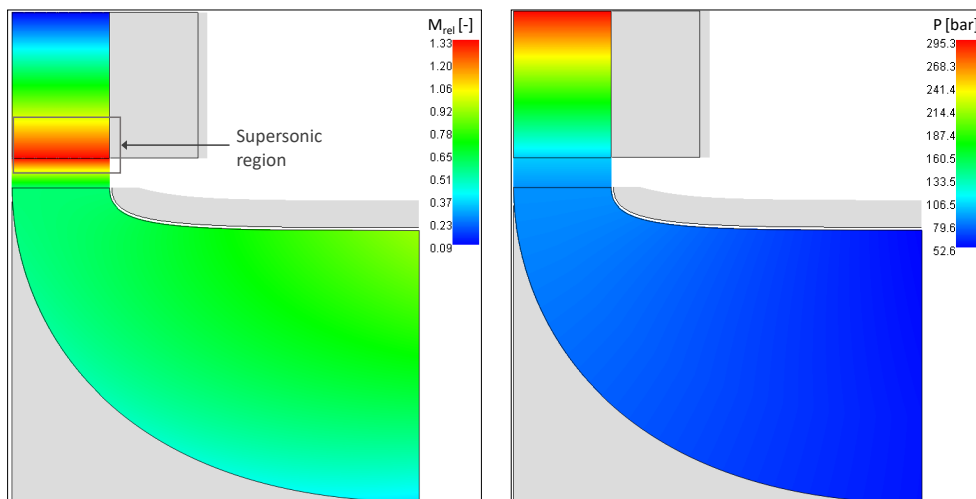


Fig. 5.10 Meridional distribution of Mach number (left) and static pressure (right) for Case 1

5.3 Computational Fluid Dynamics Analyses

As a means of getting an overall understanding of the flow behaviour in the turbine blade passages, a higher fidelity model of the stator and rotor passages is developed taking into account viscous effects. Comparisons are drawn based on results obtained from 1D meanline design and 2D throughflow analysis (presented in Table 5.3), and from CFD-based outcomes.

5.3.1 Axi-Symmetric Calculations

Before delving into the higher fidelity but more time-consuming 3D CFD simulations of the radial turbine, one can obtain a better understanding of the meridional flow and get results which represent the physics more than 1D/ 2D inviscid modelling by using the 2D axi-symmetric CFD-based throughflow code in AxCFD which incurs faster convergence.

Computational Mesh

Discretization of the single passage computational domain is done in both the stream-wise and inter-blade directions. A mesh sensitivity study is performed to examine the relationship between computational time, which is related to total nodal count, and convergence of results. Mesh refinement is carried on until results reach an asymptotic limit of negligible change in important parameters.

Spanwise boundary conditions at the blade mean span are taken from the main project database of AxSTREAM after the 2D analysis results are saved. The boundary conditions are mass flow rate= 354.5 kg/s, inlet total enthalpy= 1480 kJ/kg, and outlet static pressure= 55.0 bar. The variation of the output results of power and total inlet pressure for different mesh refinements is noted in Table 5.5, along with the time that the solution takes to reach convergence, and the percent deviation of the total inlet pressure (based on the value obtained from the finest mesh). Between the coarsest mesh and the finest mesh there is much disparity in power values which indicate efficiency differences as well; almost 11% change in power which coincides with total-to-total/ total-to-static efficiencies discrepancy observation.

Table 5.5 Performance results for axi-symmetric mesh sensitivity study of Case 1

Number of elements [-]	$\dot{W}$ [MW]	P_{0in} [bar]	Time [min]	P_{0in} Deviation [%]
756	83.61	236	7.56	5.01
2,376	89.07	241	13.0	2.95
9,840	92.49	245	25.9	1.41
19,044	93.07	246	40.7	0.86
36,554	93.77	248	78.7	0.38
70,560	94.38	248	128	0

As mesh density grows, results begin to approach an unchanging limit but at the cost of longer calculation times. This is observed in Figure 5.11 where a quasi-mesh-independent region can be defined when the P_{0in} deviation becomes less than 1.5% and the power values stabilise. Convergence takes more than triple the computational time between meshes with 19,044 elements to 70,560 elements for a negligible change in results.

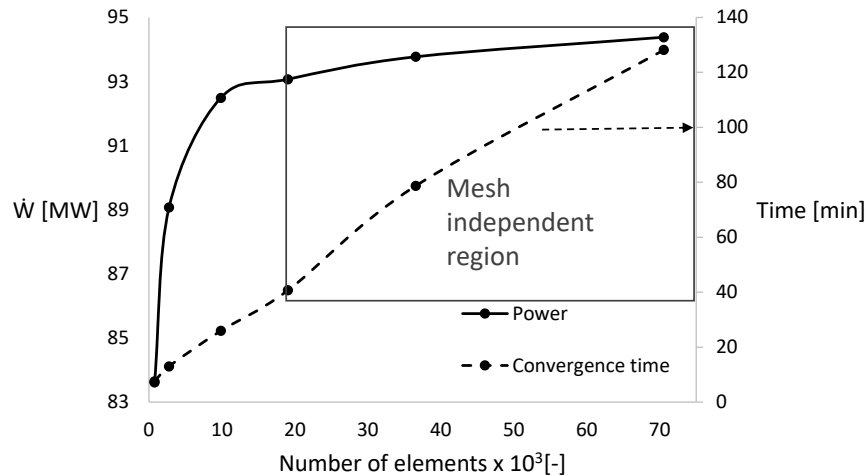


Fig. 5.11 Axi-symmetric mesh independence graph for Case 1

The coarsest mesh out of the accepted cell density is used for the purposes of this section which is the mesh with 19,044 elements; the mesh with its stationary and rotating domains is displayed in Figure 5.12. In a similar work, performed by Ricci et al. [142] on 2D CFD analysis for an axial turbine, they used approximately 18,000 elements for a single blade passage.

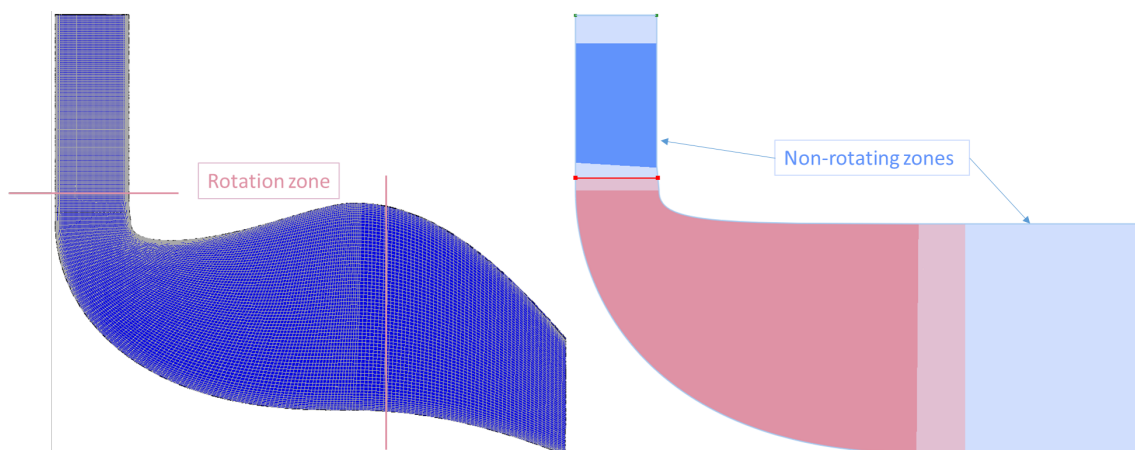


Fig. 5.12 Axi-symmetric mesh for Case 1

It is worth noticing that for all the meshes employed, the total inlet pressure value is significantly lower than what is expected from the turbine design. The difference between the nearest value of 248 bar and the required total pressure from the cycle of 297 bar is over 16%. A small variation is usually expected between design expectation and results obtained from further analyses, but a 16% disagreement between them is an extreme value to be accepted for the generated design.

Boundary Conditions Definition

An alternative boundary condition definition of $P_{0_{in}} = 297$ bar (instead of specifying $\dot{m}$) is imposed in the axi-symmetric model to check whether there would be similar differences in the calculated mass flow. The required mass flow value to achieve the specified total-to-static pressure drop is 21% higher than the design flow rate. The power value obtained from the $P_{0_{in}} / P_{out}$ specification is higher than that calculated from the other boundary conditions because of the associated higher flow velocities at an increased mass flow rate. The data presented in Table 5.6 provide an indication that the geometric design of Case 1 is flawed for the purposes of the required turbine.

Table 5.6 Performance results for different boundary conditions applied to 2D CFD of Case 1

Boundary condition	$\dot{m}$ [kg/s]	$P_{0_{in}}$ [bar]	P_{out} [bar]	$\dot{W}$ [MW]	η_{tt} [%]	η_{ts} [%]
$\dot{m}_{in}$ & P_{out}	354.8	246	55.0	93.07	88.00	83.16
$P_{0_{in}}$ & P_{out}	430.7	297	55.0	121.6	85.67	80.01

Despite the turbine design being not fit for the operating conditions environment, which is dictated by the cycle, it is worth exploring the flow fields within the passages as a means of insight into the bulk flow behaviour of sCO₂ fluids in radial turbines. Meridional distribution of Mach number in the relative frame of reference is pictured in Figure 5.13 for the two boundary condition definitions. In both cases, the bulk flow behaves similar in terms of acceleration and deceleration but with some differences in areas of low momentum. Figure 5.13 shows a representation of the M_{rel} distribution where in comparison to the display of Figure 5.10, obtained from the streamline curvature analysis, the more realistic non-uniformity of the flow is apparent. As indicated by the power outputs of each case (from Table 5.6), the stator outlet Mach number is lower for the case with the plot on the left because of the lower attributed pressure drop. Besides that, the examination of flow behaviour shows a rapid deceleration upon rotor entry near the hub because of the sudden change in available area and lower circumferential velocities there, plus a small region of flow separation ($\lim_{M_{rel} \rightarrow 0}$) right after the major rotor geometry curvature change.

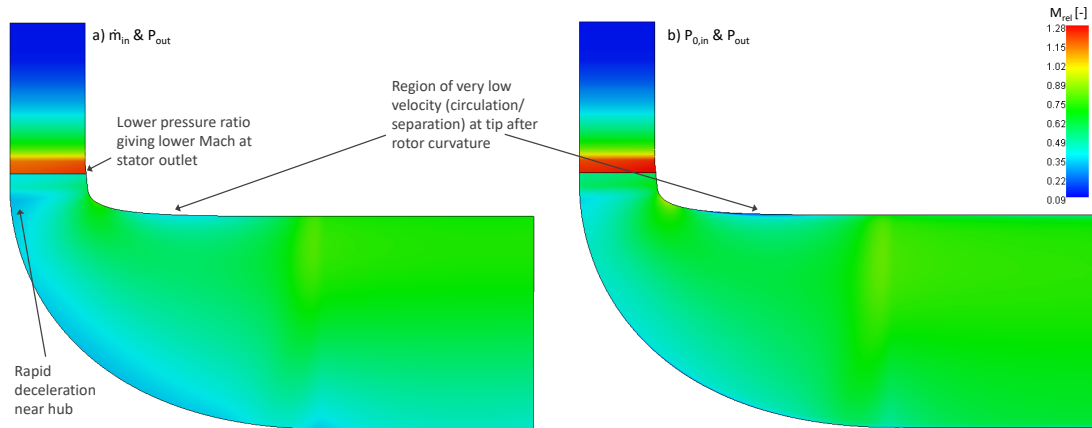


Fig. 5.13 Relative Mach number meridional distribution for different axis-symmetric boundary condition definition for Case 1: $\dot{m}_{in}$ & P_{out} (left) and $P_{0,in}$ & P_{out} (right)

5.3.2 3D CFD Results

3D simulations are carried out for one blade passage of the Case 1 turbine to form a basis of acquiring comprehension of the turbine flow which can be used for comparison purposes with other designs, given that there is no available validation data for the turbine designs under examination. For the steady compressible simulations using the $k-\omega$ SST, a high-resolution discretization method is required for the advection term to obtain low values of y^+ . It is important to note here that although resolving the viscous boundary layer region is highly important for better capture of the associated losses, the main focus of this study is to analyse the bulk flow behaviour rather than fully accounting for the critical boundary layer interactions.

3D Mesh Generation

Similarly to the mesh sensitivity study of the axis-symmetric simulations, the turbine performance parameters are checked against varying mesh densities for the three-dimensional analysis performed in this section. Correspondingly to the 2D CFD modelling, two types of boundary condition combinations are imposed and the results are listed in Table 5.7. As indicated in red, values of total inlet pressure and mass flow vary trivially as mesh size is increased; the largest observed changes refer to the efficiency values which can be imputed to improved wall boundary layer resolutions as number of elements increase. When compared to the outcomes obtained from axis-symmetric modelling, the 3D results show that there is even a higher disparity against design requirements; a 25% lower $P_{0,in}$ value for the specified mass flow, or 34% higher $\dot{m}$ for the pressure drop is needed to match the initial design intent.

Table 5.7 Performance results for 3D CFD mesh sensitivity study of Case 1 using two different boundary condition definitions

Boundary condition: $\dot{m}_{in}$ & P_{out}					
Number of elements [-]	P_{0in} [bar]	$\dot{W}$ [MW]	η_{tt} [%]	η_{ts} [%]	Δs [J/kg K]
49,960	221	67.56	73.83	67.43	68.58
201,272	222	73.93	79.01	73.04	55.62
759,528	222	78.87	83.71	76.80	45.08
962,874	222	78.89	82.68	76.78	47.73
Boundary condition: P_{0in} & P_{out}					
Number of elements [-]	$\dot{m}$ [kg/s]	$\dot{W}$ [MW]	η_{tt} [%]	η_{ts} [%]	Δs [J/kg K]
49,960	482.9	106.1	72.39	65.40	89.13
201,272	479.4	116.7	77.44	70.84	77.98
962,874	474.2	121.8	81.04	74.88	73.55

The finest mesh of 962,874 elements is used for flow visualisation. The y^+ on the rotor surfaces is ≤ 2 ; it is harder to obtain low values of the non-dimensional distance on the stator and rotor shroud because of the high Reynolds number ($Re \approx 1.7 \times 10^7$, given that the fluid has a high density). The range of y^+ values on all surfaces is between 0.14 – 200 (200 being the highest acceptable limit of y^+ as referred to SoftInWay manual recommendations). Cerdoun et al. [143] also noted high values of y^+ in specific regions of the CFD mesh (0.3 – 92.2).

Lower y^+ can be obtained by further increasing mesh element count but owing to the licensing restrictions of AxCFD and the limited computational resources available to run the simulations, the mesh of 962,874 cells, which requires more than a day-worth of computation time for convergence, is employed and is displayed in Figure 5.14. H-O grid topologies are used in both the stator and rotor domains for smooth alignment with the flow direction, and refinement is made in regions near blade walls and leading/ trailing edges to account for high normal gradients [144] as discussed in Subsection 3.5.1. Other researchers have used coarser meshes with around 500,000 nodes and were capable of obtaining reasonable solutions based on their requirements because of different constraints [96].

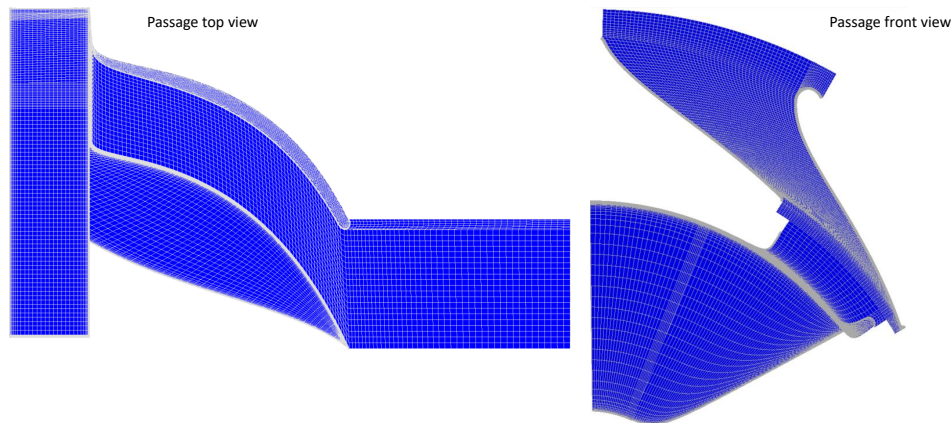


Fig. 5.14 3D blade passage H-O mesh for Case 1

Convergence

Inlet/ outlet mass imbalance, monitoring of total-to-total efficiency stabilisation, and root mean square (RMS) residuals of mass, density, pressure and turbulence equations (k and ω) are used as criteria to assess solution convergence. Figure 5.15 shows residual plots for the time-marching solution starting from an initial guess against number of iterations. For the lower mesh densities (indicated by quality 3 – 5 – 7), the residual values are $\leq 10^{-6}$ but for finer meshes, the values take longer to reach steady higher values. This is observed in similar works by Wei [96] where increased residual values are accepted for off-design conditions and for finer mesh simulations. Mass imbalance is also a determining factor for convergence and the percentage difference between inlet and outlet mass flow is limited to 2% (lower for coarser meshes). The formulation uses the first order upwind scheme given its stability, and due to the fact that although second order schemes are known to be more accurate, the solution in this case becomes unstable and very difficult converge when higher order methods are employed.

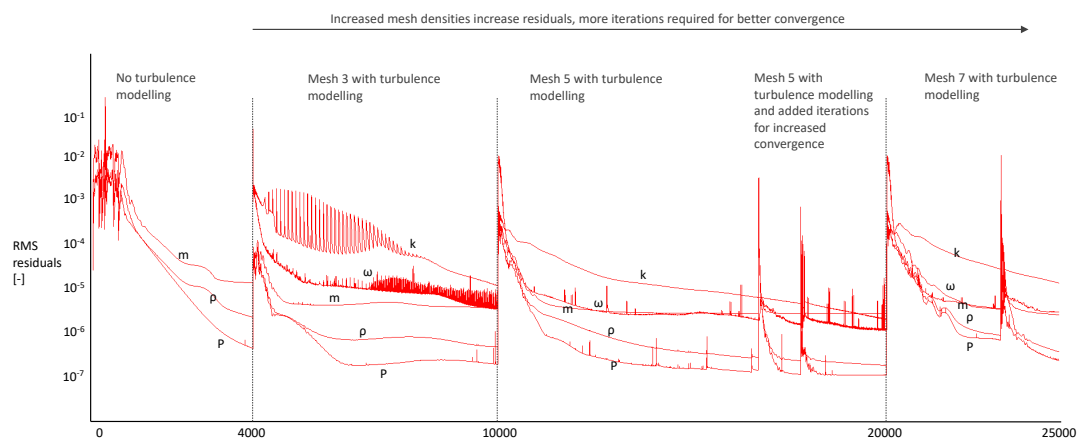


Fig. 5.15 Convergence plots for 3D CFD simulations of Case 1

Flow Analysis

Design point 3D CFD results for the Case 1 turbine using boundary conditions of total pressure at inlet, and static pressure at outlet are discussed next. Figure 5.16 is used as a means of visualisation of the spanwise and streamwise turbine passage sections that are introduced hereafter. Sections along three different locations at 10%, 50% (mid-span) and 90% of span are used for contour displays in blade-to-blade views, along with two hub-to-tip planes, one after the main curvature of the rotor, and the other at rotor exit.

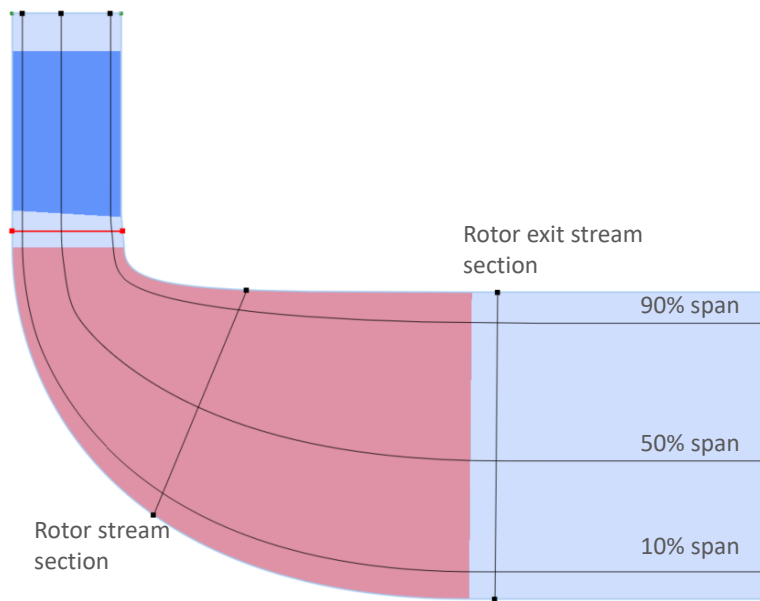


Fig. 5.16 Visualisation of blade-to-blade span-wise planes and hub-to-tip cross-sections for Case 1

Figure 5.17 shows relative Mach number distribution (absolute for stator) at three locations across the span. At the throat of the stator, the flow reaches its supersonic conditions with the onset of the trailing edge shock; high losses are associated with shockwaves which are mitigated by changing airfoil shape to lower Mach number values. Low Mach number values near the pressure side (PS) leading edge indicate a loss region because of unsuitable flow incidence originating from the stator. The acceleration of the flow near the leading edge of the rotor suction side (SS) also incurs more turbulent flow. Noting that rotor inlet relative flow angle (β_4) at mean span is 20.51° compared to the equivalent 51.46° which is observed in Figure 5.7.

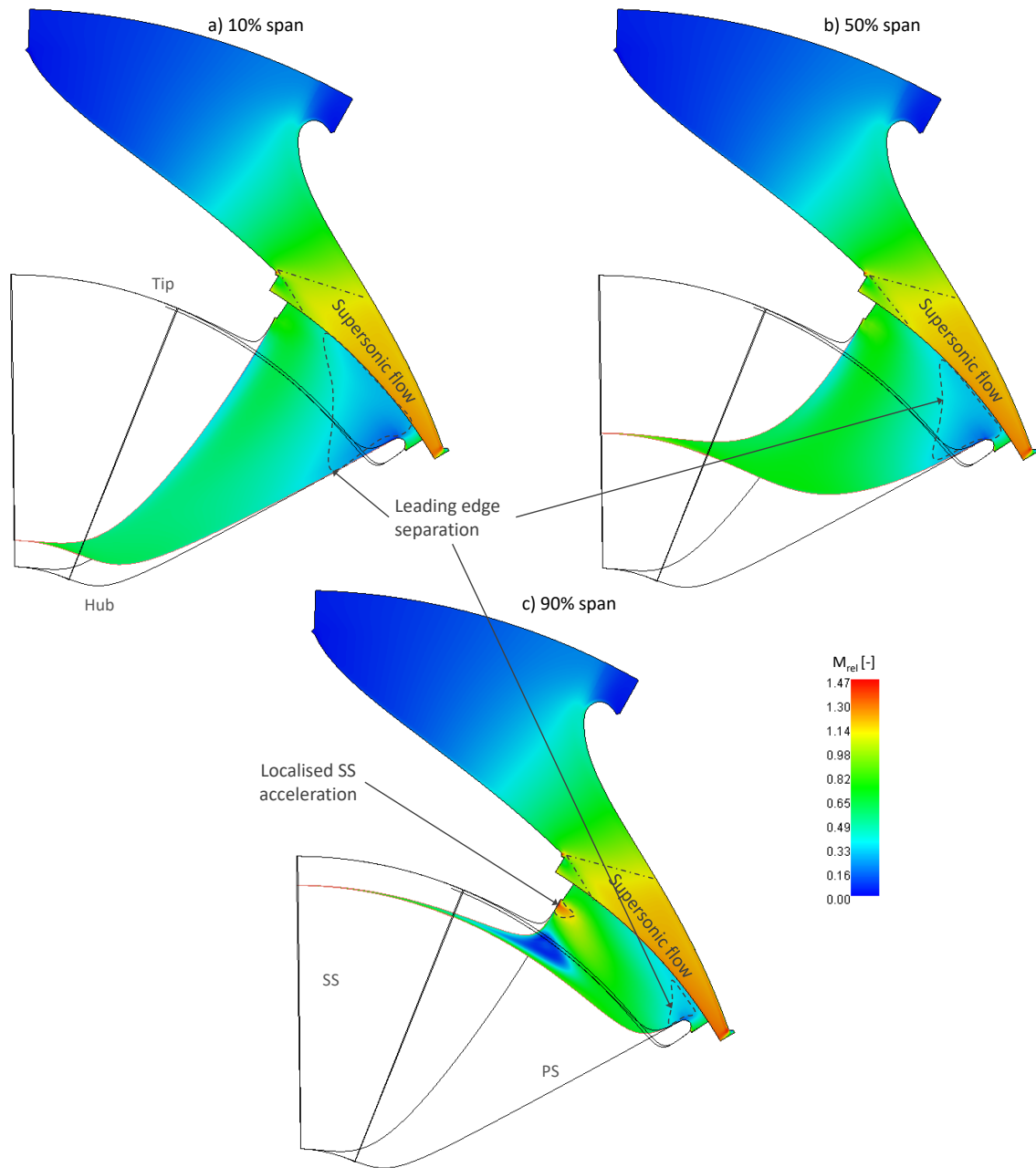


Fig. 5.17 Relative Mach number distribution at three spanwise turbine passage locations for Case 1

Figures 5.18 and 5.19 can be viewed in conjunction to further investigate loss zones in the radial turbine rotor passage of Case 1. This blade-to-blade view shows more detail regarding the flow characteristics on the suction and pressure surfaces of the rotor blade, and omits details of stator and rotor leading edge flows. Boundary layer separation is observed towards the end of the span just after the rotor blade angle experiences a major

deflection on the suction side; the separation zone in Figure 5.18 corresponds to the high entropy generation area in Figure 5.19. Smaller regions of entropy rise are noticed near the leading edge because of the reattachment of the leading edge separation (because of incorrect incidence) on the pressure side with the higher velocity flow on the suction side. Rotor wakes, produced by the velocity differential between the viscous layers at trailing edge from the suction and pressure sides, also correspond to higher irreversibility especially at mid-span.

Similar flow behaviour patterns have been remarked in works on radial turbines and turbochargers performed by other researchers. Wei [96] reported separation on the blade suction side for a supercritical CO₂ radial turbine, and Rahbar et al. [121] outlined leading edge separation on the pressure side depending on rotor blade design and turning.

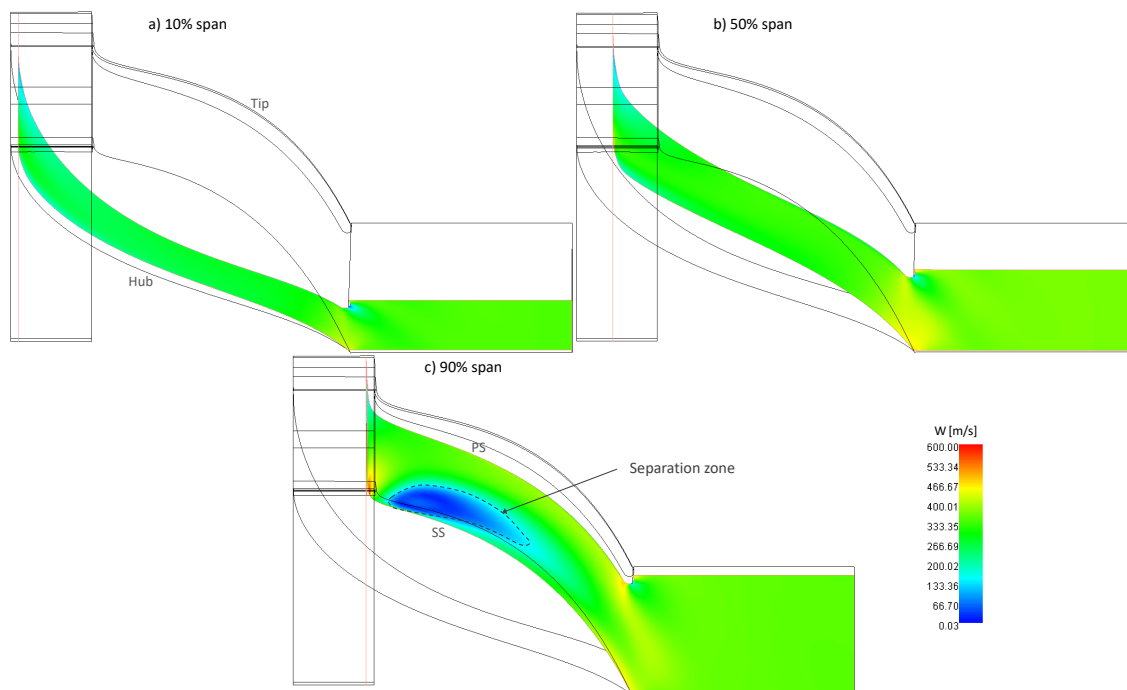


Fig. 5.18 Relative velocity distribution at three spanwise rotor passage locations for Case 1

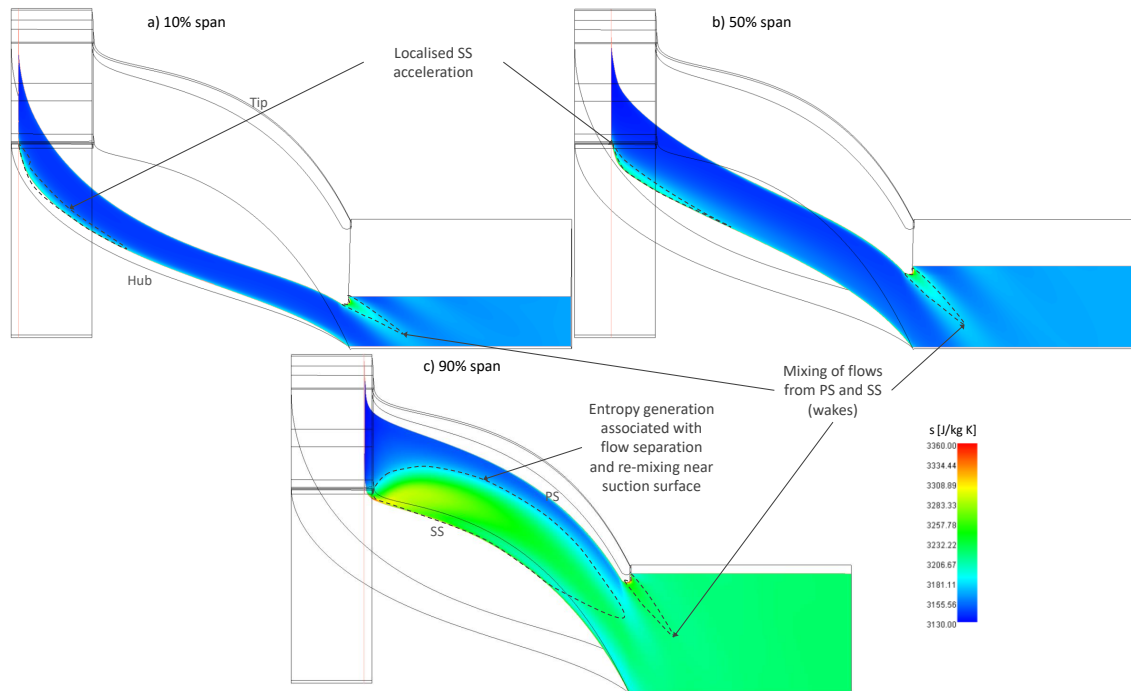


Fig. 5.19 Entropy contours at three spanwise rotor passage locations for Case 1

Relative velocity contours on a hub-to-tip cross-section just after the rotor blade turning are shown in Figure 5.20. With a rotor shroud clearance of 1.046 mm the flow migrates from the high pressure side to the lower pressure suction side inducing an acceleration in that direction and a creation of a tip-gap vortex. On the pressure side, the moving of the low-inertia flow from hub to tip interacts with the tip leakage flow to create an area of high velocity. Near the tip of the suction side, the area of low relative speed coincides with the area of flow separation shown in image c) from Figure 5.18.

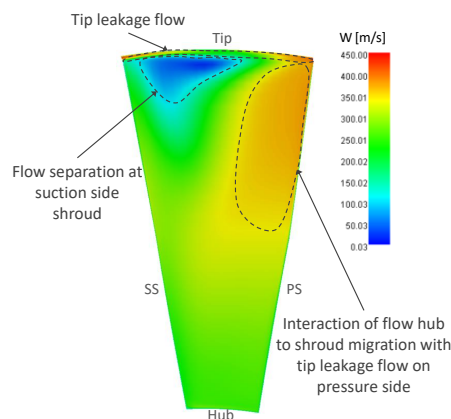


Fig. 5.20 Relative velocity contours at rotor stream section for Case 1

An additional plane at rotor exit with contours of total pressure in the rotating frame of reference is plotted in Figure 5.21. The warm-coloured patches (regions tending to red) are areas with the highest total pressure whilst cooler-coloured tones (tending to blue) are zones where total pressure is lost. The bluish-green zone near the hub may correspond to the pressure loss arising due to the 3D endwall flow combining with hub separation as described by Akturk et al. [145]. Tip leakage flows also add to the pressure losses associated with the reduction of the available potential work extraction.

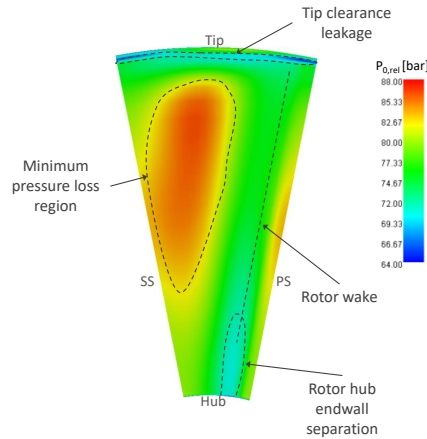


Fig. 5.21 Relative total pressure contours at rotor exit stream section for Case 1

5.3.3 Modifications and Off-Design Conditions

A number of minor adjustments on the Case 1 analysis are made to check whether this could bring the results closer to the design requirements. Varying rotational speed and pressure ratios is similar to performing off-design analysis. For those, calculations are carried out in 2D CFD rather than the time-consuming 3D; if any alteration shows improvements, then it would be investigated further in 3D modelling.

Figure 5.22 represents the variation of mass flow and total-to-static efficiency with the change in pressure ratio and three off-design rotational speeds ($\pm 10\%$ design speed) of 23,254 rpm, 25,838 rpm and 28,422 rpm. Knowing that the design is choked at stator outlet ($M \geq 1$), a pressure ratio larger than around 4 (choking $\Pi_{ts} \approx 4$) does not incur a higher mass flow rate for any of the rotational speeds. With smaller pressure ratios, lower mass flows are required for the faster running turbines; however, to get mass flows in the range of the target value of 355 kg/s, the pressure ratio required is much lower than the operating Π_{ts} of the modified NET Power cycle. It seems therefore that for any off-design point that could be considered, the turbine will be working at choke conditions. The turbine approaches its maximum efficiency just before it reaches choking conditions because of the associated high velocities (higher work extraction from Euler equation – Eq. 3.56) without experiencing shockwave losses. In general higher running speeds are associated with more efficient turbines.

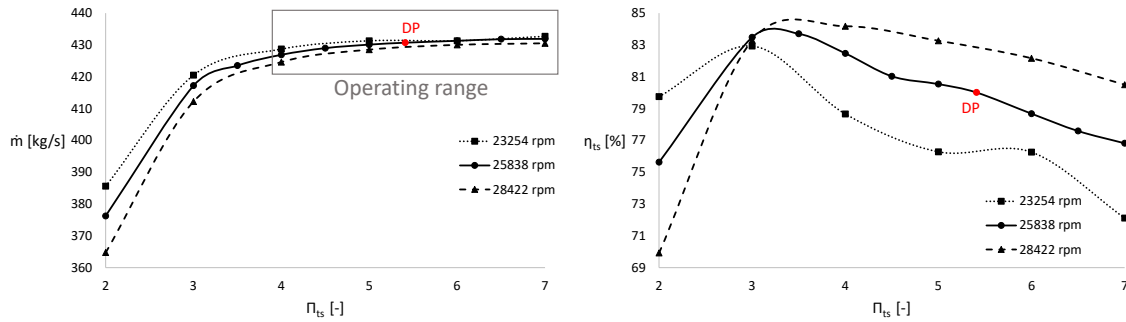


Fig. 5.22 Variation of mass flow and total-to-static efficiency with total-to-static pressure ratio at three rotational speeds for Case 1

A more thorough explanation of off-design analysis is presented in Subsection 6.3.3 of the following chapter, however, for comparative and validation purposes the study is also carried out for the Case 1 turbine. Figure 5.23 illustrates the off-design performance of Case 1. The meridional section on the left-hand-side (a), corresponds to conditions giving the best efficiency with reference to Figure 5.22, and that is the case because there is not a large fluctuation of relative velocity within the passage, and because the flow throughout is fully subsonic, then there are no associated shockwave losses. On the other hand, the section to the right-hand-side (b), corresponding to worst efficiency conditions of -10% rpm and total-to-static pressure ratio of 7, shows areas of abrupt flow velocity change (vortices, separation) and two regions of supersonic conditions, one at stator outlet and one at rotor outlet because of the much reduced pressure at exit.

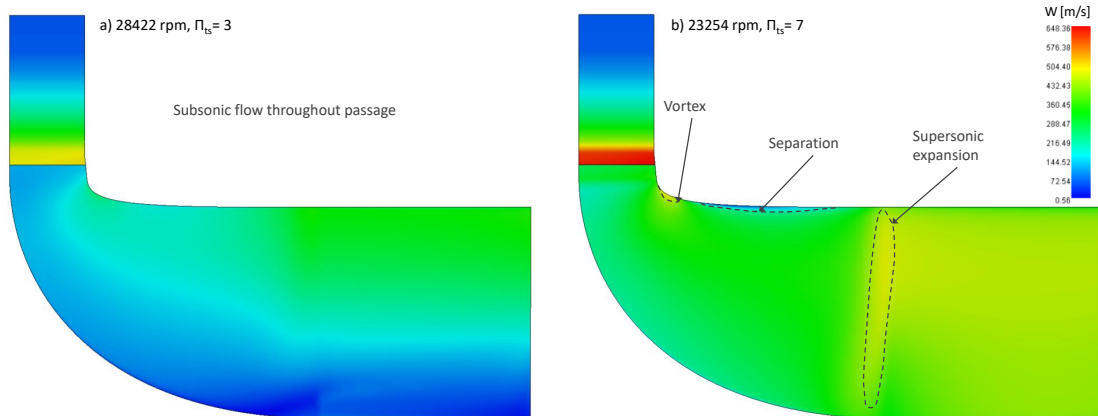


Fig. 5.23 Off-design relative velocity meridional distribution from axi-symmetric simulations of Case 1: 28,422 rpm , $\Pi_{ts}=3$ (left) and 23,254 rpm , $\Pi_{ts}=7$ (right)

Other changes are tested through modifying the number of stator and/ or rotor blades to reduce blockage and by modelling the rotor passage alone with varying incidence angles. However, these changes have a very minor impact on the results. Hence, it is

possible to conclude that the Case 1 design does not meet the design intent on a number of grounds and therefore a redesign exercise is called for taking into advantage the lessons learnt from the initial design investigation.

Chapter Summary

This chapter covers the design and analysis stages of a radial turbine. From the wide range of solutions obtained from the preliminary design step, target criteria of satisfactory geometries are stated. A design with an inlet diameter of 480 mm operating at 25,838 rpm is selected with more information presented. Further analysis is carried out using through-flow analysis which shows that the design seems satisfactory. However, 2D and 3D CFD results are not in full agreement with the design intent with almost a 16% and 21% difference compared to the preliminary design data respectively. The selected Case 1 turbine is deemed inadequate for the required design because it does not achieve the operating conditions to which it is being developed for; the input to the preliminary design tool is unsuitable and requires change. The steep rotor curvature is identified to be responsible for the generation of extreme flow separation; this could have been picked on during the preliminary design stage, however the study was still conducted for learning purposes. Nonetheless important flow behaviour traits are gained for comparison with other generated designs, and an insight to the major loss components within the radial turbine passage is given.

CHAPTER 6

Re-Examined Turbine Design

This chapter follows the same procedure of the turbine design and analysis methods employed in the previous chapter albeit with new generated results. Based on the conclusions drawn from the initial turbine design, an updated preliminary design space is produced. From that design space, an alternative turbine design is selected for further analysis using throughflow calculations, 2D and 3D CFD modelling as formerly performed. In order to obtain a fuller insight into the re-examined turbine design, a more extensive analysis of the flow is carried out along with the simulation of off-design conditions. Finally, static stress examination is undertaken in order to explore the mechanical soundness of the new design.

6.1 Updated Preliminary Design Space

Based on the results obtained from Chapter 5, a number of turbine design modifications are pointed out. As was noticed in the analysis of the previous design, a region of flow separation could be identified in the location corresponding to the change of direction from the radial to the axial plane within the rotor passage; together with the fact that the turbine dimensions did not allow for the required mass flow to pass through or for the necessary pressure drop to take place. Therefore, a widened set of geometric ranges, still rooted on the initial values obtained from the meanline design of Section 4.2, are used to generate a new preliminary design space to select more designs. The reason for this move being that other studied designs from the preliminary design space presented in Section 5.1 did not lead to satisfactory outcomes.

The updated data set corresponding to this widening of the geometric range is listed in Table 6.1. The turbine boundary conditions retain the same values as before in order

to satisfy the requirements imposed by the cycle model (Section 4.1) with the change being in the definition of the thermal boundary condition as the total inlet temperature in contrast to the total inlet enthalpy defined in Table 5.1. In order to compensate for the newly defined increased turbine radii range, a simultaneous broader and lower span of rotational speed is designated to maintain maximum rotor tip speed values. The scope of both the stator and rotor diameters is increased to allow for a more smooth turning of blade angles, while all other variable ranges are left unaltered.

Table 6.1 Updated input data for preliminary turbine design

Variable	Unit	Value
Boundary conditions		
Inlet total pressure	bar	297
Inlet total temperature	K	1173
Outlet total pressure	bar	60
Mass flow rate	kg/s	354.2
Inlet rotor metal angle	°	90
Shaft rotational speed	rpm	17000 – 24000
Geometric parameters and constraints		
Stator outlet mean diameter	m	0.4 – 0.6
Rotor blade height ratio	-	2 – 3
Rotor diameter ratio	-	1.5 – 3
Rotor inlet height	m	0.01 – 0.04
Hub reaction	-	0.3 – 0.5
Stator inlet diameter	m	0.4 – 0.8
Inlet flow angle	°	1-179
Rotor outlet hub diameter	m	0.05 – 0.3

The updated AxSTREAM preliminary design space corresponding to the input parameters of Table 6.1 is displayed in Figure 6.1 with 130 possible designs. In comparison to the preliminary design space of Figure 5.1, similar trends of higher efficiency turbines are obtained, with the more efficient turbines clustered near the larger velocity ratios. A greater number of turbines with efficiencies towards the lower end of the design spectrum are obtained in this case, which goes in hand with the lower power range reaching a maximum of 108 MW when compared to the 120 MW peak of the initial preliminary design space. Therefore, it is expected that the newly selected turbines would have a lower power output at smaller efficiencies. Similar constraints of $\dot{W} < 100$ MW and $\eta_{ts} < 80\%$ are applied to the design space explorer to allow for an easier selection of designs. Individual turbine design selection is also performed with the same consideration of tip speed values and velocity triangle configurations.

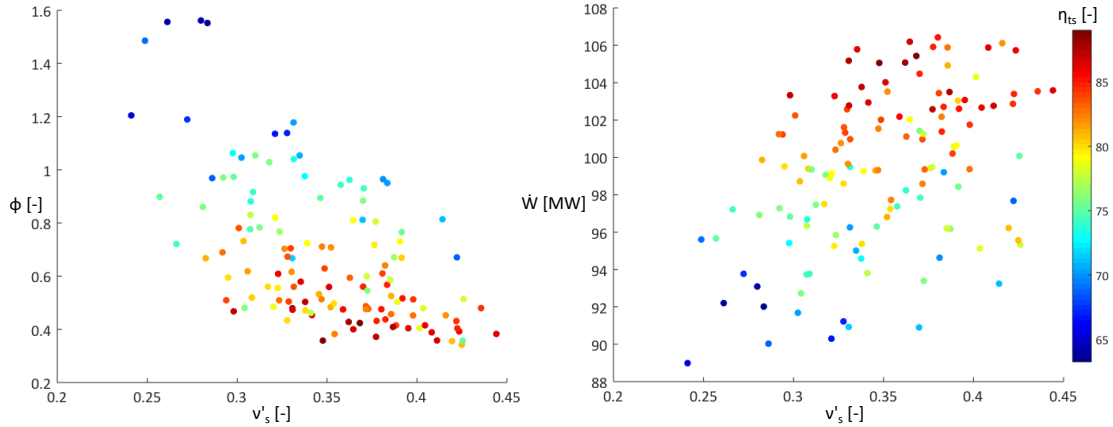


Fig. 6.1 Updated preliminary design space: flow coefficient vs velocity ratio (left) power vs velocity ratio (right)

6.2 Revisited Turbine Design and Analysis

Only one turbine from the design space is presented and analysed in this section. Due to the wider range of rotational speeds and inlet radii, a large number of designs are discarded because of the extremely high rotor blade tip speed velocities. Other selected designs from the new preliminary design space were examined using throughflow analysis with a few using CFD modelling, and failed to meet the assigned acceptable performance criteria and/ or the cycle operating conditions.

6.2.1 Case 2 Design

From the few selected designs, one particular design (Case 2) that meets the defined criteria, and which is characterised by a smooth rotor turning from the radial to the axial direction is presented in Figure 6.2, with the meridional dimensions and geometric parameters given in Figure 6.3 and in Table 6.2 respectively. The shaft rotational speed is 21,409 rpm and the rotor tip clearance is 0.782 mm. In relation to the geometry of Case 1 (Table 5.2/ Figure 5.5), the stator blade count has an additional three blades together with a slightly increased inlet radius, meaning that the throat distance (o_s) is smaller which might induce higher Mach numbers at stator outlet. As a consequence the rotor has a somewhat larger radial space for the radial to axial turning to occur, although axially it is shorter than the datum geometry, Case 1.

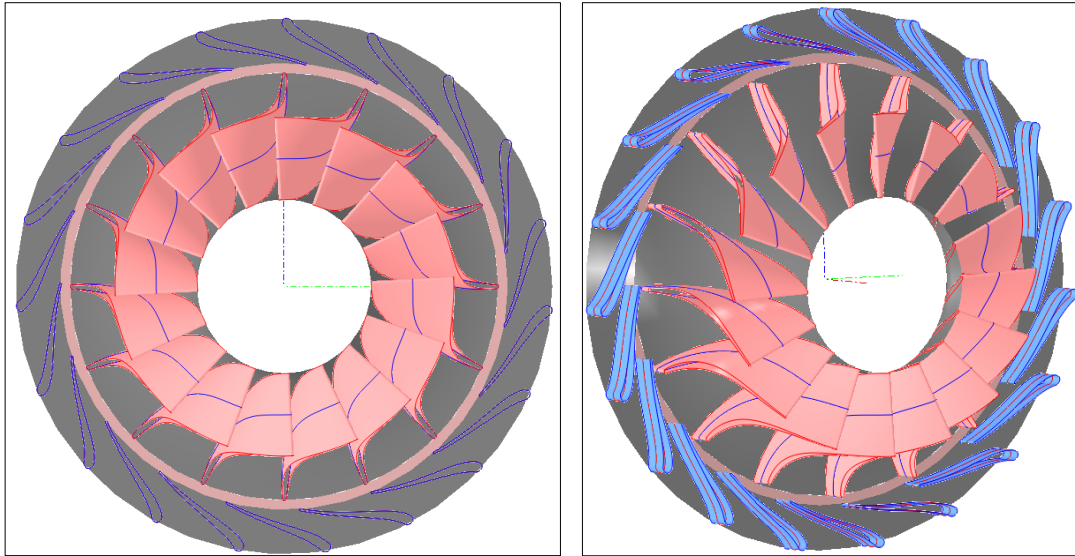


Fig. 6.2 3D model of Case 2

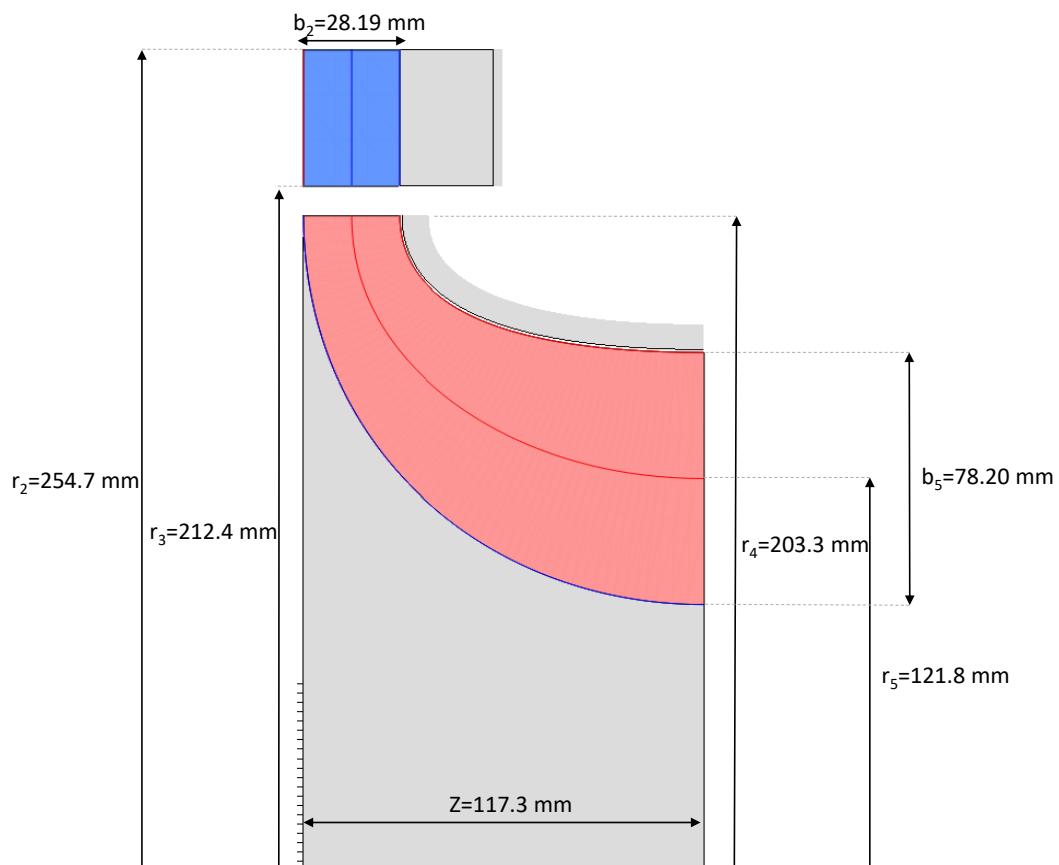


Fig. 6.3 Meridional dimensions of Case 2

Table 6.2 Geometric parameters of design case model

Stator			Rotor		
Variable	Unit	Value	Data	Unit	Value
N_S	-	19	N_R	-	16
c_S	mm	99.90	c_R	mm	176.0
q_S	mm	70.24	q_{R_m}	mm	47.81
t_2	mm	9.150	t_4	mm	3.520
t_3	mm	1.830	t_5	mm	3.520
o_S	mm	16.82	o_R	mm	28.35
θ_2	°	112.42	θ_4	°	90
θ_3	°	13.86	θ_{5_m}	°	36.37

The difference in blade designs between Case 1 and Case 2 can be visualised more clearly in Figure 6.4. A distinction can be made in terms of rotor blade geometry in the highlighted region where Case 2 (left) shows slower and flatter turning. A larger radius at outlet for Case 2 might incur higher outlet velocities which would necessitate a diffuser to make use of the flow potential.

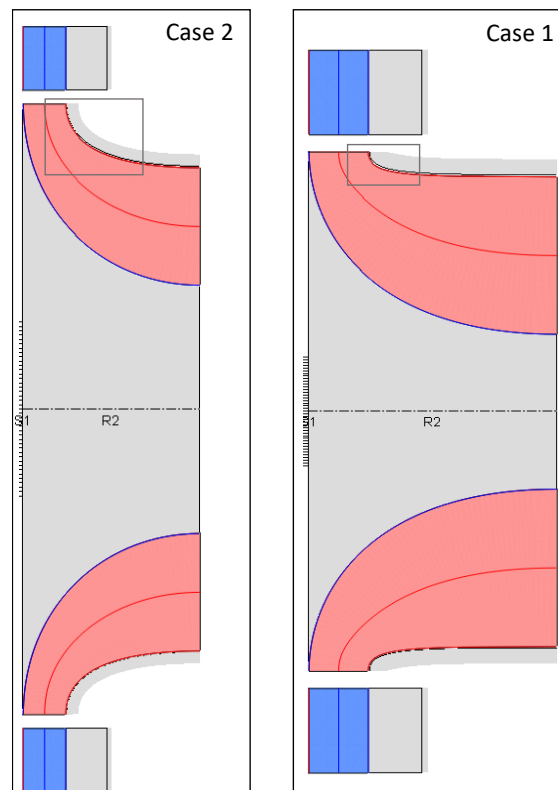


Fig. 6.4 Meridional comparison between revisited design, Case 2 (left) and initial design, Case 1 (right)

6.2.2 Case 2 Performance Evaluation

A streamline curvature analysis is also carried out for the Case 2 design with a number of key performance results shown in Table 6.3. As noted, the total-to-static efficiency of Case 2 is lower than that of Case 1 as expected with an anticipated power output just over the required 100 MW. Results between preliminary design and analysis employing 3 sections (hub, mean and tip) are quite similar; since the 2D analysis showed significant variations when compared to the 3D CFD results presented in Chapter 5, the throughflow analysis is not carried out in detail for Case 2 and its usage is limited to providing a quick check and confirmation against the identified design needs.

For reference, the obtained rotor velocity triangles, from the 3-streamline analysis, are displayed in Figure 6.5 with the correlated thermodynamic properties at each of the turbine stations of Case 2 presented in Table 6.4.

Table 6.3 Performance results comparison between PD and 3 streamline analysis for Case 2

Variable	Unit	PD	3 sections
$\dot{m}$	kg/s	354.2	354.6
$\dot{W}$	MW	101.7	101.6
η_{tt}	%	88.46	88.25
η_{ts}	%	83.80	83.40
v_s	-	0.330	0.344
ψ	-	1.42	1.42
ϕ	-	0.423	0.433
R	-	0.275	0.282
Π_{ts}	-	5.50	5.52

Table 6.4 Thermodynamic properties data for Case 2 turbine

Data	Unit	Nozzle inlet (2)	Nozzle outlet (3)	Rotor inlet (4)	Rotor outlet (5)
T_0	K	1173	1173	1173	943.5
P_0	bar	297	270	270	60.2
h_0	kJ/kg	1400	1400	1400	1114
s	J/kgK	2595	2614	2614	2636
T	K	1171	1009	993.1	927.6
P	bar	294	103	93.1	54.0
h	kJ/kg	1398	1193	1174	1095
ρ	kg/m ³	123.8	52.73	48.64	30.54

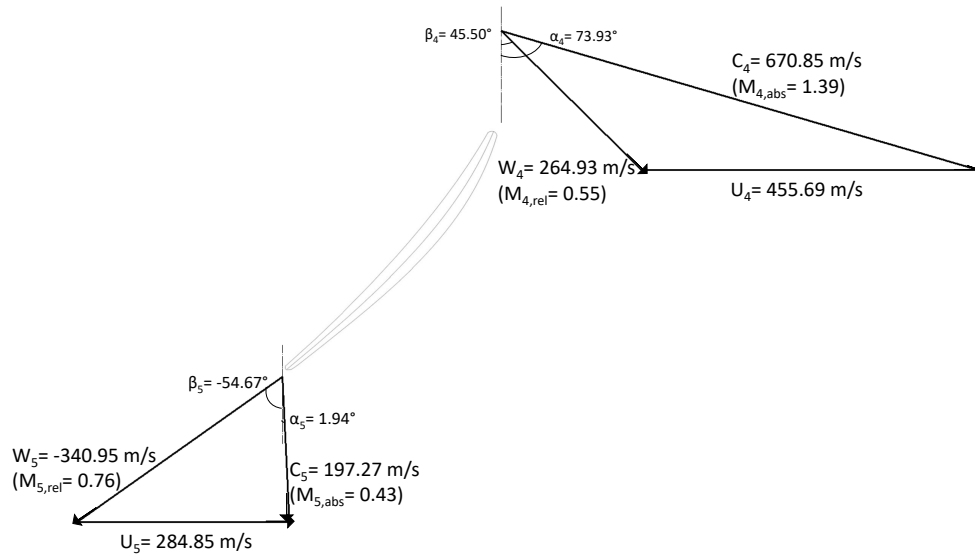


Fig. 6.5 Velocity triangles for Case 2 turbine from streamline analysis

The degree of reaction of Case 2 is around 28% higher than that of Case 1 meaning that the nozzle is responsible for most of the static enthalpy/ pressure drop in the turbine. This is visualised in Figure 6.6 where the static enthalpy fall is shown to take place mostly between turbine inlet up to the stator exit because of the very high absolute velocity occurring across the nozzle, with only a small fraction of the static enthalpy drop taking place in the first half of the rotor passage. Yet again, the effect of having $M \geq 1$ in the vaneless passage on sudden change in thermodynamic properties is not envisioned in the results of the streamline analysis.

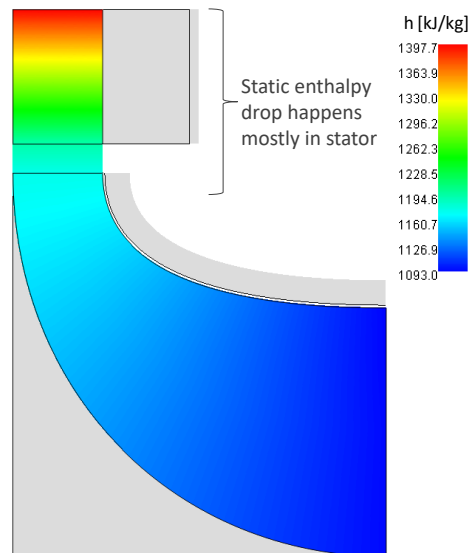


Fig. 6.6 Meridional distribution of total enthalpy for Case 2 turbine

6.3 Computational Fluid Dynamics Investigations

In order to investigate the turbine performance with a greater degree of fidelity, and acquire an insight into the flow physics of the redesigned turbine, computational fluid dynamics modelling is performed for Case 2.

6.3.1 Axi-Symmetric Calculations

To inspect whether to take the current turbine design further along the design process, a computationally inexpensive 2D model is used with the same approach as the one carried out in Subsection 5.3.1; as done in the previous chapter, a mesh sensitivity study is also carried out. Performance results using $P_{0in} = 297$ bar, $T_{0in} = 1173$ K and $P_{out} = 54$ bar (from the main project database of AxSTREAM) as boundary conditions are presented in Table 6.5. The computed mass flow rate is only 5% off (373.5 kg/s) the design target $\dot{m} = 354.6$ kg/s, and therefore is in much closer agreement than what is observed for Case 1. The efficiency does drop by 2 – 3 percentage points when viscous effects are considered in 2D CFD; otherwise all other parameters are in good agreement and within an acceptable range to the results shown in Table 6.3.

Table 6.5 2D CFD performance results for Case 2

Variable	Unit	Value
$\dot{m}$	kg/s	373.5
$\dot{W}$	MW	107.4
η_{tt}	%	86.72
η_{ts}	%	81.34
ψ	-	1.4
ϕ	-	0.441
R	-	0.321
Π_{ts}	-	5.50

Figure 6.7 shows the meridional distribution of relative velocity for Case 2 obtained from 2D CFD simulations. The lower turbine efficiency can be explained by the large indicated area of flow diffusion within the rotor passage which is a source of losses through mixing. Given that this stage of the design process shows promising results, capable of satisfying the defined design intent, it is decided to proceed with this design and conduct a thorough investigation of it with 3D CFD tools.

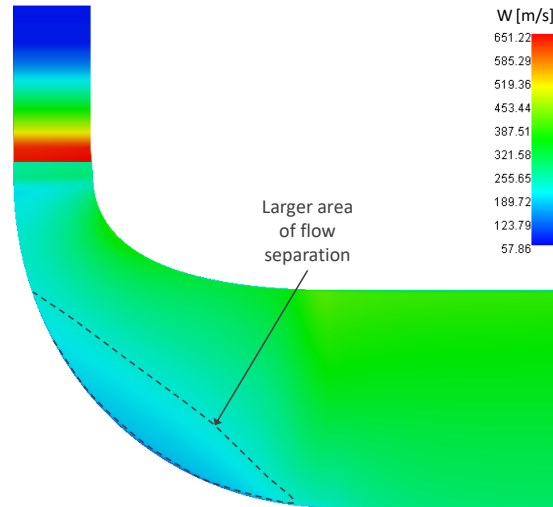


Fig. 6.7 Meridional distribution of relative velocity from 2D CFD for Case 2

6.3.2 3D CFD Results

All necessary steps to establish a working CFD model, from a mesh sensitivity study to setting convergence criteria and defining boundary conditions are followed for Case 2 as was done in the steps outlined in Subsection 5.3.2. Details on the variation of performance parameters with the size of the computational mesh are found in Table D.1 of Appendix D.

The finest mesh with 971,613 H-O grid elements is used for the design point 3D CFD simulations. The adopted problem formulation is with the inlet total pressure and temperature, and outlet static pressure as boundary conditions like in Subsection 6.3.1.

Performance data obtained from the numerical 3D turbulent simulations is listed in Table 6.6 which can be compared to the results from 1D and 2D analyses found in Tables 6.3 and 6.5. Global variables of mass flow, power and non-dimensional parameters all show good agreement with preliminary design and axi-symmetric results. The small difference in thermodynamic properties between 1D and 3D geometry (enthalpies and temperatures) lead to a greater difference in power output. The total pressure at turbine outlet is 62.9 bar with a total temperature of 975 K which are both slightly higher than the values in Table 6.4. The most pronounced discrepancy can be observed in the values of efficiencies that can reach 20%. This change is not fully uncommon with comparisons of 15% efficiency difference between meanline design and CFD results noted by Sauret et al. [83] in their design of a R134A radial turbine, and almost 24% efficiency difference reported by Meijboom [87] between 1D design and 3D CFD modelling for a supercritical CO₂ radial turbine. The source of this variance can be attributed to a number of factors, the turbine 3D blade geometries not being fully optimised, the blade profile loading not fully rectified, and the effects of viscous losses and tip clearance not accounted for in earlier analysis stages.

Table 6.6 3D CFD performance results for Case 2

Variable	Unit	Value
$\dot{m}$	kg/s	372.5
$\dot{W}$	MW	108.5
η_{tt}	%	69.87
η_{ts}	%	63.05
ψ	-	1.34
ϕ	-	0.511
R	-	0.355
Π_{ts}	-	5.51

To better explore the flow physics and the design related losses, contours of key parameters are plotted on different sections throughout the passage. Multiple stream sections (hub-to-tip) are created in both the stator and rotor passages in order to conduct an in-depth investigation, along with three blade-to-blade planes. The location of these surfaces are pictured in Figure 6.8.

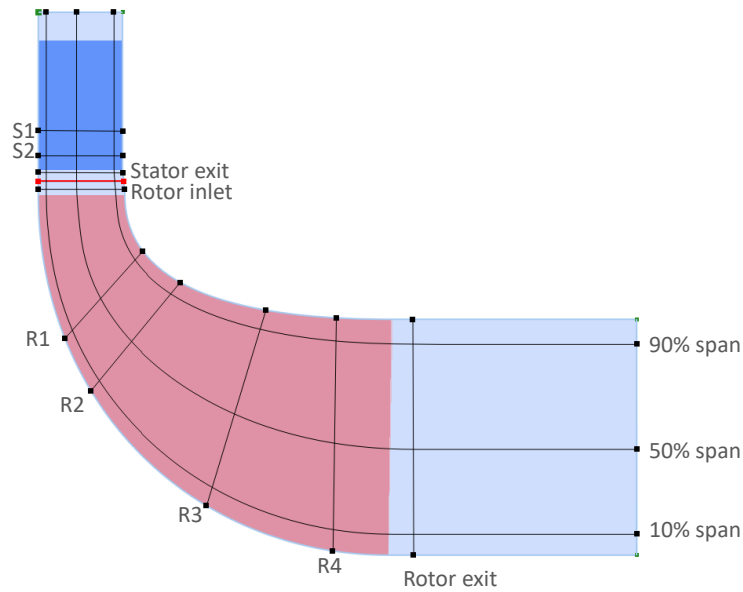


Fig. 6.8 Visualisation of blade-to-blade span-wise planes and hub-to-tip cross-sections for Case 2

Three main flow characteristics can be identified from the relative Mach number distribution in Figure 6.9; the first being the area of incidence loss near the pressure side which lessens from 10% to 90% span with the mean relative flow angle at rotor inlet $\beta_4 = 28.22^\circ$. Secondly, the presence of a highly turbulent region at rotor leading edge which is caused by incorrect incidence and the suction side acceleration interacting with the pres-

ence of the rotor tip clearance. Lastly, the low M_{rel} values on the suction side indicate a region of flow separation that begins at SS leading edge and grows along the flow as end-span is approached (also reported in Dong et al.'s work [106]). Compared to the M_{rel} distribution of Case 1 (Figure 5.17), which denotes similar flow behaviour, the magnitude and intensity of loss regions in Case 2 are greater and that is justified by the low efficiency values.

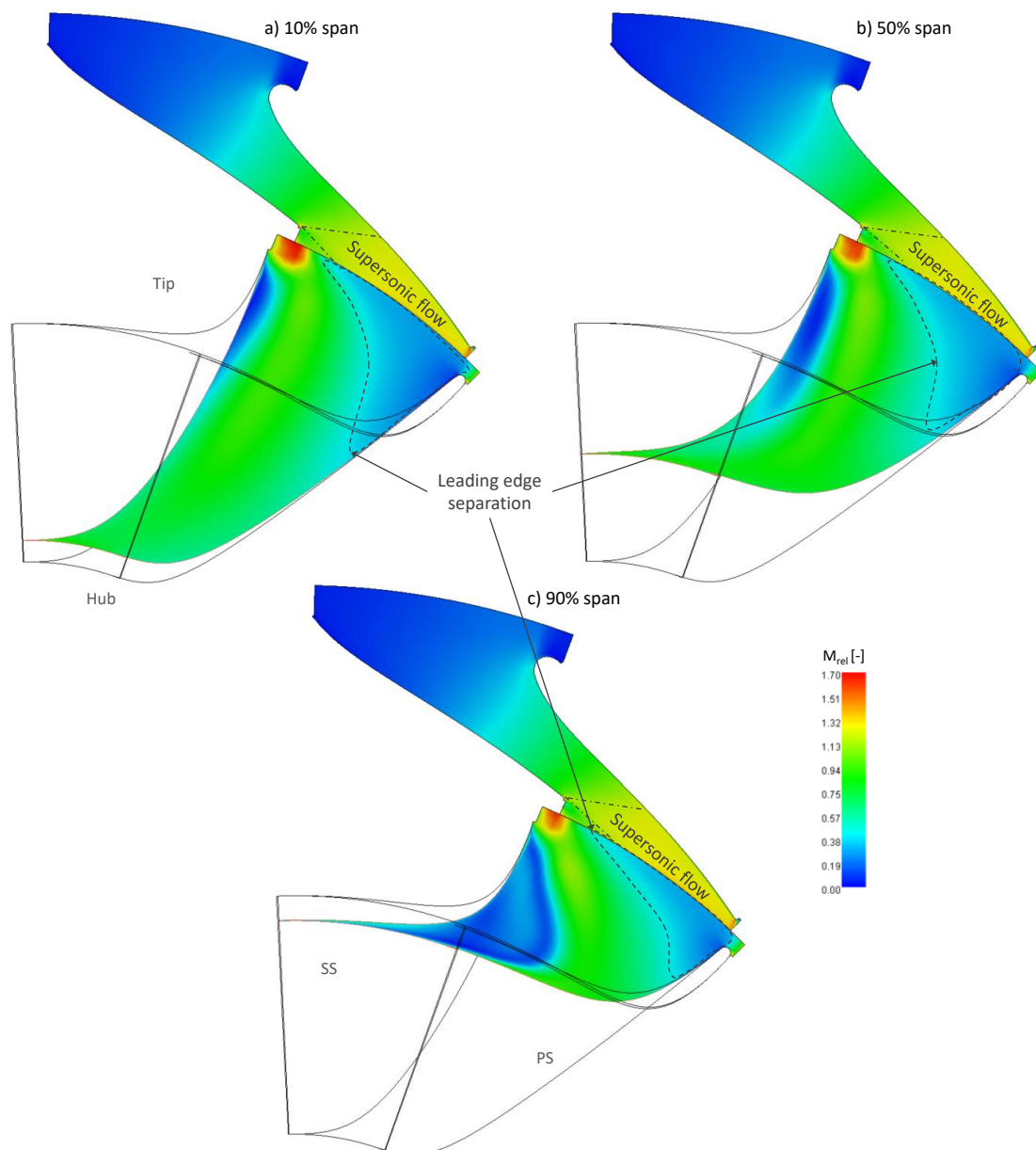


Fig. 6.9 Relative Mach number distribution at three spanwise turbine passage locations for Case 2

Focusing on the rotor passage, Figures 6.10 and 6.11 are viewed in conjunction to define major loss zones. Comparable flow trends to Case 1 are observed. The areas of high entropy values correspond to the loss generation and are located at vortices, flow separation and reattachment, mixing between stator and rotor, and wakes (or their merging). The severity of the loss locations and their expanse help explain the low efficiency obtained for the Case 2 turbine design. The entropy rise across the passage is more than double the initially predicted value (88.92 J/kg K vs 41.07 J/kg K). The flow deceleration in some parts of the passage indicate separation or flow reversal.

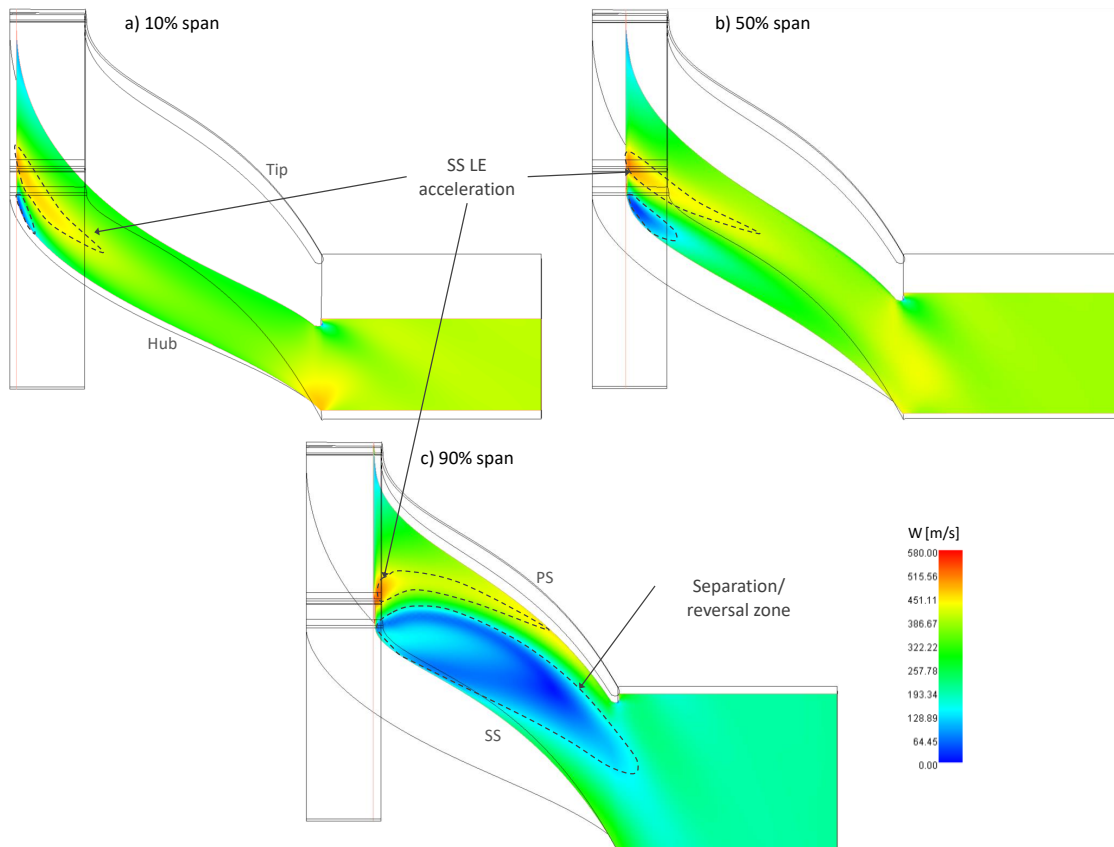


Fig. 6.10 Relative velocity contour at three spanwise rotor passage locations for Case 2

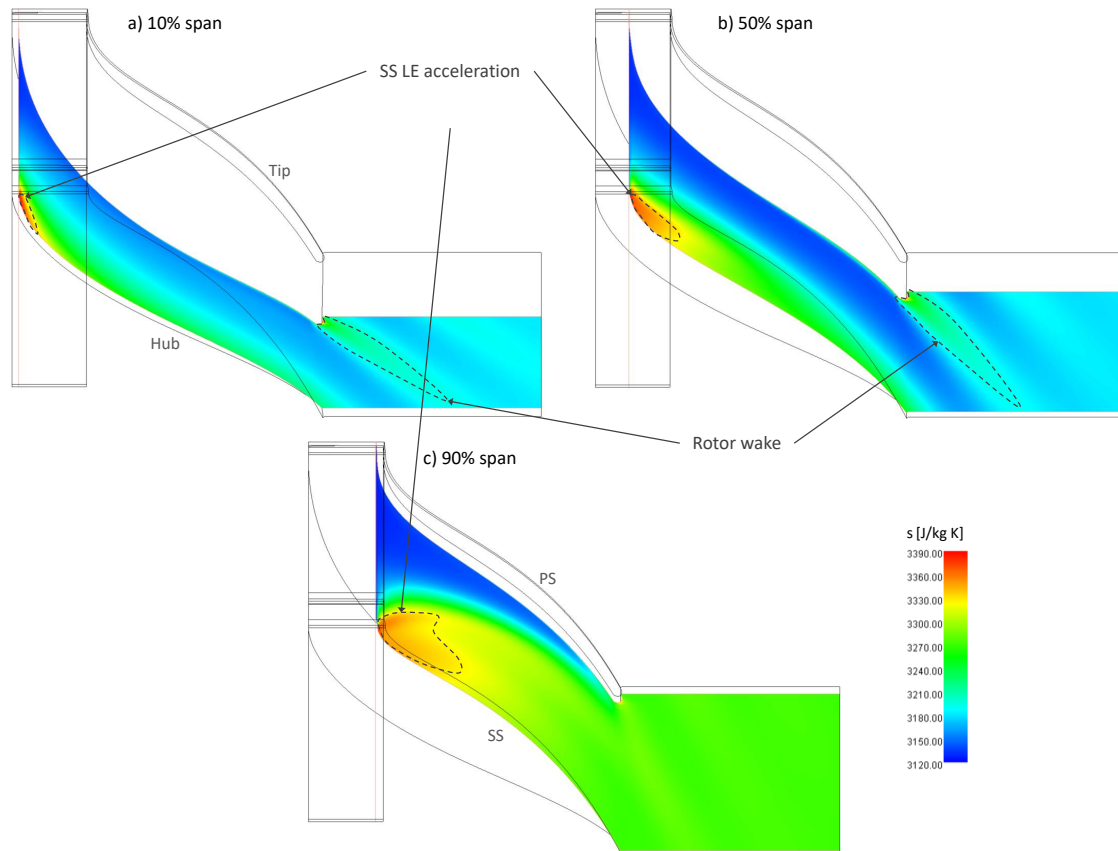


Fig. 6.11 Entropy contours at three spanwise rotor passage locations for Case 2

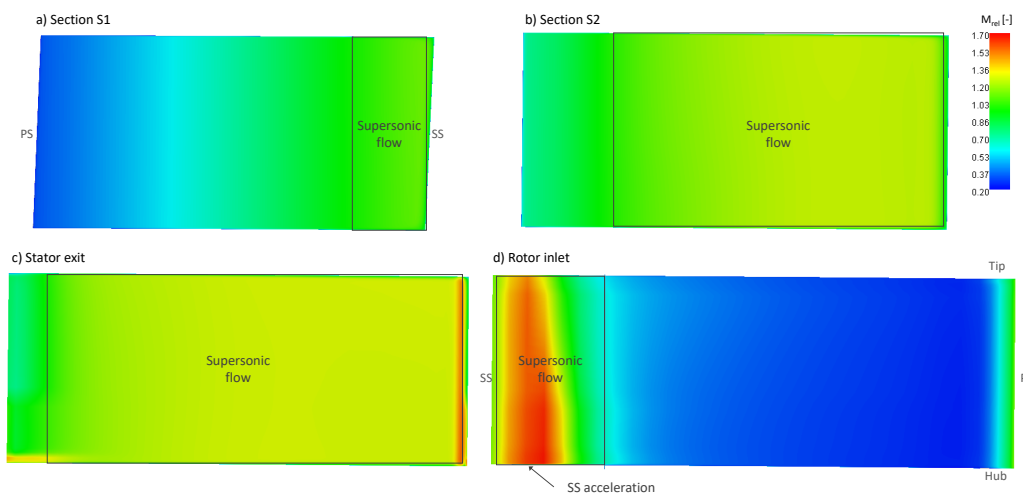


Fig. 6.12 Relative Mach number distribution at stator passage and outlet and rotor inlet cross-sections for Case 2

Figure 6.12 is used to indicate the areas within the passage that encounter supersonic flows; those exist near the stator trailing edge (starting at the minimum passage area of the throat), in the vaneless passage and in a small portion near rotor inlet. The supersonic region begins at the high-velocity suction surface of the nozzle and expands towards the stator pressure side downstream. At stator exit, although the average Mach number is greater than 1, the supersonic flow regime does not occupy the full outlet passage. The rotor inlet suction side encounters a small region of $Mach \geq 1$ with the incoming flow from the nozzle. Realising that most of the stator losses are associated with shockwave effects due to the supersonic flows, a deduction that the stator blade requires better design to reduce the outlet Mach number can be made.

Total pressure losses occur in the rotor passage as indicated in Figure 6.13. At sections after rotor entry, the most pronounced total pressure loss (blue colour) happens on the suction side from hub to tip because the pressure side still maintains the high pressure characteristic. As the flow moves downstream, the total pressure loss vortex loses its magnitude compared to the bulk flow (green colour with no gradient), shifts towards the shroud region, and extends to reach the pressure side near rotor exit because of the formation of tip leakage.

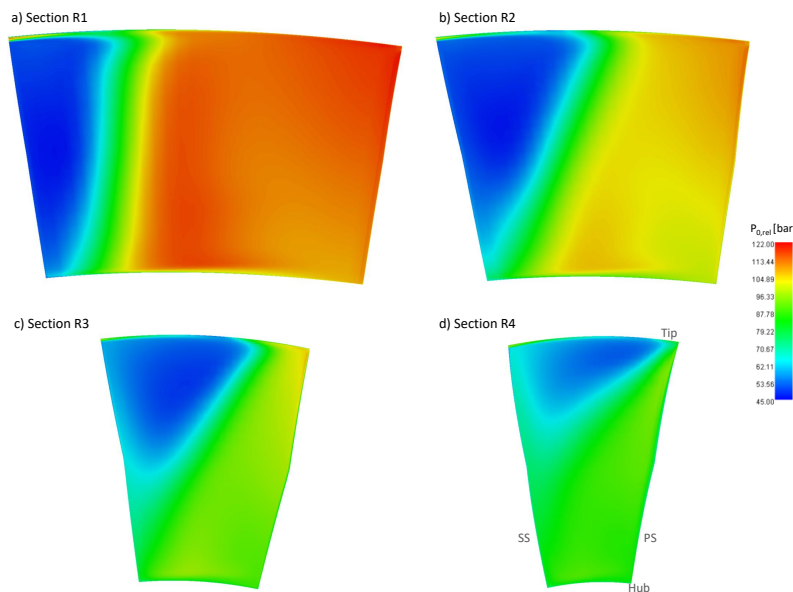


Fig. 6.13 Relative rotor pressure distribution in rotor stream-sections for Case 2

The large tip clearance leakage area is visible at the rotor exit stream-section in Figure 6.14. This loss area is considerably larger compared to the equivalent leakage flow observed in Case 1 (Figure 5.21) meaning more total pressure losses are present in Case 2.

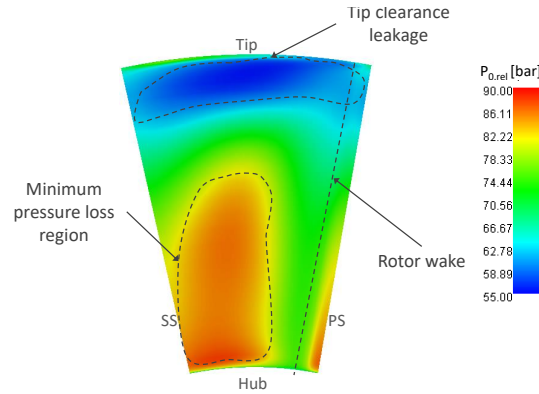


Fig. 6.14 Relative total pressure contours at rotor exit stream section for Case 2

6.3.3 Off-Design Results

Next, off-design conditions are imposed to obtain performance characteristics of the Case 2 turbine over an operating range.

Off-Design Characteristics

The turbine performance maps are generated using 2D CFD simulations rather than the full 3D ones, on account of the computational resources available as well as the long time required for the 3D simulations. The conditions extend over the range of total-to-static pressure ratio between 2 and 7 at three rotational speeds of 19,268 rpm, 21,409 rpm and 23,550 rpm. The variation of mass flow rate and stator outlet Mach number with the off-design conditions is shown in Figure 6.15. In general, the mass flow is expected to increase with higher pressure ratios but have lower values at higher rotational speeds, until choking occurs whereby the mass flow rate reaches its maximum asymptote. The radial turbine becomes choked at around $\Pi_{ts}=3$.

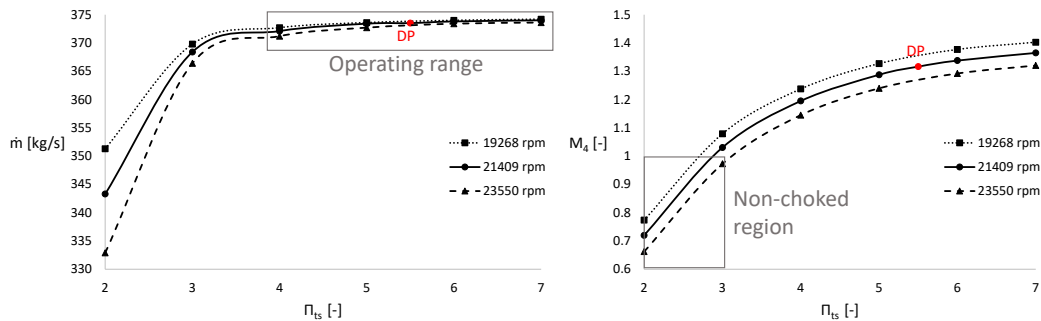


Fig. 6.15 Variation of mass flow and stator outlet Mach number with total-to-static pressure ratio at three rotational speeds for Case 2

Both total-to-total and total-to-static efficiency maps are plotted in Figure 6.16. The difference between the efficiency values is solely attributed to the loss in the available kinetic energy at rotor outlet, i.e. related to the exit swirl (α_5) as explained in Subsection 3.2.2, because the axial velocity is not a function of rotational speed (but is related to pressure ratio). The trend for both efficiencies are broadly similar except for the value difference due to the mentioned loss. The highest efficiency value occurs with a pressure ratio of about 3 at 23,550 rpm and the lowest with $\Pi_{ts}=7$ at 29,268 rpm. The maximum Mach number at stator outlet is lower for higher ω (as deduced from Figure 6.15) meaning that the entropy rise in the stator exit channel is also lower, hence the higher efficiencies.

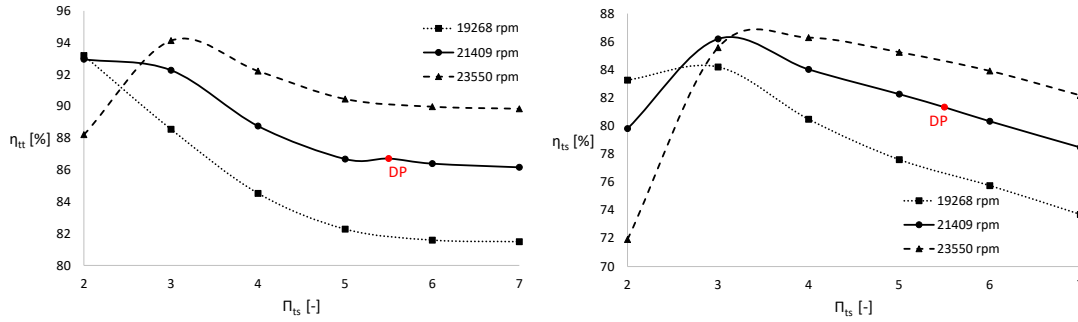


Fig. 6.16 Variation of total-to-total and total-to-static efficiencies with total-to-static pressure ratio at three rotational speeds for Case 2

Table 6.7 shows the change in relative flow angle at rotor inlet and absolute flow angle at rotor outlet at the off-design conditions. The flow angle at rotor inlet (β_4) plays a role in determining incidence losses. The recommended values of rotor incidence angles for a radial turbine are in the range of -40° – -20° [91]; incidence angles around the suggested values are only obtained in two cases (-37.01° & -44.19°), all which correspond to pressure ratios below choking conditions and those cases correspond to relatively high efficiencies. Ideally, the closer the absolute flow angle at rotor outlet is to zero, the lower the kinetic energy loss will be; therefore the lower the difference between η_{tt} and η_{ts} is. Values of $\alpha_5 \approx 0^\circ$ are attained at higher rpms because of the lower Mach number at stator outlet/ rotor inlet which in turn leads to a lower relative Mach at rotor outlet, and thus can have lower absolute flow angles for similar pressure ratios at different speeds.

Table 6.7 Variation of relative flow angle at rotor inlet and absolute flow angle at rotor outlet at off-design conditions for Case 2

Π_{ts}	2	3	4	5	6	7
$\beta_4 [^\circ]$						
19268 rpm	-1.07	43.97	49.88	51.10	51.28	51.25
21409 rpm	-37.01	-44.19	40.92	44.21	45.29	45.65
23550 rpm	-58.45	0.47	26.21	33.57	36.22	37.29
$\alpha_5 [^\circ]$						
19268 rpm	56.5	36.8	15.95	0.02	-10.64	-17.22
21409 rpm	63.06	-27.85	26.09	9.56	-2.35	-10.27
23550 rpm	69.81	48.48	34.49	18.65	5.87	-3.04

Off-Design 3D CFD Analyses

The total-to-static pressure ratio and rotational speed of the most and least efficient points are used in 3D CFD to visualise the flow behaviour for each one. Kinematics and performance results for each case are presented in Table 6.8. As mentioned before, there is a discrepancy in efficiency values between 2D and 3D CFD calculations because of higher fidelity account of losses computed by the latter. The better efficiency condition does not relate to higher power output because of the lower flow speeds in accordance to Eq. 3.56. The entropy generation across the stage is around four times more when running at 19,268 rpm and a high pressure ratio. The condition with high rotational speed poses a material limitation issue relating to the tip speed value $U_4 = 507.91$ m/s.

Table 6.8 Kinematics and performance parameters at two off-design conditions for Case 2

Condition	$\dot{m}$ [kg/s]	C_4 [m/s]	W_4 [m/s]	U_4 [m/s]	C_5 [m/s]	W_5 [m/s]	U_5 [m/s]	$\dot{W}$ [MW]	η_{It} [%]	η_{Is} [%]	Δs [J/kg K]
23550 rpm $\Pi_{ts} = 3$	360.6	496.66	137.75	507.92	189.01	193.49	301.21	65.77	82.40	72.39	25.47
19268 rpm $\Pi_{ts} = 7$	376.9	659.40	308.35	415.56	296.97	430.36	246.43	103.8	65.19	57.22	108.4

Figure 6.17 shows the relative Mach number distribution for both cases at mid-span. One can note the absence of the shockwave and supersonic flows at the stator outlet channel for the conditions with high rpm. The right-hand-side figure also shows that the inefficient turbine exhibits loss features that include the presence of a vortex, incidence losses ($\beta_4 = 35.44^\circ$) and suction side separation or flow reversal as explained in Section 6.3.2. Whereas, the left-hand-side figure shows smooth flow acceleration and deceleration as would be ideal in a radial turbine case with proper rotor incidence of $\beta_4 = -12.93^\circ$. The aim would be to design a turbine that coincides with such flow behaviour, however, op-

erating conditions imposed by the cycle might not allow the elimination of supersonic flows.

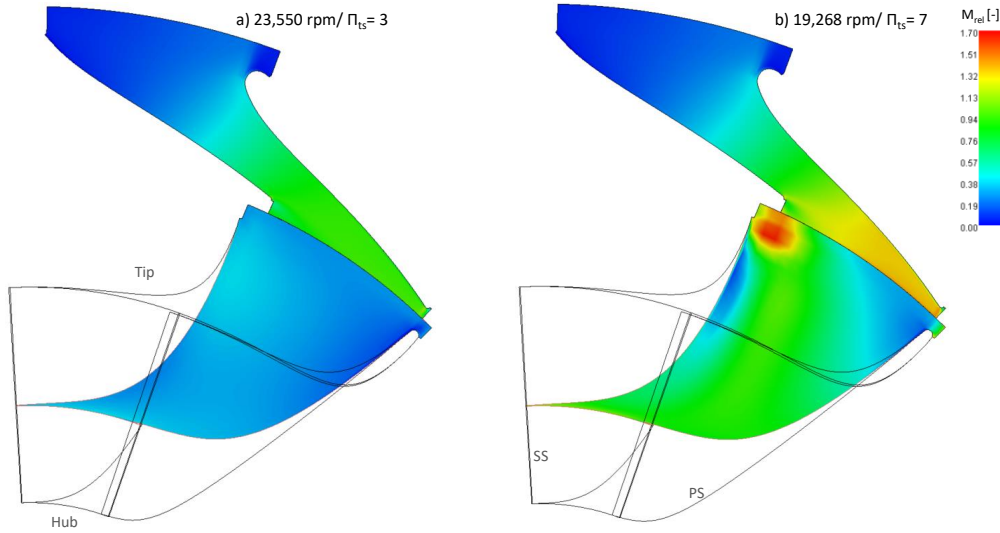


Fig. 6.17 Off-design relative Mach number distribution at mid-span from 3D CFD simulations of Case 2: 23,550 rpm, $\Pi_{ts}=3$ (left) and 19,268 rpm, $\Pi_{ts}=7$ (right)

Investigation of the total pressure losses at the rotor exit hub-to-tip surface (Figure 6.18) emphasises the efficiencies at each off-design condition. The figure on the left shows a small fraction of tip leakage losses with a pressure loss region near the hub (25% P_0 loss near hub); while the one on right left, corresponding to the inefficient conditions, displays more pronounced shroud leakages with a proportionate value of 48% with respect to the areas of least pressure drop (red zone).

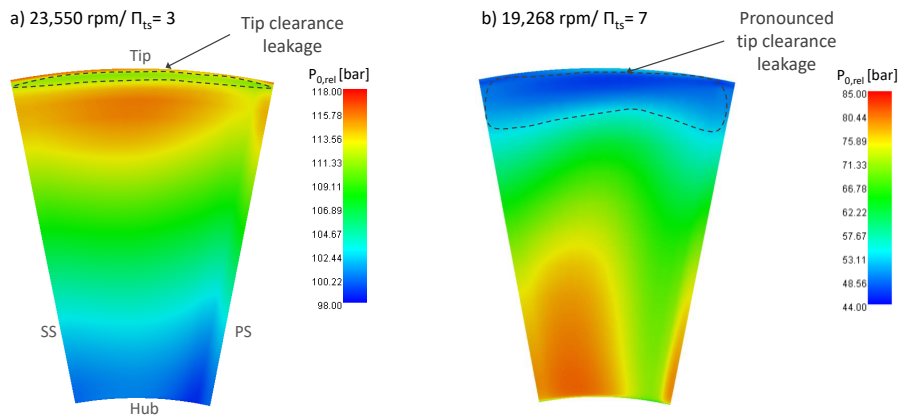


Fig. 6.18 Off-design relative total pressure distribution at rotor exit cross-section from 3D CFD simulations of Case 2: 23,550 rpm, $\Pi_{ts}=3$ (left) and 19,268 rpm, $\Pi_{ts}=7$ (right)

The design point conditions of the Case 2 turbine (Section 6.3.2) show flow behaviours that are more analogous to the inefficient characteristic at the worse off-design conditions.

6.3.4 Isolated Rotor Simulations

An isolated rotor model is primarily developed for use in stress calculations but is also utilised for examination of the rotor performance without the inclusion of the stator effects. The boundary conditions are taken as dictated by the main project database of AxSTREAM (Subsection 6.2.2) with $P_{0in} = 270$ bar, $T_{0in} = 1173$ K at an absolute flow angle of 73.93° and $P_{out} = 54.0$ bar. Performance results of the rotor only model at design point are listed in Table 6.9. The computed total-to-static pressure drop is lower than the defined value of with the CFD value of P_{0in} being 256 bar. The mass flow rate is very close to the design 354.6 kg/s rate along with much better efficiencies compared to the full turbine cascade. The power output is around 14% lower than the 100 MW because of the lower pressure ratio (absence of stator means no static pressure drop there). The solution indicates that the a large portion of losses arise in the stator-rotor passage and the incorrect flow conditions supplied by the nozzle.

Table 6.9 3D CFD performance results for Case 2 rotor

Variable	Unit	Value
$\dot{m}$	kg/s	356.3
$\dot{W}$	MW	86.38
η_{tt}	%	87.80
η_{ts}	%	80.46
ψ	-	1.21
ϕ	-	0.423
R	-	0.406
Π_{ts}	-	4.75

The visualisation of several parameters is seen in Figure 6.19 where two blade passages are set side by side at mid-span. At the rotor leading edge, the flow experiences, as expected, an acceleration on the suction side and a slowing down on the pressure side with the corresponding pressure changes. There still exists a rise in entropy on the blade suction surface with the diffusion of the flow. However, relative velocity and total pressure distributions do not indicate full flow separation or reversal. In addition to that, the absence of the incidence loss region, and the lower velocities at rotor inlet allow for a more uniform flow distribution on the rotor blade.

The isolated rotor model definitely gives better performance characteristics than the full turbine passage (stator and rotor). However, the power output is 86.38 MW which is lower than the target 100 MW range meaning that either a greater pressure drop is required, or a new design of an integrated nozzle is needed. In the former case, the higher

pressure ratio can be obtained via higher velocities (increasing rpm or changing blade angles) however, it is important to maintain the speeds imposed by material limits.

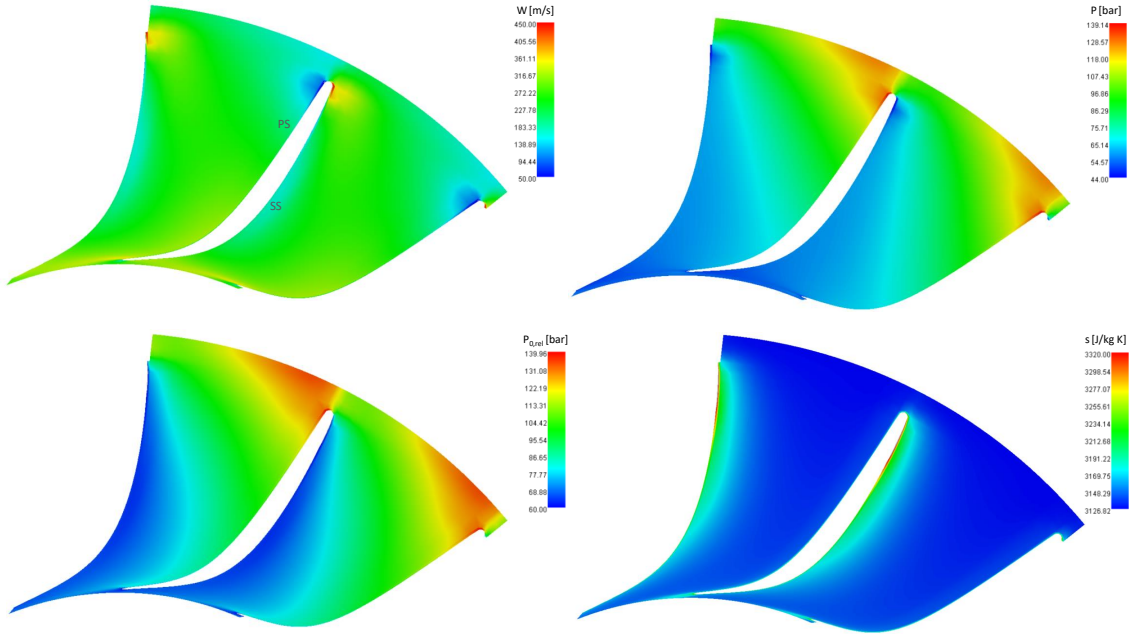


Fig. 6.19 Mid-span parameters distribution for rotor only 3D CFD model of Case 2

6.4 Rotor Stress Analysis

Meijboom [87] explains the importance of taking the blade gas loads into consideration, rather than just the centrifugal stresses, when assessing the mechanical integrity of the radial turbine. Therefore, rotor simulations are crucial for attaining the blade aerodynamic loadings that are generated by the high pressure ratio of the dense sCO₂ fluid. The aerodynamic data is coupled to the stress model to account for fluid-structure interactions when mechanical design analysis is performed.

6.4.1 AxSTRESS Problem Formulation

AxSTRESS is used to perform static stress modelling on the isolated rotor.

Shaft Dimensions

The first step in developing the static stress model is fixing the rotor blades on a disc which is in turn fixed on the rotating shaft. Hence, before constructing a suitable disc design, an estimation of the shaft maximum diameter is performed to determine the disc base radius.

Following the procedure outlined in Subsection 3.6.1, the mass of the rotating element of the turbine (rotor blades + disc) is calculated by multiplying the material density by its volume. The definition of the turbine geometric domains is outline in Table 6.10 and was performed using Solidworks to generate initial blade and disc geometries in order to obtain order of magnitude figures.

Table 6.10 Volumes of fluid domains and blade domains of the radial turbine

Volume of	Value in $10^5 \cdot \text{mm}^3$
Full nozzles domain (fluid+blades)	84.2
Single nozzle fluid domain	6.17
Full nozzle fluid domain (13 blades)	80.2
Nozzle blade set	4.03
Full rotor domain (fluid+blades)	257.4
Single rotor fluid domain	14.8
Full nozzle fluid domain (16 blades)	236.7
Rotor blade set	20.7
Turbine blades	24.7
Turbine disc	48.7

The volume of the rotating components is then $69.4 \times 10^5 \text{ mm}^3$ which yields a mass of 58.6 kg (62.07 kg for the whole turbine), using IN 718 with a density of 8442.37 kg/m^3 as the blade material. The axial length of the turbine from the shaft fixing is taken as 0.21 m. The axial thrust on the turbine is taken from AxSTREAM/ AxCFD. The value of the axial force calculated from the AxSTREAM preliminary design is 286,349 N, whilst from the 3D CFD results the axial thrust on the turbine is 767,800 N. Although this is a considerable discrepancy, the practical implications for the purposes of this study are minor since the difference between the forces affects the shaft outer diameter by just 1.4%. The torque is identified from the turbine power output and shaft rotational speed. After obtaining all the required input to Eq. 3.60, the non-imaginary root of the equation corresponds to the value of the shaft outer diameter which is in the range of 120 mm.

Mesh Generation

A generic impeller model is created by using a base blisk-type spline curve that follows the meridional flow path of the blade with small extensions at leading and trailing edges for fillet radii, and a manipulation of the disc contour curves to develop a suitable back-plate. A mesh is then generated to discretize the 3D CAD model for equation solving. A mesh refinement study is carried out to compare how the solution changes as element sizes are reduced. A mesh with 216,470 nodes, visible in Figure 6.20, is considered adequate for the purposes of this work. This mesh size is imposed largely due to the computational resources available but it is representative of coarse meshes described in

comparable literature reports.

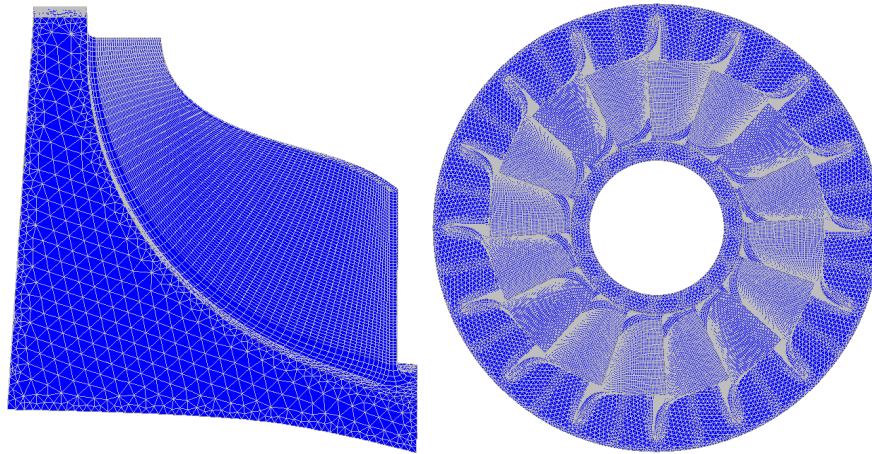


Fig. 6.20 AxSTRESS computational mesh

Disc Design

Changing the disc design and the backface thickness can impact the maximum stress values and concentrations on the disc itself. Figure 6.21 shows two disc/ backface configurations where a) has a maximum stress of 1013.44 MPa and large zones of increased displacement, whilst b), which has a thinner base and a more smoothly filleted backface, has a maximum von-Mises stress value of 840.86 MPa. The disc design of b) is used for the current stress calculations although it does not contain scalloping which is a technique of mass removal on the backface to reduce stresses [135].

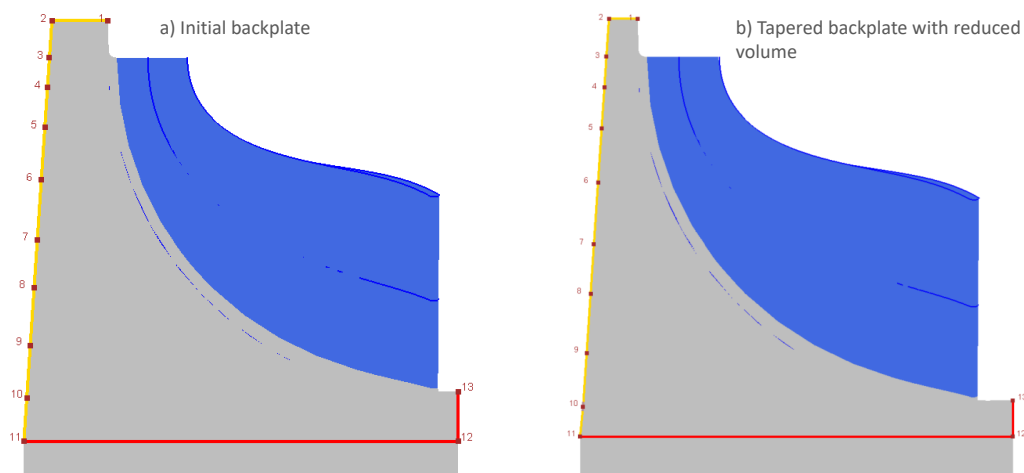


Fig. 6.21 Two disc/ backplate designs: initial backplate (left) and updated backplate (right)

6.4.2 Design Point Stress

The criterion against which safe operation is measured is the maximum allowable von-Mises stress for the turbine which is dictated by 0.2% yield strength of the material at the working temperature. The 0.2% S_y for IN 718 falls significantly as temperatures are increased beyond 650°C, whilst no experimental values for temperatures beyond 760°C could be identified from a thorough literature survey. The 0.2% S_y for $T = 760^\circ\text{C}$ is 758 MPa [137]. The maximum temperature on the rotor (from the rotor simulations) is 827°C, so an assumption of having the corresponding yield strength be 560 MPa is made based on materials data and trends [136]. A safety factor of 1.4, which is a usual value for turbomachinery components, as used by Khader [135], is used to ensure an error margin for mechanical integrity, which makes the maximum allowable stress to be 400 MPa.

The design point loadings from the rotor only model are used as aerodynamic loads in the static stress solver along with a constant thermal load of 1100 K. The distribution of von Mises stresses is displayed in different views using Figures 6.22 and 6.23. For better visualisation of the areas of high stress concentration, the contour limits of the displayed von Mises stress is set to the maximum 400 MPa; the grey regions are areas with stress over the permissible ceiling. The maximum value of 988 MPa is reached at the point of contact between the backplate of the rotor disc and the rotating shaft because the material mass at outer radii induce loads on the regions of smaller radii. Overall, the rotor blade itself can be seen to have stresses below the assumed material limits. The grey areas are generally the regions of high stress accumulation because of attachments of either disc to shaft or blade to disc as reported by Schwalbach et al. [146] in their paper on structural analysis of a radial turbine. The stresses on these areas can be lowered by careful mechanical design that is beyond the scope of this work.

Figure 6.25 shows the total deformation (ϵ) on the rotor whereas a more detailed displacement component distribution is shown in Figure 6.24. Maximum deformation of 0.505 mm takes place at rotor tip because of the high circumferential speeds and the deformation happening in all directions in that region (Figure 6.24), while no deformation happens at the disc base due to its rigid attachment to the shaft. Similar displacement trends on radial turbine blades are reported by Tanwar et al. [129] and Odabae et al. [147] but at a smaller scale because of the different operating conditions. Radial growth deformation (in the z axis) is mostly present at rotor leading edge, as observed in Figure 6.24, with the high load from the inlet gas which requires a tip clearance to accommodate any material expansion.

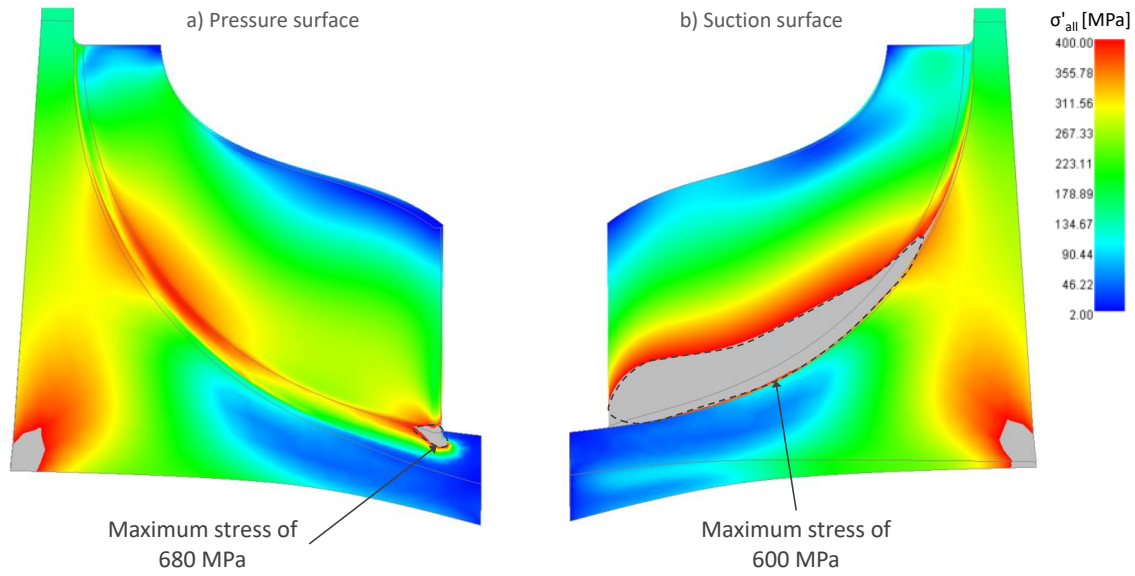


Fig. 6.22 von-Mises stress distribution on rotor blade surfaces: pressure surface (left) and suction surface (right)

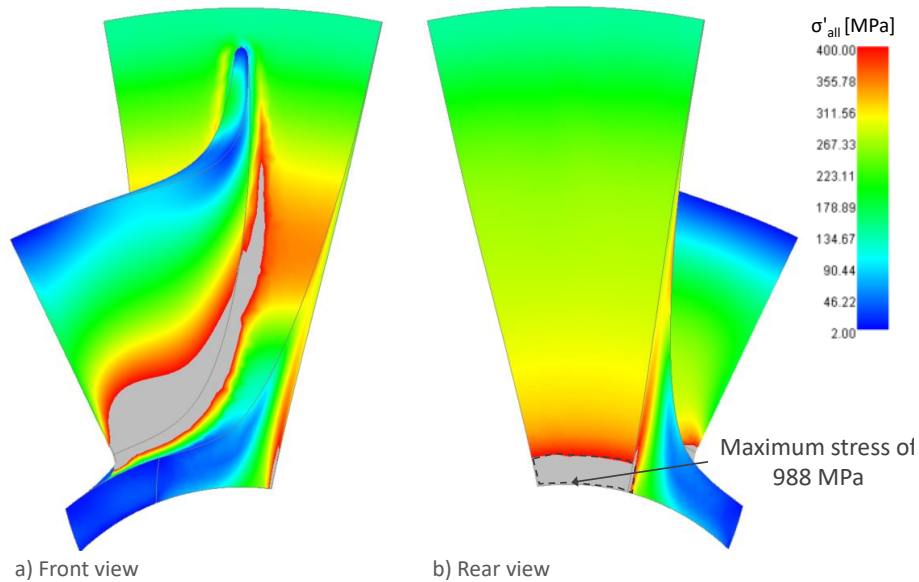


Fig. 6.23 von-Mises stress distribution with different views: front view (left) and rear view (right)

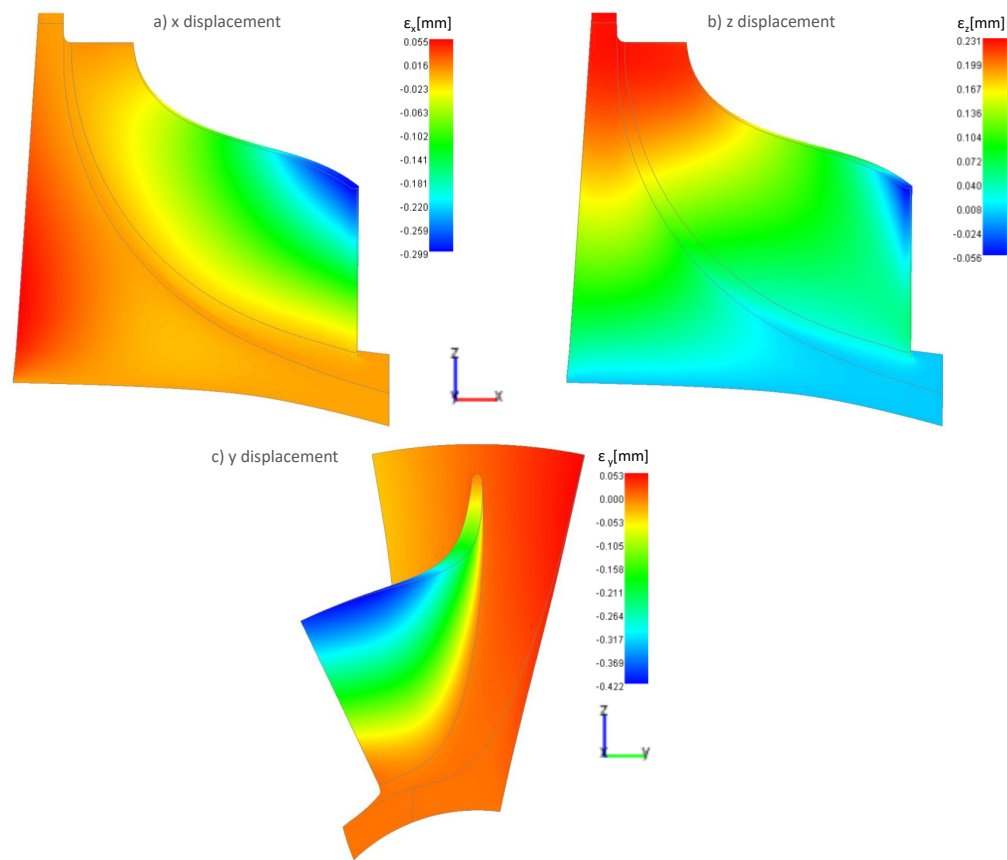


Fig. 6.24 Components of impeller deformation

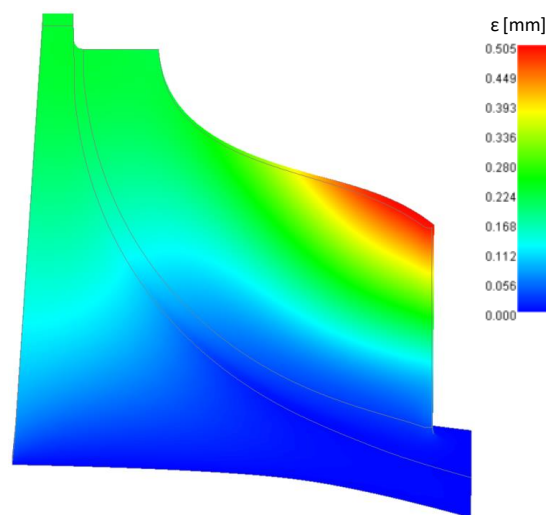


Fig. 6.25 Impeller total deformation

6.4.3 Off-Design Stress Spectrum

A stress load spectrum can be identified for conditions of operation ranging from minimum load (low ω , low Π_{ts}), passing through design point conditions up to maximum load (high ω , high Π_{ts}). The two extreme conditions are based on off-design environments, with the lowest aerodynamic load of 19,268 rpm and $\Pi_{ts}=3$ and highest blade loading of 23,550 rpm and $\Pi_{ts}=7$, and are used for obtaining the necessary fluid pressure distributions on the blade from the rotor only AxCFD simulations. The expected von Mises stress distribution for either case is displayed on the pressure and suction surfaces in Figures 6.26 and 6.27 respectively. The conditions with high aerodynamic loads incur larger stress values over wider regions; for example, even at the pressure side blade root, there are areas of stress that exceed the maximum allowable 400 MPa value. The left-hand-side figures show smaller stressed regions than those obtained at design point because of the more relaxed environment. The maximum exceeded von Mises stress values over a wide range of turbine operating speeds and pressure ratios vary from 470 MPa at blade root to 1197 MPa at disc root; the maximum rotor blade expansion ranges from 0.492 mm to 0.546 mm, all of which are within the shroud clearance of 0.782 mm. A more detailed disc design is essential to lower stress areas in similarity to the low loading environment of the left image of Figure 6.26. The consideration of stress and material limitations is discussed in Chapter 7.

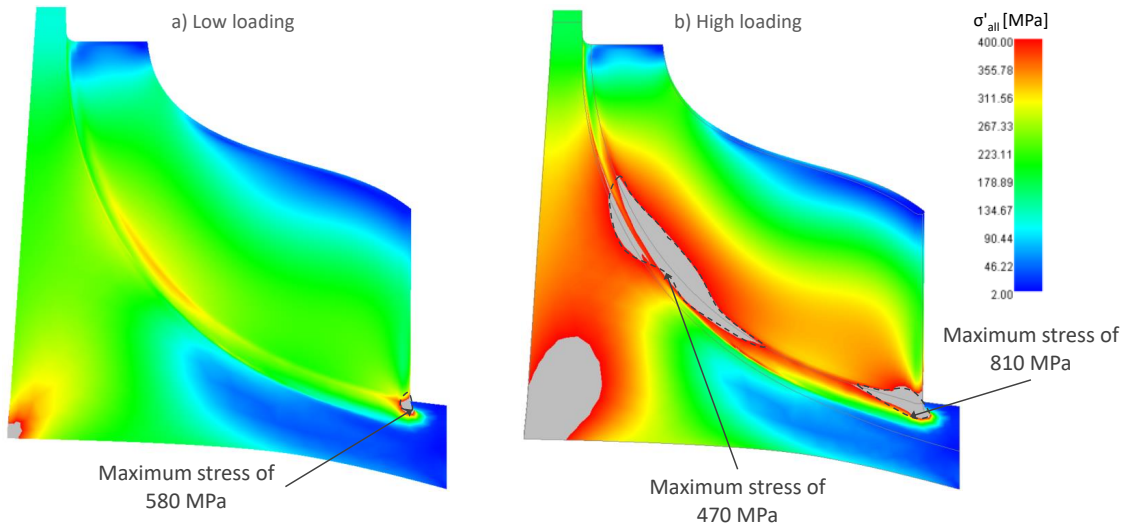


Fig. 6.26 von Mises stress distribution on the pressure surface for extreme loading conditions: low (right) and high (left)

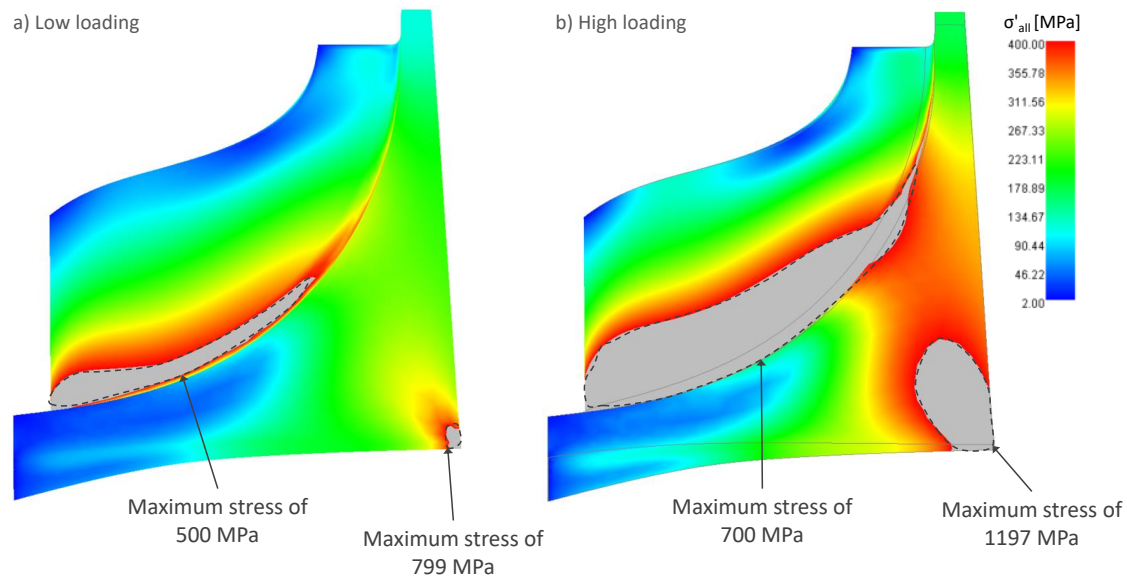


Fig. 6.27 von Mises stress distribution on the suction surface for extreme loading conditions: low (right) and high (left)

Chapter Summary

The final results chapter covers the same methodology adopted in the previous chapter for the generation of an updated turbine design that can satisfy the boundary conditions required by the cycle. The new design is around 510 mm in diameter and runs at 21,409 rpm with a smooth rotor turning angle. Although the turbine fulfils its purpose, the expected total-to-static efficiency from CFD calculations is relatively low at a value of 63.05%. Flow behaviour comparisons between the revisited design and the initial one are made to understand loss mechanisms and ways of improving performance. Results obtained from the isolated rotor CFD simulations show a better performance than that from the integrated turbine passage where the rotor total-to-static efficiency is 80.46%. The rotor experiences a normal expected flow behaviour across both pressure and suction sides; this means that the conditions exiting the stator, in the turbine passage case, are not suitable for the designed rotor and require adjustment through vane design modifications. Static stress analysis is performed for the turbine wheel to inspect the structural integrity of the proposed design. High stress regions can exceed the maximum allowable limit of 400 MPa and are concentrated at blade and disc roots. The discussion of the results and the interconnection of the full study is disclosed in the following chapter.

CHAPTER 7

Discussion

The discussion chapter is aimed at linking the previous chapters together, and at providing a high-level discussion of the results of this work. In it, a justification of the choices made and critical observations of the outcomes are presented. It gives the author's recommendations on the enabling technologies required to continue the development of a radial turbine suitable for an uncooled, oxy-fuelled, supercritical CO₂ cycle with a power output of around 60 MW_e (a 100 MW_{th} turbine). Those include comments on the issues already observed during literature review (Section 2.3.2) such as materials and ancillary components. A comparison of operating speeds and sizes is made between the designed radial turbine and other developed steam and gas turbines of similar power outputs.

7.1 Thesis Observations

Energy engineering is the use of science which covers an extremely wide spectrum of fields that includes design, management, and operation studies of energy technologies to deliver solutions that meet the growing needs of society under rigorous limitations. An array of very different power generation proposals is present to address the global requirements of providing clean power in an efficient manner while using minimalist resources of land and fuel. Solutions include renewable energy generation from natural non-perishable resources, improving energy efficiency of fossil-fuelled power plants, the use of alternative fuels (hydrogen, biofuels), the integration of novel technologies to existing power systems, and the creation of newly-developed concepts. Chapter 1 offers an extensive presentation of some of the most innovative modifications and advancements to power production cycles to increase energy efficiency and mitigate emissions. Natural gas cycles are identified to be continuously prominent in the future direction of electricity gen-

eration, however the associated greenhouse gas emissions are to be persistently reduced. The NET Power cycle comes to the forefront of methods directed at the environmentally friendly use of natural gas fuel for its unprecedented incorporation of oxy-combustion for carbon capture, and the utilisation of unconventional supercritical CO₂ as a plant working fluid.

The original invention of the NET Power cycle requires multiple development steps to reach a sufficient technology readiness level for actual adoption. The critical components for consideration are discussed in Chapter 2, along with the main barriers to progress that need to be overcome. One of the most notable cycle units for study is the driving element which is the power turbine. Its functionality imposes completely new design requirements which include operation at elevated temperatures and pressures in an unorthodox fluid environment. The design suggested by the developers is a cooled axial turbine which expands the fluid using multi-stages from conditions of near 300 bar/ 1150°C to around 50 bar/ 775°C, achieving a claimed net cycle efficiency of $\approx 59\%$ [86].

One of the principally identified benefits of supercritical CO₂ cycles, that is a chief reason for the conducted work, is the high density fluid which entails much smaller turbomachinery components for a given cycle compared to cycles using subcritical gases. Owing to the fact that the NET Power turbine receives flow at supercritical conditions, a remarkable observation on the small size of the blades in the first stages could be made, although the exact dimensions are not yet published. The investigation of the use of a radial turbine design to perform the expansion process and extract useful work, for a cycle similar to the NET Power, is considered worthy for reasons based on the performed literature survey and described in the purpose of this work. The rationales are associated to the ease and low-cost of manufacturability, robustness, compactness, and the efficiency of centrifugal configurations for reasonably high stage pressure ratio values using dense flows when compared to axial turbine counterparts.

Although the radial turbine is presented as a possible scenario for the NET Power cycle, its implementation is limited to design constraints. Radial expanders are generally efficient for small power scales, depending on the volumetric flow rate, and have not yet been thoroughly investigated with extreme blade cooling, hence why modifications to the original NET Power cycle are conducted. The alterations made are to tackle the issues that are currently infeasible to achieve with present-day materials; the turbine inlet temperature is therefore lowered to an acceptable value of 900°C.

Lowering the combustion temperature goes against the common trend of high firing temperatures but is essential in this case for obtaining an initial design of a radial turbine that could later be investigated for cooling. Indeed, there are implications linked to the removal of turbine cooling which are classified in different categories and can have either detrimental or advantageous effects to specific plant components and the overall cycle. Cycle modelling results from Chapter 4 show the lower net electric efficiency of the modified cycle compared to the original Allam cycle, 49.08% against 55%. The six percentage point deficiency is a highly significant drawback but is based on a simple one-to-one comparison of cycles without thorough thermodynamic cycle optimisation for

improvements. Cycle optimisation work done by performing sensitivity analysis on operating pressures, pressure ratios, and heat recuperation will lead to efficiency gains. At the same time however, the removal of cooling flows assist in minimising the parasitic loads of the cycle and in alleviating the penalties accredited to the heat exchange with the coolant. In other words, if advancements in metallurgy allow better strength and heat resistant properties, then turbine inlet temperatures can be increased gradually beyond, 900°C (meaning higher cycle efficiencies), without the use of cooling. In addition to that, a techno-economic study of the usage of blade cooling would lead to a better understanding of the financial and technological returns of the suggested cycle changes. The lowered power output put in place, to be able to design a relatively efficient radial turbine, does not imply a non-functional scale of work extraction, in fact there are various power plants that operate for such ranges of 60 MW electric power [140]. The cycle modelling piece of this PhD work is used as a basis to obtain sensible turbine boundary conditions for design purposes. The main points from the cycle modelling are:

- Turbine inlet temperature= 900°C
- Turbine inlet pressure= 297 bar
- Turbine pressure ratio= 4.95
- Turbine total power output= 100.7 MW_{th}/ Turbine electric power output= 67.33 MW_e
- CO₂ mole fraction= 94.06%

Guidance found in literature regarding the operating size and speeds of supercritical CO₂ turbomachinery components in general and radial turbines in specific are applicable for mostly indirect cycles, i.e. cycles coupled to external heat sources, of lower operating temperatures and pressure (lower than 900°C and 297 bar). The meanline design process of obtaining a gamut of the radial turbine meridional dimensions and the shaft rotational speed, which is outlined in Chapter 4, is essential in realising whether a radial turbine could be feasible if a design is developed. It is the first step in giving grounds for the consideration of a centrifugal configuration done by comparing rough performance quantities, such as degree of reaction and blade loading with reliable sources. In retrospect, during the meanline turbine design phase, results could not be compared quantitatively with other designs due to the absence of data for such scales and applications. Qualitative judgement based on key parameters is used with regards to sCO₂ Brayton designs, oxy-fuel designs, or with similar small-scale turbine sizes. Since the meanline design does not incorporate any loss models, it is considered as the target to which an optimised and refined turbine design is aimed at in terms of efficiency, in other words, an ideal case of a radial turbine for the modified cycle is to have an efficiency reaching 86%. The assumptions of using pure CO₂ as the working fluid is a defining limitation of the work where in reality the fluid mixture will affect the performance of the turbine design. The most important results are summarised as:

- Turbine inlet diameter ≈ 560 mm
- Turbine axial length ≈ 140 mm
- Turbine rotational speed $\approx 25,000$ rpm
- Supersonic flow expected between nozzle row and rotor inlet
- Relatively low degree of reaction with a large amount of static pressure drop happening across the stator
- Stage loading expectation of 1.40

Many researchers have performed similar processes for their turbomachinery design work either by developing in-house design tools or by using research and commercial specialised software. A possible scenario for this work could have been to create a radial turbine design tool for sCO_2 fluids by incorporating specific loss models and creating designs using blade geometry generation tools. However to satisfy the main aim of the project, a methodology (explained in Chapter 3) using existing computational tools is sufficient. Due to the lack of validation data, the radial turbine development is taken through the different design stages for cross-validation between design steps and for obtaining different modelled designs that can be compared against each other. This juxtaposition is perceived throughout the two design chapters, from preliminary design to CFD modelling, and across the Case 1 (Chapter 5) and Case 2 (Chapter 6) designs as well.

There are noticeable discrepancies between the preliminary design performance prediction and the results obtained via enhanced analysis methods of CFD for both design cases. For either case, the throughflow analysis is not capable of capturing representational values of entropy generation or performance parameters, which is most probably attributed to the use of unsuitable loss models that are not specific for real gas flows in radial turbines due to their scarcity in literature, and for the AxSTREAM bounded loss models. However, kinematic quantities of flow velocities are close to CFD results. The change in property methods used along the turbine design phase will induce marginal errors with a 5% difference in thermodynamic quantities computed from REFPROP and from AxS Carbon Dioxide. The difference with the Peng-Robinson EOS is not of great consequence because the pressure and temperature values are the parameters which the turbine is designed for. The analyses performed up to 2D CFD, although in some cases could be unreliable, are valuable because of their ease in evaluating hundreds of designs within 1000s of seconds before submission to 3D CFD. Despite not being presented in the thesis, 2D analysis has been performed for over 60 designs and 3D CFD for a select 5 other possible designs.

For Case 1, the turbine operation is far from the boundary conditions requirement and that is only perceived after incorporating viscous effects through 2D and 3D CFD modelling. The expectation of the required mass flow rate obtained from 2D and 3D CFD is

21% and 34% higher than the cycle need of 354.8 kg/s respectively. Even though the obtained total-to-total efficiency is an appreciable value of 81.04%, the increased mass flow rate due to inappropriate turbine design means that more heat input (i.e. more natural gas fuel consumption) for similar power output is required. This inevitably lowers the cycle efficiency and can have deleterious effects on the performance of other cycle components such as the heat exchanger and compressors (more compression work). The mismatching turbine operating conditions are linked to the inappropriate input definition imposed at the initial preliminary design step and that is the reason that during the revisitation of the turbine design, the range of geometric constraints and values is revamped based on remarks made in the initial design. As mentioned earlier, the initial turbine design step is still crucial for cross-examination purposes against the updated turbine design. The key points from Chapter 5 are:

- Turbine Case 1 is not capable of satisfying the cycle operating conditions
- There are major discrepancies in turbine performance results between the anticipated 1D design and CFD outcomes
- There is a region of flow separation and reversal beyond the rotor blade sharp curve angle that leads to losses
- The operation of the radial turbine under study will always be under choking conditions
- Loss mechanisms within the turbine passage are identified

The results obtained from the updated design prove that the final radial turbine operates much closer to the pressure and mass flow requirements compared to the initial design. The mass flow rate obtained from CFD simulations is only 5% higher than the design intent, in other words, if the turbine is to operate at the given mass flow of 354.6 kg/s, the total inlet pressure is also a 5% change from the initial 297 bar (with $P_{0,in} = 282$ bar). This variance is acceptable and should not affect the cycle performance tremendously; the main parameter that would however is the total-to-total efficiency of 69.87% which is significantly different from the isentropic efficiency value used in the modified cycle model, which will have a negative effect on the net electric cycle efficiency. The effect is visualised in Figure 4.1 where the expected cycle efficiency for a turbine inlet temperature of 900°C is $\approx 41.9\%$.

The low efficiency value of the non-optimised turbine design arises because of the losses associated from entropy generation mechanisms including viscous boundary layer friction, mixing, and irreversible processes of shockwaves [109]. These areas are identified from CFD results of Case 2 and when compared to those of Case 1, their expanse is much greater. The passage oblique shockwave creates an unwanted local flow compression between stator and rotor where the Mach number is greater than 1, and also generates indirect effects of boundary layer separation and diffusion. Supersonic flows are expected

from the meanline design phase because of the high pressure ratio and high density flow resulting in high loading on the turbine blade.

There are definite similarities in flow behavioural patterns (not to the same extent) between Case 1 and Case 2 that include suction surface-related flow separation or reversal that extends along the span, and rotor flow incidence losses. Tip leakage flows create a loss in stagnation pressure and lead to vortices related to the mixing of two non-identical speed flows: the leakage flow with the mainstream flow near the suction surface. Flow separation and recirculation are present in both cases at suction sides with a more pronounced region for Case 2 resulting in further friction losses that diminish the efficiency value. Also the turbulent stator wake for Case 2 has a strong influence in the mixing process with the mainstream bulk flow of the rotor. 3D CFD results for both cases allow the calculation of the non-dimensional stagnation pressure loss and entropy loss coefficients. Not implying that the design of Case 1 is suitable for the required operating conditions, it is still used for comparative reasons because of its acceptable efficiency and lack of other validation data. Table 7.1 shows that the Case 2 turbine has higher total pressure and entropy loss coefficients contrasted to Case 1. The higher rotor total pressure loss for Case 2 is apparent when comparing Figures 5.21 and 6.14 where Case 2 has more remarked regions of stagnation pressure drop in a stream section. Although the entropy rise values across both designs are similar in magnitude (73.55 J/kgK for Case 1 and 88.92 J/kgK), the specific entropy rise is used to show the relative increase in entropy for each case because they operate at different mass flow rates. The quantities listed in Table 7.1 are indications of the level of losses in the designed turbomachines but do not describe the full breakdown of losses in the radial turbine. A detailed loss contribution appraisal and calculation based on the sources is not necessary as part of this thesis.

Table 7.1 Loss indication comparison between Case 1 and Case 2 designs

		Case 1	Case 2
Nozzle stagnation pressure loss coefficient [-]	$\frac{P_{02} - P_{03}}{P_{03} - P_3}$	0.147	0.167
Rotor stagnation pressure loss coefficient [-]	$\frac{P_{04} - P_{05}}{P_{05} - P_5}$	24.21	33.97
Specific entropy rise [J/K]	$\frac{\Delta s}{\dot{m}}$	0.155	0.239

The simulations of the rotor passage only is carried out to compare it to the full turbine passage, and for the obtained results to be used in stress analysis. Standard rotor aerodynamic flow behaviour is observed with typical acceleration followed by diffusion on the suction side, and the opposite on the pressure side. Much smaller regions of loss generation are noted in comparison to the full simulated turbine. Some indispensable points taken from the CFD simulations of Case 2 include:

- Case 2 results show only 5% change compared to required cycle conditions

- 3D CFD results of Case 2 indicate a relatively inferior turbine efficiency that can be improved through blade shape optimisation
- Losses are attributed to the nozzle shockwave and the improper guidance of the flow towards the rotor inlet
- Off-design analysis shows that the design point conditions do not correspond to the highest performance and thus the design requires further refinement work
- The isolated rotor model gives a total-to-static efficiency of 80.46% which enables the deduction that most of the losses are generated from the improper stator design

The static stress analysis performed for Case 2 signals a concern for the insufficient material strength at certain areas of the radial turbine. The aerodynamic part of the turbine itself (the designed rotor), even at higher loading than design point conditions, is mostly limited to stresses that are lower than the defined allowable von-Mises stress (400 MPa). The areas of stress concentration are defined as the disc root and the blade root, both of which are not designed in detail for this work, and thus stress can be lowered. Material considerations and limitations related the performed stress work are presented in Sections 7.2 and 8.3. Below is a summary of the findings:

- Case 2 results show only 5% change compared to required cycle conditions
- For design point conditions, the rotor wheel exhibits high stress at the disc root with 988 MPa and at the rotor blade root reaching 680 MPa
- The assumed maximum allowable stress is not representative of actual current material data and thus the results should be used to obtain suitable materials in the future
- The mechanical design requires refining to reduce the stress peaks
- The range of exceeding von Mises stress values experienced by the whole rotor assembly over a range of operating conditions is between 470 MPa and 1197 MPa. The scope of displacement which occurs mostly at the rotor tip covers 0.492 mm – 0.546 mm

7.2 Enabling Technologies

With regards to the technological issues identified in Section 2.3.2, associated to the development of turbomachinery components for sCO₂ cycles, a few remarks can be made upon the outcomes presented in the thesis based on the design of the radial turbine. The outlined discussion to follow is rooted on information found in literature and on deliberations with researchers that perform work on related topics. There is a variety of matters linked to delivering the supercritical CO₂ radial turbine technology on a utility-scale

level, however only a select few are addressed here based on their importance and on the author's experience. For relevancy, counsel is given in this chapter rather than in the literature review (Chapter 2) because it is more suitable to be drawn after the design process is completed. All of the requisites of the turbine design stem from general advancement trends in other conventional gas turbines as listed in Subsection 1.2.1.

Materials Consideration

Material selection needs to be put at the forefront of the design considerations for the uncooled radial turbine for different aspects; the first for the creep and fatigue life, defined by material properties associated to the thermal and mechanical stresses and to the operating environment, and the second for the synergistic effect of environment chemistry and parameters (temperature and pressure) on the material degradation. As seen in Figure 7.1, materials have continuously evolved throughout the years to meet the high temperature operation market [148]. The proposed temperature of 900°C sits on the limit of operating temperatures that do not require cooling for turbomachinery blades; nonetheless, temperature is not the only critical limitation for the proposed work.

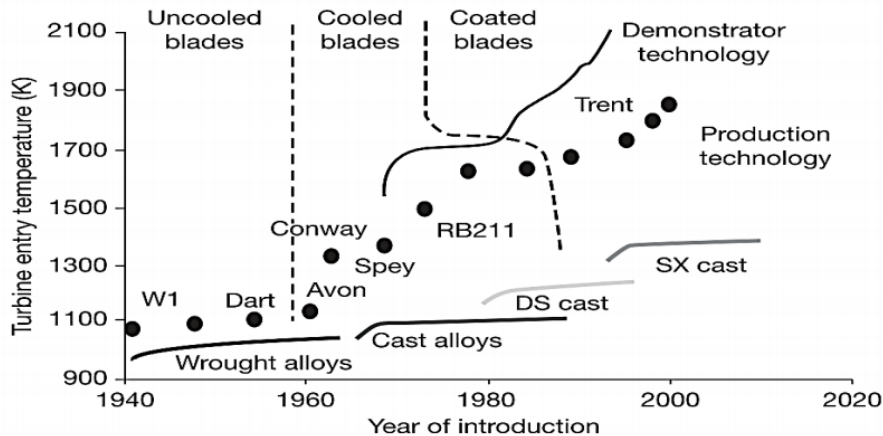


Fig. 7.1 Timeline of material evolution with increasing temperatures [148]

The large thermal gradients, based on the high thermo-physical heat transfer properties of the fluid, create thermally induced stresses for a high pressure/ high temperature radial turbine that go beyond material fatigue bounds [149]. The required strength of material at peak temperature with the mechanical stress resulting from the loads and rotational speed might dictate the use of complex single-crystal superalloys if no cooling is employed; thermal barrier coatings can be explored to enable the use of cheaper materials. Wright et al. [150] show that for the temperature limit of 900°C the use of cast-alloy IN738 for turbine vanes and blades is considered. Figure 7.2 illustrates the wide availability of materials, namely steel technology, for the application in heat exchangers of

state-of-the-art steam plants without taking corrosion testing into consideration [151]. These cast alloys could be of use in the turbomachinery casing for their high pressure capability but not in the turbine blades where aero-mechanical stresses are significant.

The high inlet temperature of the turbine would dictate the use of superalloys usually employed in land based or aeroderivative gas turbines. Developments in metallurgy to reach single-crystal superalloys and coatings allow for a better creep deformation resistance in comparison to equiaxed crystal materials. Figure 7.3 depicts some of the oxide dispersion strengthened superalloys that are used in niche high temperature aircraft gas turbine vanes [152]. The higher Larson-Miller parameter means a higher stress tolerance for thermally activated creep rupture. TMS 196 for example has a material temperature capability of 1150°C [152]. The application of such costly materials to elevated temperature sCO₂ turbine components, and the analysis of their reaction to the operating environment, have not been carried out fully, and require further investigation.

Thus, the extensive material database available for conventional gas and steam cycles is not appropriate for sCO₂ applications as observed in Figure 7.4 where a narrow selection of the high pressure- high temperature materials is limited to expensive Ni-based alloys [153]. The colours of the points in Figure 7.4 indicate the corrosion resistance of the material, where the red/ orange gradients stipulate poor performance.

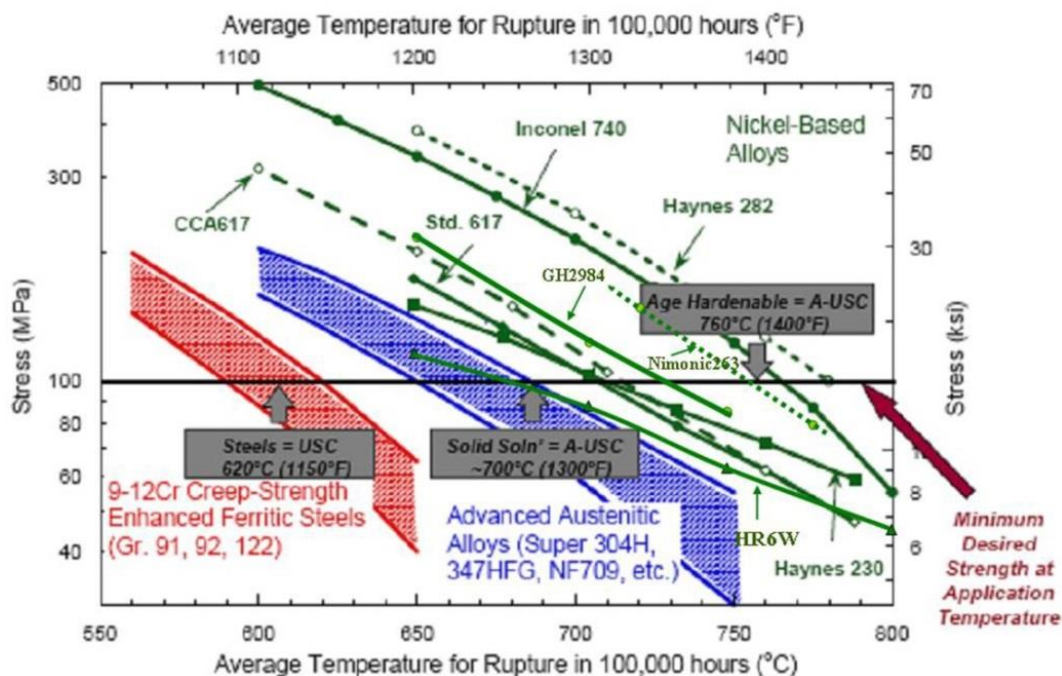


Fig. 7.2 Creep material selection for components of advanced steam power plants [151]

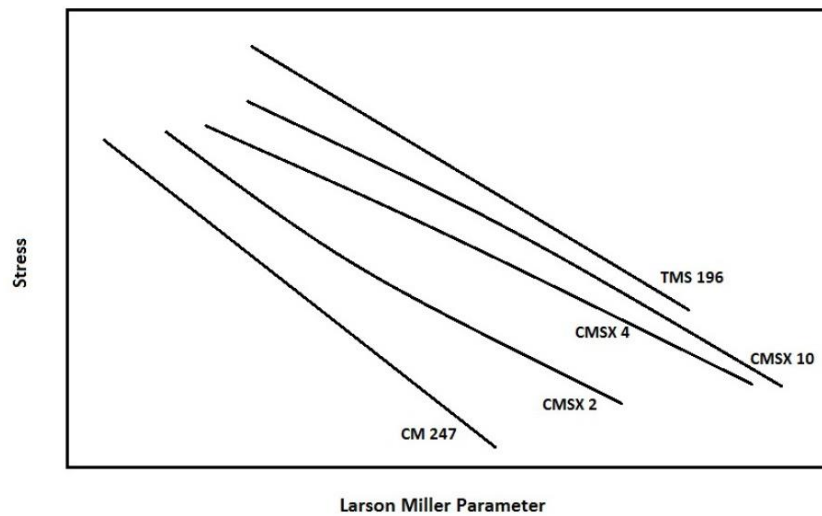
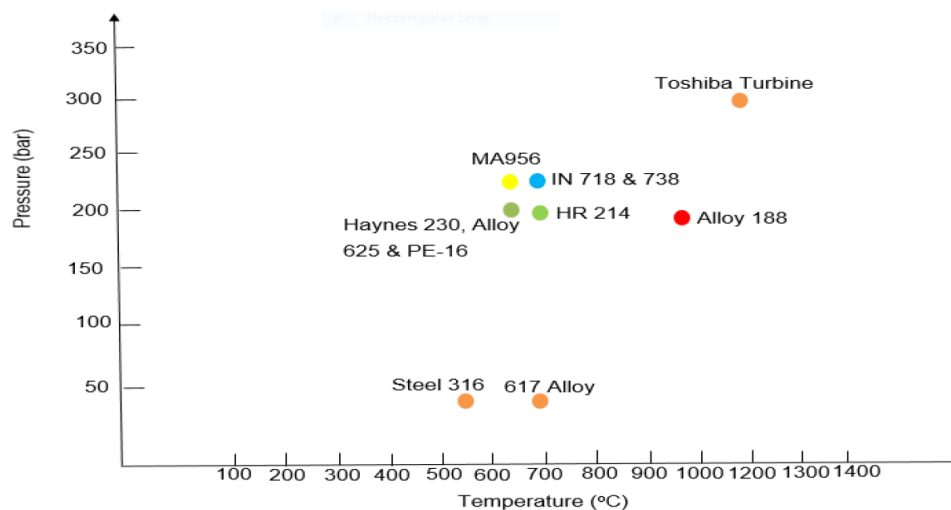


Fig. 7.3 Creep rupture improvements of superalloys [152]

Fig. 7.4 Material selection for $s\text{CO}_2$ cycle components at different operating temperatures and pressures [153]

Erosion is reported in turbines using pure supercritical CO_2 , so special attention needs to be given for erosion-resistant materials or coatings for the case of this cycle especially that the turbine operates with other contaminants coming from oxy-combustion [78]. Corrosion has been investigated for high temperature $s\text{CO}_2$ tests under different conditions visualised in Figure 7.5. Although short-term corrosion testing has been conducted by CSIRO for pure $s\text{CO}_2$ with temperatures and pressure scales up to 1000°C and 250 bar respectively [151], results are limited because of the need to obtain more adequate long-term testing with actual conditions of impurities especially when H_2O is present in the

working fluid. Within the same framework, increased corrosion has been reported with temperature rise, whereas the consequence of pressure change was found to be negligible. To mitigate the detrimental effect of internal oxidation and surface carburisation, the more expensive alloys with a high Chromium and Nickel content are necessitated; Austenitic steels have reduced corrosion resistance at high temperatures. Most oxidation and carburisation laboratory tests are also limited to ambient pressure environments with pure CO₂ fluids [150].

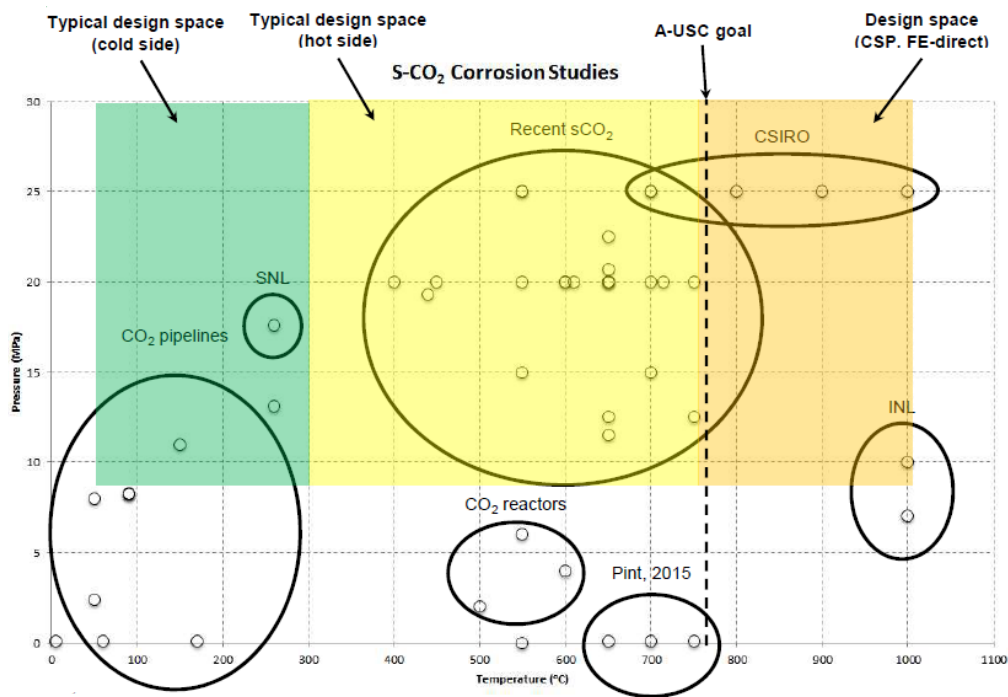


Fig. 7.5 Current sCO₂ material corrosion testing at different operating conditions [151]

From the information stated above, one can deduce that the current medley of materials still endures major testing gaps, and is not suitable for the proposed operating conditions of the turbine (900°C and 297 bar), unless cooling is incorporated. Collaborative efforts for all the material aspects is required for the actual adoption of the designed radial turbine.

Bearings

Bearings, to provide support for the relative motion between the rotating parts, are of great design importance for the actual adaptation of the proposed radial turbine. As the density of the fluid lowers from supercritical high density to subcritical conditions at the extreme pressure of the Allam cycle, a large axial force is generated on the turbine component; the

axial thrust load issue is resolved using suitable thrust bearings [98]. Mechanical contact-type bearings such as rolling, sliding or flexing elements are not considered by researchers for sCO₂ applications. Non contacting, fluid-film hydrodynamic bearings are suggested by some [154, 155] for their prevalence in high load/ high speed industrial applications. . A conclusion was obtained that oil-film bearings are not suitable for supercritical CO₂ utilisation because of the solvent-like characteristics of the sCO₂ fluid [156]. Recommendations include a specific hybrid hydrostatically-assisted hydrodynamic gas-foil bearing design, developed by Mechanical Solutions Inc. [157], that provides both radial and thrust support in a single combination component. The maximum temperature cited is up to 650°C with a material coating which limits its applicability to the design in hand if no cooling is provided. Shaft size and weight will also limit the suitability since the inner diameter of the bearing geometry is 2 inches (≈ 51 mm) which is smaller than the expected shaft diameter of 120 mm (calculated in Section 6.4). Another possibility is rare earth element magnetic bearings which could levitate for 100 MW shaft scales and are developed by Electron Energy Corporation [158]. In any case, because of the operating temperature and pressure ranges of the designed turbine, bearings and other components like gears need to be isolated from the turbine itself via shaft end seals (as discussed in the next heading). Turbomachinery design houses can provide answers regarding bearing design for the configuration of the designed turbine based on shaft sizes, but it seems that commercially-ready solutions are not yet available.

Shaft End Seals

Suitable shaft end seals for sCO₂ cycles can limit the leakage of the supercritical CO₂ fluid to minimise cycle efficiency penalties. They are required to separate the high pressure turbine from the rest of the ancillary components as shown in the generic Figure 7.6 [159] for an axial turbine. Knowing that the temperature at turbine inlet is 900°C, the seal at the end of the high-temperature/ high-pressure side needs to be placed as far as possibly permissible from the turbine inlet without affecting rotordynamics. General Electric and the Southwest Research Institute [159] concluded that labyrinth seals are insufficient for providing the required low-leakage characteristics for the high-pressure fluid because the physical clearance is optimised for a single operating condition, and thus with the change of clearance gaps during different operation modes, leakage losses are aggravated. Non-contacting, dry gas lift-off face seals are suggested for high-temperature operation, with experimental testing by Sandia National Laboratories in collaboration with Flowserve Corporation at conditions of 700°C, 4,400 psi (≈ 303 bar) and a shaft speed of 40,000 rpm [71]. In the case of the operating temperature of the modified NET Power cycle, Joule-Thomson or throttle cooling will need to be provided at that shaft end from compressor discharge to decrease the temperature to around 700°C and allow safe and effective seal operation. Commercially-ready dry gas seals are limited to shaft sizes in the range of 37.5 mm for high rotational speeds (near 50,000 rpm) and for slower speeds, they can reach up to 10 times bigger shaft diameters [154]. Looking back at the conditions of the designed

turbine in terms of temperature (900°C), pressure (297 bar), rotational speed ($\approx 20,000$ rpm) and shaft diameter (≈ 120 mm), dry gas lift off seals are not a far measure from use for a $60 - 70 \text{ MW}_e$ supercritical CO_2 radial turbine.

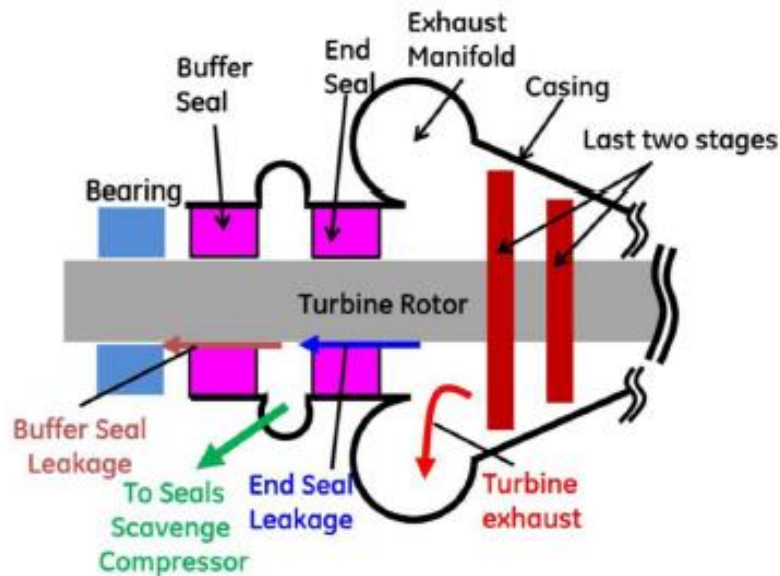


Fig. 7.6 Schematic drawing of an axial turbine with a shaft end seal, buffer seal and bearing [159]

Electrical Configuration

The turbine shaft rotational speed is in the range of 20,000 rpm which would require an appropriate choice of a step-down gearbox to speeds of 1,800 rpm or 3,600 rpm for coupling the turbine to a synchronous generator, or for the requirement of separating the unknown speed of the compressor which is not designed yet. Not only is the selection of the gearing configuration dependant on shaft speed but also on the power scale; McClung et al. [154] reports that GE has gearing configuration for power ratings reaching 60 MW. The availability of a gearbox providing a speed reduction from 12,000 rpm to 3,600 rpm for a 50 MW turbine scale is also disclosed by Bidkar et al. [160]. Geared turbine generator sets are widely available for steam turbines with power output up to 40 MW_e as reported by GE and Mitsubishi Hitachi [161, 162]. Gearbox manufacturers can provide information on the applicability of having a speed reduction ratio of around 6 from near 20,000 rpm for a $60 - 70 \text{ MW}_e$ turbine.

7.3 Turbine Design Comparison

One of the unique selling points of supercritical CO₂ turbines is their compact modular design in comparison to other power systems using conventional steam or air. The anticipation of size distinction in literature is pictured in Figure 7.8 where a sCO₂ turbine of the same power output of a steam one is expected to be twenty times smaller in size [163].

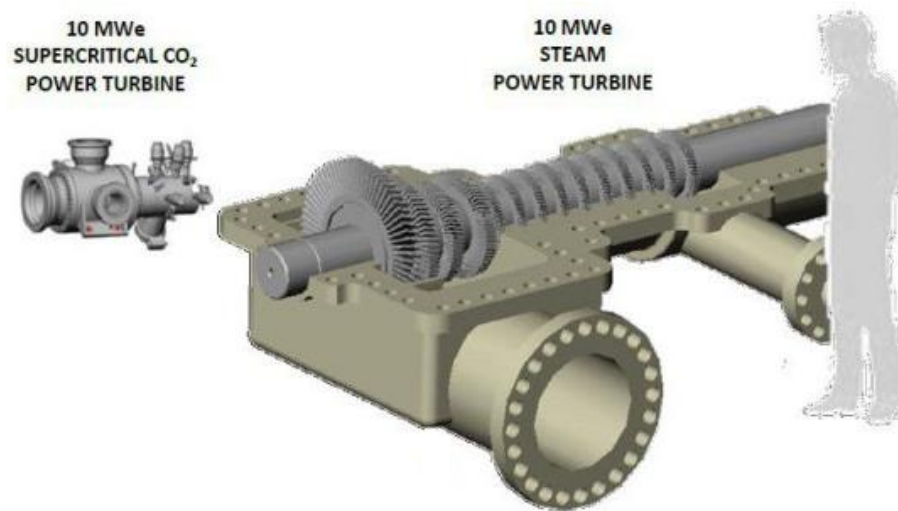


Fig. 7.7 Comparison of turbine size for sCO₂ and steam [163]

In relation to the motivation of choosing a radial turbine configuration for the generation of 60 – 70 MW of electric power, the sCO₂ designed turbine (Case 2) is placed side-by-side in Figure 7.8 with a steam turbine (9UK75 Polish steam turbine [164]) and a gas turbine (Siemens SGT-A65 [165]) of similar power output (figure not to scale though the sizes reported in Table 7.2 are those of the turbine component only not of the whole assembly of Figure 7.8). The contrasted turbines include other equipment and component parts that are not considered when comparing sizes in Table 7.2. Even though the dimensions of the sCO₂ turbine will change upon further optimisation work, the mean size will only vary by a few centimetres. The supercritical CO₂ expander has the potential of extracting high magnitudes of work in a much more compact unit. The reduced-size feature is inherently associated to high rotational speeds that will require gearing mechanisms for electrical generator integrations.

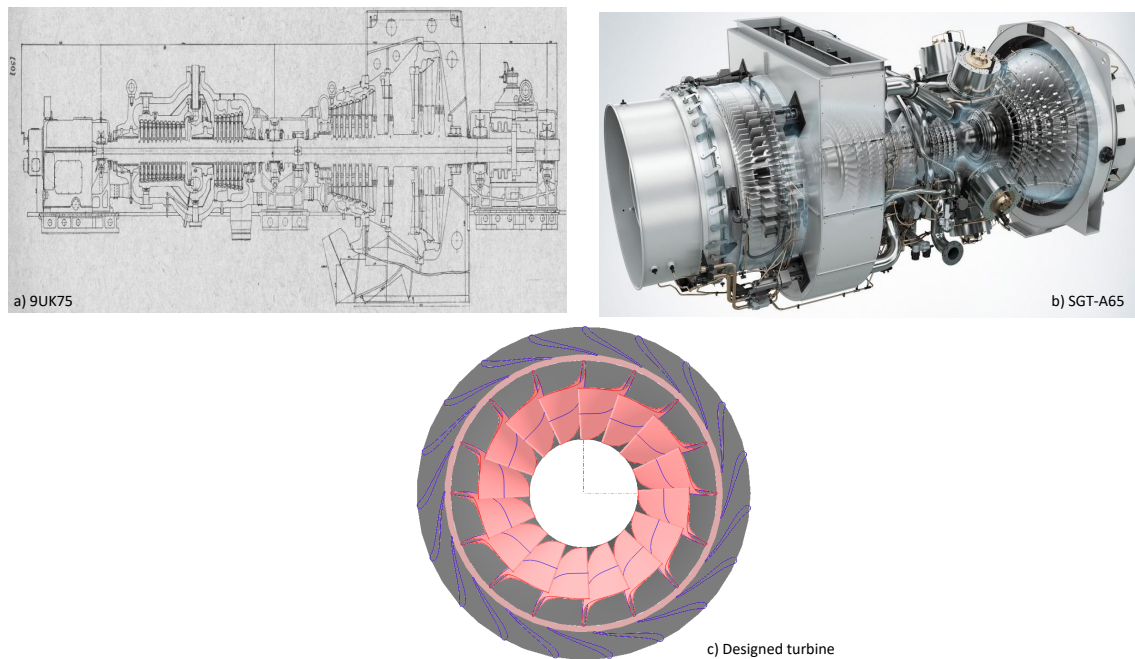


Fig. 7.8 Comparison of designed turbine against steam (9UK75) and gas (SGT-A65) turbines of similar power output [164, 165]

Table 7.2 Operating size and speed comparison of designed turbine against (9UK75) and gas (SGT-A65) turbines of similar power output [164, 165]

	9UK75	Designed turbine	SGT-A65
Power output [MW_e]	~ 75	~ 60	~ 60
Rotational speed [rpm]	$\sim 3,000$	$\sim 20,000$	$\sim 3,000$
Length [m]	~ 10	~ 0.1	~ 2
Diameter [m]	~ 5	~ 0.5	~ 3

A like-for-like comparison is not fully possible between the proposed radial turbine and the 50 MW_{th} propriety Toshiba turbine of the NET Power demonstration plant (viewed in Figure 7.9) because of the different power output and the non-revealed turbine data. The axial design developed by Toshiba is for a higher temperature and thus integrates cooling. Based on the radial turbine power limit, for scales greater than 100 MW_{th} , an axial configuration is suggested by the author. The crossover point of power scale is attributed to factors that include the easier manufacture of a small radial turbine than an axial configuration, the better performance of axial turbines for larger flow rates, and the greater experience in cooling larger axial blades.

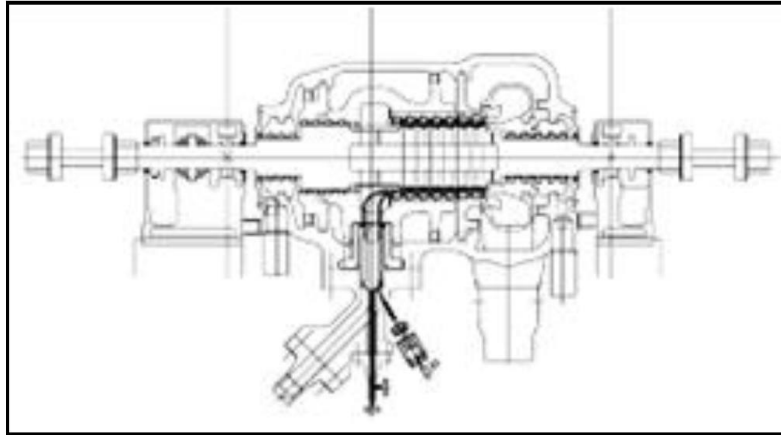


Fig. 7.9 Toshiba 50 MW_{th} turbine cross-section drawing [36]

Chapter Summary

This chapter provides an overall discussion of the results presented throughout the whole thesis. It goes into a wider, more general way of results comparison and understanding without fully drawing conclusions. A discussion of the practicalities associated for the implementation of a radial turbine is given based on key aspects relating to turbine design that require consideration, including material selection, bearings, seals, and gearboxes. The details of the development of the required enabling technologies are not provided since this is beyond the scope of expertise of the author. They are not presented as part of the future work because they are not directly related to the actual work performed during the PhD study (thermo-aerodynamic and mechanical design). From the final review of the results and the comparison with other possible technologies, it is clear that a radial turbine configuration has potential to be used as the power turbine in a mid-scale supercritical CO₂ cycle, but also that a lot of work needs to be carried out in several crucial areas to reach a commission-ready level of the design.

CHAPTER 8

Conclusions and Future Work

This chapter, concludes the thesis by summarizing the major findings. Based on the end results, recommendations for future work on the radial turbine design are proposed.

8.1 Meeting the Research Tasks

The main results of this piece of research are the outcomes achieved for the defined set of objectives to reach the principal aim. The initially identified goal is to assess the suitability of a radial turbine in a utility-scale, oxy-combustion, supercritical CO₂ power cycle. The assessment itself does not produce a final measurable quantity, but is established by creating a workflow environment for the design of a suitable radial turbine for the required cycle conditions, and by ensuring a thorough understanding of both the procedure and the performance study undertaken via different computational tools. The research findings are linked back to the research tasks, related to the objectives, outlined in Subsection 1.4.2 as follows:

Task 1. To identify the gaps in knowledge and gather relevant information by conducting a thorough literature review and communicating with experts in the field:

- Literature emphasis is on turbomachinery designs for small-scale supercritical CO₂ Brayton cycles without similar experience on mid/ large-scale cycles
- Reported work is focused on the thermodynamic analysis of oxy-combustion supercritical CO₂ cycles rather than component level turbine design
- Radial turbine designs have not been the subject of thorough investigations for utility-scale power cycles

- Turbine design considerations regarding mechanical integrity, flow behaviour and electrical integration are mostly discussed from a qualitative perspective
- Fundamental understanding of the behaviour of a supercritical CO₂ flow within a radial turbine is conducted for small power scales
- There is a lack of quantitative data for validation of varying fidelity models of turbine design applicable to such cycles

Task 2. To define an appropriate cycle model based on requirements and acquire the operating environment of the radial turbine through cycle process modelling:

An uncooled oxy-combustion, supercritical CO₂ cycle model was developed which gives a net electric efficiency of 49.08% (LHV). The required turbine operating conditions are a mass flow rate of 354.2 kg/s, an inlet temperature and pressure of 900°C and 297 bar respectively and a pressure ratio of 4.95. The computed thermal power output of the turbine is 100.7 MW_{th} and the fluid mixture is composed of 94.06% (mole fraction) carbon dioxide.

Task 3. To obtain a range of overall turbine dimensions and performance criteria by using validated design methods and recommendations found in literature:

The geometric design envelope of the radial turbine, for the selected power limit, is around 500 mm × 150 mm (diameter × axial length), and the shaft rotational speed is in the vicinity of 20,000 – 25,000 rpm. Supersonic flows with $M \approx 1.4$ are expected between near nozzle exit (because of the acceleration in the vanes) and rotor inlet.

Task 4. To gather a selection of possible preliminary turbine geometries via a systematic investigation of a number of potential turbine designs with specific constraints:

Over 200 preliminary designs were generated based on the turbine boundary conditions. Promising designs were selected based on geometrical features such as passage blockage and tip speed ratios, plus constraints related to the shape of velocity triangles and turbine performance.

Task 5. To assess different working designs based on performance requirements by means of throughflow analysis:

Throughflow streamline curvature analysis was used as a sifting step for the pool of generated designs by selecting only designs that meet efficiency (> 80%) and power criteria (> 100 MW). An indication of the possible loss mechanisms and their presence areas was obtained. This intermediate assessment step gave favourable results for both turbine cases and was used to discard various other designs.

Task 6. To study turbine flow-field and evaluate turbine performance by employing numerical methods of advanced flow modelling:

The turbine design procedure was initially illustrated with Case 1. Notwithstanding promising performance indications obtained in the intermediate steps of the design, the final configuration did not meet the design intent and required 33% increase in mass flow to achieve the required pressure drop. This datum design was however useful in highlighting the interactive process that engineers must follow in order to obtain a viable radial turbine design. The final turbine design (Case 2) showed a deviation of 5% from operating requirements but with a reduced total-to-total efficiency of 69.87%. From this relatively low efficiency, two major observations can be made: 1) there are inherent difficulties associated with the flow behaviour for such turbine designs that require major optimisation work, 2) contrary to initial belief, radial turbines might not be more efficient than axial configurations for the given turbine boundary conditions, and having multiple stages or a mixed radial-axial design might be required to achieve better performance. Regions and mechanisms of loss generation were identified to be associated with the nozzle outlet supersonic losses, the stator wake interaction with rotating blades, the improper incidence to rotor inlet, the area of separation on the rotor blade suction surface, and tip leakage flows. Off-design maps showed that the turbine runs fully at choke conditions for the range of possible cycle operation. 3D CFD results of the isolated rotor passage stipulated that many of the losses contributing to the low efficiency of the full turbine domain are generated from improper nozzle flow (incidence and shockwave losses).

Task 7. To investigate the mechanical integrity of the turbine by performing computational stress calculations:

The static stress simulations of the rotor assembly indicated that at design point, the disc and blade roots have high stress concentrations reaching extreme values of up to 1000 MPa at disc-shaft attachment. Whilst the rotor blades exhibit a von Mises stress lower than the defined maximum allowable stress of 400 MPa the blade root region experiences stresses which are around 50% higher.

Task 8. To determine areas for further work based on the availability of enabling technologies:

Comments were made on the limitations associated to the designed radial turbine in terms of material selection, seals and bearings consideration, and gearing configurations. Currently, no material is tested for the operating conditions of the turbine which poses fundamental questions regarding the availability and production cost, especially if superalloys or ceramic coating is to be utilised to inhibit material degradation. Gas foil bearings are suggested as suitable to fit the radial turbine design

but require further investigation based on cooling availability, and shaft speed and size (initial estimates show a shaft diameter of ≈ 120 mm). The same inspection is needed for the design of appropriate gas lift-off shaft-end seals.

8.2 Key Conclusions

The final observation is that a radial turbine configuration is suitable for use in a utility-scale (100 MW) supercritical CO₂ power cycle at the pressure and temperature levels of 297 bar and 900°C. The advantage of using a radial turbine is its compactness and its ability to endure a high pressure ratio in a single stage, about 5 for the designed turbine. The flow behaviour in the final turbine design aligns well with results from other works on high fluid density radial turbines. However, the current design requires re-visitation to improve the overall turbine efficiency especially to optimise the design of the nozzle vanes to reduce the associated losses. In addition to that, the current design does not meet the mechanical constraints imposed by the currently available materials at the operating environment. Nonetheless, it is crucial to note that in the aim of having better performance a radial turbine might not be sufficient on its own and a mixed radial-axial design could be necessitated.

8.3 Future Work

The approach towards achieving the goal of employing a radial turbine in a mid-scale power plant requires further work. Future tasks are to be carried out as follows:

- Incorporate suitable loss models in early design stages to avoid numerous reiterations of preliminary design steps
- Include the fluid mixture of steam, nitrogen, argon and oxygen along with carbon dioxide into all design stages for performance assessment
- Identify the mechanism of collecting the combustion gases and delivering them to the radial turbine with the absence of a volute
- Acquire better computational resources to perform more refined CFD modelling using purposefully constructed meshes and with higher calculation orders
- Work on optimising the final turbine design (Case 2) via careful geometry changes for efficiency improvements rather than redesigning and re-analysing new turbine configurations
- Create more comprehensive off-design turbine performance maps over a larger range of possible operating conditions and speeds

- Perform more rigorous stress simulations (for both nozzle blades and rotor) to obtain the necessary aerodynamic design changes that are required to lower and disperse stress concentrations. Modal, harmonic and hot-to-cold analyses are all part of the required stress investigation for the radial turbine. Attention needs to be given to improve the FEA mesh. Campbell diagrams to understand the response of the turbine blades to vibrations are also needed
- Integrate the optimised turbine efficiency and performance maps into the cycle model for more accurate thermodynamic cycle fine-tuning
- Carry out experimental testing for validation purposes
- Generate sufficient design data to ensure, via advice from manufacturers and turbomachinery design houses, so that all relevant technologies such as materials, bearings, seals, and gearing configurations can be developed

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APPENDIX A

Cycle Modelling Supplementary Results

Table A.1 Stream results of modified cycle model

Feed	T [°C]	P [bar]	$\dot{m}$ [kg/s]	CO ₂ [%]	H ₂ O [%]	O ₂ [%]	N ₂ [%]	Ar [%]
AIR-01	15	1.01325	49.49	0.04	0.97	20.75	77.32	0.92
AIR-02	286.36	7.5	49.49	0.04	0.97	20.75	77.32	0.92
AIR-03	116.84	6.75	49.49	0.04	0.97	20.75	77.32	0.92
AIR-04	54.13	6.5	49.49	0.04	0.97	20.75	77.32	0.92
NG-01	15	70	2.96	NG mole fractions				
NG-02	143.83	300	2.96	NG mole fractions				
OXI-01	17.40	120	108.95	84.98	0.10	13.34	1.05	0.53
OXI-02	46.83	303	108.95	84.98	0.10	13.34	1.05	0.53
OXI-03	109.70	302	108.95	84.98	0.10	13.34	1.05	0.53
OXI-04	614.50	300	108.95	84.98	0.10	13.34	1.05	0.53
OXY	15	120	10.99	-	-	99.5	0.2	0.3
STO-01	26	80	7.93	98.00	0.11	0.13	1.19	0.57
STO-02	33.46	111	7.93	98.00	0.11	0.13	1.19	0.57
STO-03	26	111	7.93	98.00	0.11	0.13	1.19	0.57
WAT	26	58.5	6.02	0.05	99.95	-	-	-
WF-01	900	297	354.19	94.06	4.13	0.12	1.14	0.55
WF-02	679.03	60	354.19	94.06	4.13	0.12	1.14	0.55
WF-03	116.84	59.25	354.19	94.06	4.13	0.12	1.14	0.55
WF-04	54.13	59	354.19	94.06	4.13	0.12	1.14	0.55
WF-05	26	58.5	354.19	94.06	4.13	0.12	1.14	0.55
WF-06	26	58.5	348.18	98.00	0.11	0.13	1.19	0.57
WF-07	38.21	67.84	348.18	98.00	0.11	0.13	1.19	0.57
WF-08	26	67.34	348.18	98.00	0.11	0.13	1.19	0.57
WF-09	39.39	81	348.18	98.00	0.11	0.13	1.19	0.57
WF-10	26	80	348.18	98.00	0.11	0.13	1.19	0.57
WF-11	26	80	340.24	98.00	0.11	0.13	1.19	0.57
WF-12	35.67	121	340.24	98.00	0.11	0.13	1.19	0.57
WF-13	26	120	340.24	98.00	0.11	0.13	1.19	0.57
WF-14	26	120	242.28	98.00	0.11	0.13	1.19	0.57
WF-15	50.44	303	242.28	98.00	0.11	0.13	1.19	0.57
WF-16	109.70	302	242.28	98.00	0.11	0.13	1.19	0.57
WF-17	614.50	300	242.28	98.00	0.11	0.13	1.19	0.57
WF-O2	26	120	97.96	98.00	0.11	0.13	1.19	0.57
CW-in	15	2	adjusted	-	100	-	-	-
CW-out	25	2	adjusted	-	100	-	-	-

Table A.2 Cooling water requirements for modified cycle

Heat exchanger	Cooling water mass flow rate [kg/s]
COND	388.38
IC-01	272.33
IC-02	902.94
IC-03	235.13
ST-IC	4.60

APPENDIX B

Meanline Results of Turbine with Fluid Mix

Table B.1 Initial parameters for turbine with fluid mix

Variable	Unit	Value
$\dot{Q}$	m^3/s	11.43
ω	rpm	24224
n_s	-	0.624
C_{0s}	m/s	811.4
v_s	-	0.555
η_{ts}	%	86.39
ϕ	-	0.488
ψ	-	1.40
R	-	0.322

Table B.2 Initial rotor data for turbine with fluid mix

Variable	Unit	Value
N_R	-	13
t_4	mm	7.107
q_4	mm	85.87
o_4	mm	61.48
A_5	mm^2	58610
t_5	mm	3.553
q_5	mm	48.31
o_5	mm	31.28

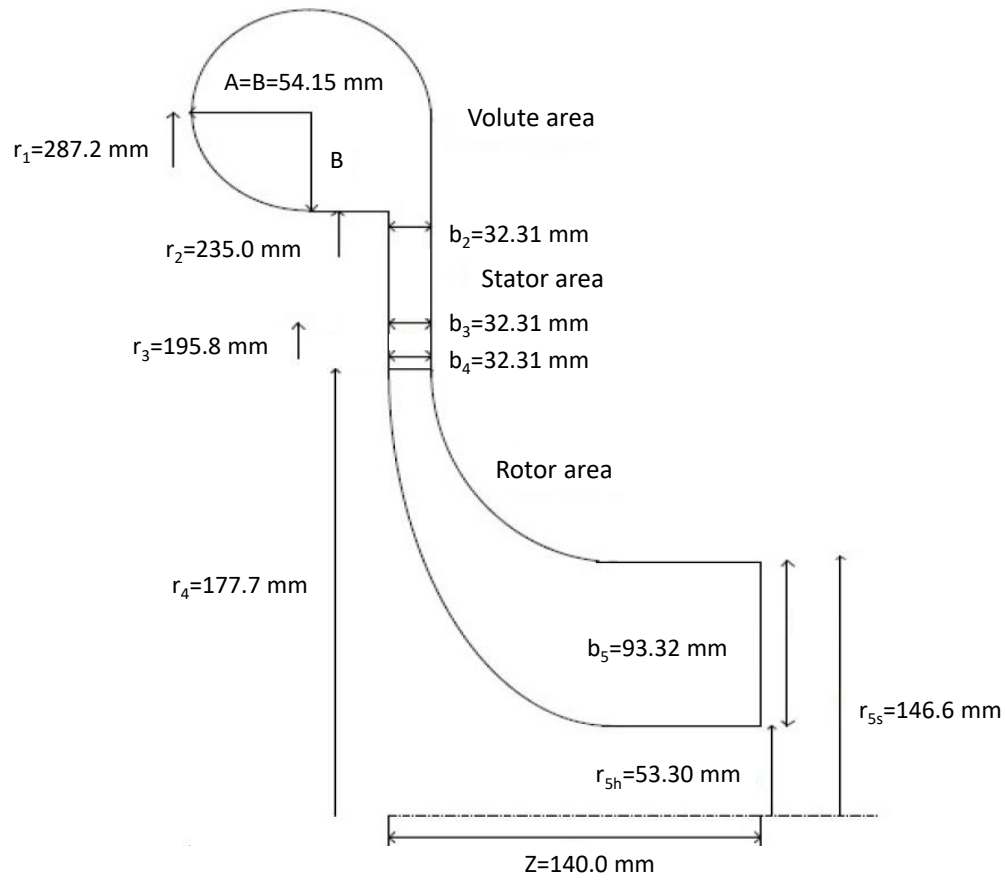


Fig. B.1 Initial turbine meridional dimensions

Table B.3 Thermodynamic properties data at turbine stations with fluid mix

Variable	Unit	Volute inlet (1)	Volute outlet/ Nozzle inlet (2)	Nozzle outlet (3)	Rotor inlet (4)	Rotor outlet (5)
T_0	K	1173	1173	1173	1173	952
P_0	bar	297	297	284	284	60.0
h_0	kJ/kg	1593	1593	1593	1593	1309
ρ_0	kg/m ³	117.5	117.5	112.7	112.7	28.05
s	kJ/kgK	33.83	33.83	33.92	33.92	34.47
T	K	1116	1080	1038	1008	934
P	bar	217	177	132	110	53.2
h	kJ/kg	1517	1470	1416	1377	1286
ρ	kg/m ³	92.20	78.40	61.51	53.12	28.05

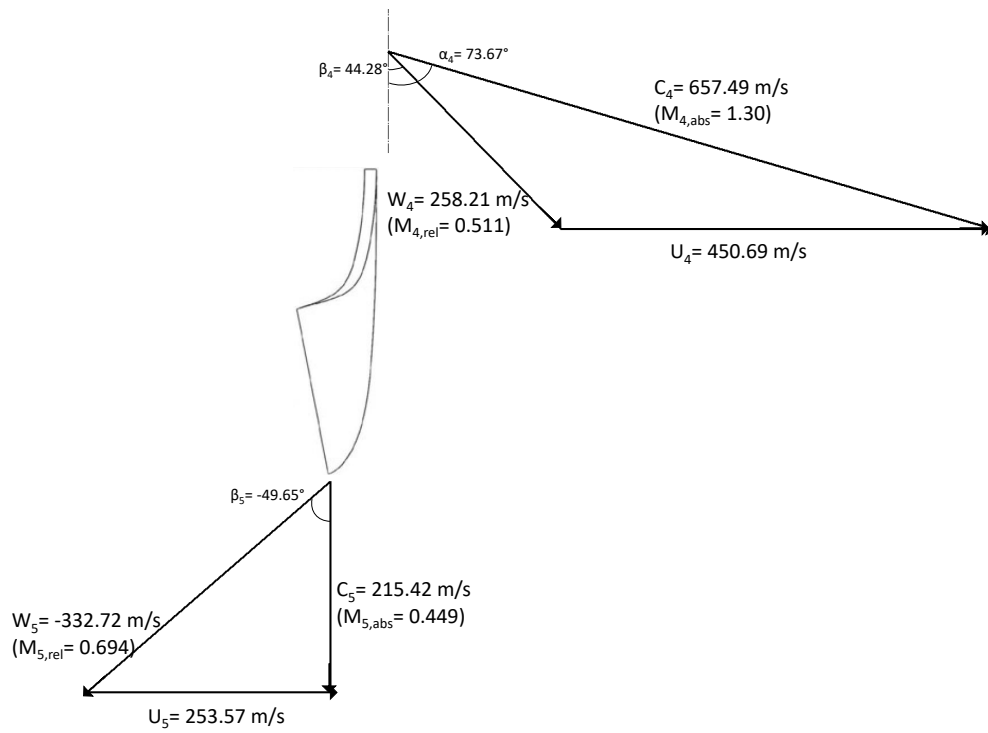


Fig. B.2 Initial velocity triangles for turbine with fluid mix

APPENDIX C

Case 1 Turbine 3D CFD Details

Table C.1 3D CFD results of thermodynamic properties at turbine sections for Case 1

Variable	Unit	Nozzle inlet	Nozzle outlet	Rotor inlet	Rotor outlet
T_0	K	1173	1175	1182	968
P_0	bar	298	276	284	64.1
h_0	kJ/kg	1480	1483	1491	1222
s	J/kg K	31.11	31.30	31.33	31.85
T	K	1171	1045	1041	944.5
P	bar	294	129	124	55.0
h	kJ/kg	1478	1316	1312	1194
ρ	kg/m ³	123.8	63.25	61.41	30.50

Table C.2 3D CFD results of kinematic data at turbine sections for Case 1

Variable	Unit	Nozzle inlet	Nozzle outlet	Rotor inlet	Rotor outlet
C	m/s	70.77	575.3	591.5	236.6
W	m/s	-	-	209.5	372.3
U	m/s	-	-	477.3	282.0

APPENDIX D

Case 2 Turbine 3D CFD Details

Table D.1 Performance results for 3D CFD mesh sensitivity study of Case 2 using two different boundary condition definitions

Boundary condition: $\dot{m}_{in}$ & P_{out}					
Number of elements [-]	P_{0in} [bar]	$\dot{W}$ [MW]	η_{tt} [%]	η_{ts} [%]	Δs [J/kg K]
51,110	224	83.97	72.66	67.50	91.12
209,308	281	93.16	73.48	67.98	86.21
971,613	282	102.2	69.77	63.17	86.40
Boundary condition: P_{0in} & P_{out}					
Number of elements [-]	$\dot{m}$ [kg/s]	$\dot{W}$ [MW]	η_{tt} [%]	η_{ts} [%]	Δs [J/kg K]
209,308	375.2	101.3	73.01	67.21	90.02
962,874	372.5	108.5	69.87	63.05	88.92

Table D.2 3D CFD results of thermodynamic properties at turbine sections for Case 2

Variable	Unit	Nozzle inlet	Nozzle outlet	Rotor inlet	Rotor outlet
T_0	K	1173	1174	1157	976
P_0	bar	297	271	324	62.86
h_0	kJ/kg	1480	1481	1523	1233
s	J/kg K	31.11	31.32	31.35	32.00
T	K	1171	1031	1025	952.9
P	bar	294	117	113	54.0
h	kJ/kg	1477	1299	1293	1205
ρ	kg/m ³	123.7	58.41	56.67	29.68

Table D.3 3D CFD results of kinematic data at turbine sections for Case 2

Variable	Unit	Nozzle inlet	Nozzle outlet	Rotor inlet	Rotor outlet
C	m/s	70.21	601.43	647.4	236.7
W	m/s	-	-	261.42	374.32
U	m/s	-	-	461.8	273.8