

CENTRAL UNIVERSITY OF TECHNOLOGY

MASTERS DISSERTATION

Characterisation of additive
manufactured small radial flow
Ti-6AL-4V turbines for application in
Organic Rankine Cycles

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for the degree: Master of Engineering in Mechanical Engineering
in the*

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Declaration of Authorship

I, Max E. COGHO, declare that this dissertation titled, “Characterisation of additive manufactured small radial flow Ti-6AL-4V turbines for application in Organic Rankine Cycles” and the work presented in it are my own. I confirm that:

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- Where any part of this thesis has previously been submitted for a degree or any other qualification at this University or any other institution, this has been clearly stated.
- Where I have consulted the published work of others, this is always clearly attributed.
- Where I have quoted from the work of others, the source is always given. With the exception of such quotations, this dissertation is entirely my own work.
- I have acknowledged all main sources of help.
- Where the dissertation is based on work done by myself jointly with others, I have made clear exactly what was done by others and what I have contributed myself.

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Abstract

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Master of Engineering in Mechanical Engineering

**Characterisation of additive manufactured small radial flow Ti-6Al-4V
turbines for application in Organic Rankine Cycles**

by Max E. COGHO

Due to the use of fossil fuels globally, there has been a significant increase in greenhouse gas emissions. The increase in greenhouse gas emissions should be limited as much as possible. More efficient use of fossil fuels can assist with curbing this increase in greenhouse gas emissions.

One method to increase efficiency of machines that use fossil fuels, is to recover some of the waste heat and convert it into energy. The Organic Rankine Cycle can be used for this purpose. For smaller scale applications the turbines used in these applications can be quite difficult to manufacture.

Using additive manufacturing, turbines can be created for use in small scale Organic Rankine Cycles. Using additive manufacturing, three turbines were manufactured from titanium alloy Ti-6Al-4V. These turbines were tested experimentally to characterize their performance.

The turbines were tested at off design conditions due to limitations to the compressed air supply. Additive manufacturing provides an effective method to produce small scale turbines for use in Organic Rankine Cycles.

Keywords:

Additive manufacturing, Ti-6Al-4V, selective laser melting, selective laser sintering, radial inflow turbine, 90° radial inflow turbine, radial inflow turbine design, radial inflow turbine characterisation, Organic Rankine Cycle. . . .

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List of Abbreviations

AM	Additive Manufacturing
Al	Aluminium
CO₂	Carbon Dioxide
CHP	Combined Heat and Power
CFD	Computational Fluid Dynamics
DAQ	Data Acquisition
DC	Direct Current
DMD	Direct Metal Deposition
DMLS	Direct Metal Laser Sintering
EES	Engineering Equation Solver
FS	Full Scale
GHG	Green House Gas
ICE	Internal Combustion Engine
ORC	Organic Rankine Cycle
PFD	Process Flow Diagram
SLM	Selective Laser Melting
SLS	Selective Laser Sintering
Ti	Titanium
UTS	Ultimate Tensile Stress
V	Venadium

Physical Constants

Gravitational acceleration	$g = 9.81 \text{ m/s}^2$
Isentropic expansion index of air	$\gamma = 1.4$
Molar mass of air	$M_{air} = 29.87 \text{ g mol}^{-1}$
Specific gas constant of air	$R = 287 \text{ J kg}^{-1} \text{ K}$
Universal gas constant	$R_o = 8.3145 \text{ J mol}^{-1} \text{ K}$

List of Symbols

Roman letters

Symbol	Description	Units
A	area	m^2
a	speed of sound	m s^{-1}
c	absolute velocity of working fluid	m s^{-1}
c_x	axial velocity of working fluid	m s^{-1}
c_r	relative velocity of working fluid	m s^{-1}
f	force	N
h	specific enthalpy	kJ kg^{-1}
h_0	specific stagnation enthalpy	kJ kg^{-1}
M	Mach number	dimensionless
$\dot{m}$	mass flow rate	kg s^{-1}
m	mass	kg
n	rotational speed	rev s^{-1}
P	power	W (J s^{-1})
p	pressure	Pa
Q	flow rate	m^3/s
r	radius	m
s	entropy	kJ K^{-1}
s_0	stagnation entropy	kJ K^{-1}
t	temperature	$^{\circ}\text{C}$
T	temperature	K
u	velocity of rotor	m s^{-1}
w_r	whirl velocity of working fluid	m s^{-1}

Greek Letters

Symbol	Description	Units
α	absolute angle of working fluid	$^{\circ}$
β	rotor blade angle	$^{\circ}$
η	efficiency	%
η_{ts}	total to static efficiency	%
ρ	density	kg/m^3
τ	torque	N m
ω	angular velocity	rad s^{-1}

Chapter 1

Introduction

1.1 Background

1.1.1 Recovery of waste heat

Globally fossil fuels are used to power industry and drive economic growth. It is an integral part of everyday life, used for personal transport, powering most homes and businesses. Therefore, it is important to use fossil fuels as efficiently as possible. This will extend the time fossil fuels are available for use as a power source, as well as reduce the impact of global warming.

The Organic Rankine Cycle (ORC) is a variation of the traditional Rankine cycle that uses organic fluids as the working medium instead of water. These organic fluids typically have a lower boiling point and higher molecular mass, making them suitable for converting low-grade heat sources into electricity [1]. The superheated working fluid is then expanded through a turbine to provide rotational energy. This energy then drives the ORC feed pump as well as an electrical generator. The feed pump pressurises the fluid to an original working state after it has been cooled down by a condenser that removes the excess heat [1].

According to Sprouse [2]: The ORC is a closed-loop cycle that functions similarly to the conventional Rankine cycle with the main difference being the working fluid. The standard working fluid in a Rankine cycle is water, where in the Organic Rankine Cycle organic fluids are used instead. This is to utilise the lower grade heat that is supplied in the exhaust gasses, where water would not be effective or viable.

When the temperature differences in the process are not sufficient, water (steam) becomes less effective because it is susceptible to losses such as shock losses [3]. An organic fluid has a higher expansion ratio for the same conditions and would lead to a more efficient turbine [1], [4].

1.1.2 Turbines

While there has been a lot of focus on systems that recovers heat on larger scale, more work can be done on systems that can be used in smaller applications such as, car exhaust gasses and small businesses that produce low to medium heat exhaust gasses. These gases can be used to power ORCs to improve efficiency and reduce the consumption of fossil fuels.

However, to produce these small ORC systems it requires small and complex parts that can be extremely difficult or expensive to manufacture conventionally. An important part of an Organic Rankine Cycle is the turbine, which generally consists

of inlet nozzle/guide vanes, a rotor and a casing. These turbines have been studied significantly on larger scales and applications where AM might not necessarily be such an attractive option. The goal is to design a small turbine that can be used in much smaller ORC systems.

A small radial inflow turbine is a type of turbine where the working fluid flows radially inward towards the center of the turbine before exiting axially. These turbines are particularly effective for applications involving low to medium power output and are often used in small gas turbines and turbochargers. They are known for their simple and robust design, which allows for efficient energy extraction even in compact systems. Radial inflow turbines have performance characteristics similar to centrifugal compressors but operate in reverse, converting fluid energy into mechanical work. Hence the radial inflow turbine is an appealing option to use in a ORC system because it can also be easily adapted for different working fluids [5]. It is not as sensitive as other turbines to the profile of the blades, thus turbines with simpler blade profiles can be created [6]. The blade profiles can also be made more robust and therefore be run at higher speeds. Radial flow turbines are simpler to manufacture when compared to axial flow turbines [7].

1.1.3 Production method

Additive manufacturing (AM) is an attractive method to produce intricate and small components, where conventional machining would be too expensive or impractical [8]. It can use a wide variety of different materials and alloys to create components with high accuracy. AM, especially laser sintering, allows for very precise control over geometry, mechanical and thermal properties, by altering the laser conditions [9].

There are some limitations on using AM, due to the process of melting a powder to form a shape. The material can flow or sag undesirably. Therefore, if an angle is too steep or an overhang is too large, supports need to be produced during the production process. These supports must be removed after production, via conventional machining. The high heat used to melt the powder also causes thermal stress in the product, therefore proper annealing is required to relieve stresses after production to ensure the product is not brittle. Further the surface finish of a product produced with AM will differ to that of a conventionally produced product, with the surface roughness typically higher than when conventional machining is used.

By using AM to create a small radial turbine, this study can further explore and determine possible applications or situations for using AM. Determining the efficiencies on the turbines, the effect on the surface roughness from a part produced by AM can be correlated and used for further development. Another benefit of using AM is in the reduction of parts needed to be produced as well as a reduction in weight by being able to print more complex structures [10].

1.1.4 Materials for small scale turbines

Titanium (Ti) has a low thermal conductivity and a high resistance to corrosion. The elastic modulus and specific strength are high compared to most metals. Ti-6Al-4V is used for compressor discs, blades, and stators, due to it having a high ultimate tensile strength (UTS) and Young's modulus at high temperatures. The above mentioned

properties makes Ti-6Al-4V an ideal candidate material to create turbines for the ORC, using AM [11].

1.2 Motivation

Most of industry is heavily dependent on energy generated by fossil fuels. However, significant amounts of energy is wasted either during or after the combustion process or industrial operation via cooling or exhaust gasses released into the atmosphere. Most internal combustion engines (ICEs) and industrial operations run on fossil fuels which cause pollution, greenhouse gas emissions and higher cost due to the wasted energy.

Heat from the exhaust gasses can be recovered and used to create/generate more power from the same amount of fuel or reduce the fuel needed. This can be achieved using an ORC [12]. The ORC should be able to use the heat from the exhaust gasses, heating a specific hydrocarbon working fluid via a heat ex-changer. This heated fluid can then be expanded through a turbine that can then be used to power an external component as well as the pump to keep the system operating.

Using conventional tooling and machining methods it is difficult to produce small radial flow turbines because of the size and profiles needed on the blades [13]. AM seems to be a reasonable approach to manufacture these components, especially when heating and/or cooling channels are to be embedded within the turbine vanes. Embedding of cooling and heating channels is planned for a follow up study. By making use of AM it is possible to manufacture more complex and smaller parts at a reduced price, since the complexity of a part does not necessarily increase the production cost or time of a component [14].

Ti-6Al-4V is an attractive material to use for a small radial flow turbine. The material and process will greatly influence the performance characteristics of the turbine and must be known before characterizing/developing a turbine. This specific alloy is suitable for the application due to its high specific strength. The heavier the material is the larger centripetal forces it will experience when rotating at high speeds, therefore it is important for the material to have a high strength to weight ratio.

There are however some questions about the performance of the components that are created using AM to produce a small radial flow turbine. One such question would be the effect of the surface roughness of the AM produced component on the turbine. In order to create a system that is effective and can be reproduced, it is important to know the characteristics of all of the different components within the system. This would assist to accurately calculate the energy recovered and adapt to different sizes according to power available from the gasses.

1.3 Research aim and objectives

This research will focus on developing correlations between theoretical turbines and small turbines produced by additive manufacturing and the characterization of these turbines for eventual application in Organic Rankine Cycles.

Objectives:

Conduct a research study that will entail the following elements:

- Perform a comprehensive literature study to determine the state of the art.
- Design a 90° radial inflow turbine that is suitable to be manufactured from Ti-6Al-4V with the aid of AM.
- Design an experiment, test bench and associated test model for AM turbines.
- Conduct an experimental characterization of the AM turbines.
- Document the performance characteristics and compare with the theoretical design.
- Create a basis for future research and development of additive manufactured small radial turbines for the use in the ORC.

1.4 Scope

The type of turbine also plays a significant role in the design of Organic Rankine Cycles, depending on the turbine-specific speed and pressure available from the working fluid. This study looked specifically at 90° inward-flow radial turbines. This was an attractive option due to the size of turbine that was to be created and the specific speeds at which these turbines would generally be used. The goal was to make this turbine small while still being able to recover a significant amount of energy.

A small 90-degree radial inflow turbine is characterized by the following features:

Flow Direction: The working fluid enters the turbine radially and makes a 90-degree turn to exit axially. This change in flow direction is a defining characteristic of radial inflow turbines [1].

Design Efficiency: These turbines are designed to efficiently handle low to medium power outputs and are known for their compact and robust design. They can achieve high efficiency in small mass-flow rate situations, making them ideal for small-scale applications like micro gas turbines and turbochargers [2].

Components: Key components include the rotor, stator, and the turbine blades. The rotor and blades are designed to manage the dynamic changes in the direction and velocity of the fluid flow [3].

Application Range: They are suitable for a variety of applications, including waste heat recovery systems, Organic Rankine Cycles (ORC), and other low-temperature power generation systems due to their ability to efficiently convert thermal energy into mechanical work [4].

Performance: The performance characteristics of these turbines include a high degree of adaptability to different operating conditions and the capability to handle a wide range of power outputs and rotational speeds [5].

Specifically, three Ti-6Al-4V 90° radial in-flow turbines were designed. The rotor blades were produced using selective laser sintering which is a type of AM also known as 3D printing. Similarly, three guide vanes were produced using the same method. The casing was produced using regular mild steel and conventional machining methods. A

custom test bench was constructed for testing and characterization of the turbine.

One of the limiting constraints with regard to AM was the minimum required thickness of the rotor blades.

The different guide vanes and rotors were produced to characterize the differences between the turbines. An open loop system was used to experimentally test these turbines using air as the working fluid. However, due to the low-pressure ratios the turbines did not operate at the design point. Thus, limiting the influence on performance of the different design configurations.

A computational fluid dynamics (CFD) analysis will not be included in this body of work. An initial 1-D design was produced to facilitate experimental determination of the performance characteristics of the additively manufactured 90° IFR turbines. In a follow-up study CFD should be employed to optimise the turbine further.

1.5 Structure

There are a total of six chapters in this dissertation. Chapter 1 gives a brief overview of the study as an introduction, which is followed by a literature review in chapter 2. Chapter 3 discusses the methodology used to design the turbines used in the experiment. Whereas chapter 4 covers the experiment and experimental methodology. In chapter 5 all the data captured is reworked and presented to characterize the turbines. There is also a brief look at the potential performance of the turbines at higher pressure ratios and when using popular organic working fluids in stead of air. Chapter 6 contains a summary and the conclusions drawn from this study while also providing some lessons learned which can be utilized in future studies.

Chapter 2

Literature review

2.1 Introduction

2.2 Influence of fossil fuels usage

2.2.1 Use of fossil fuels and sustainability

Since the start of the industrial era there has been an increase in the production of carbon dioxide (CO_2) [15]. This is due to the increase in the use of energy. Larger factories as well as electricity in each home since then has kept the demand growing. With the increase in demand, extraction and the burning of fossil fuels has increased to supply sufficient energy [15]–[17].

According to Höök M and Tang X, [18] a large portion of today's energy is produced from fossil fuels. They reported that approximately 80% of energy produced in 2011 was from fossil fuels; being mostly oil, natural gas and coal. Oil constitutes the largest portion of these fossil fuels at 32.8%, coal 27.2% and natural gas 20.9%. These fossil fuels are vital to the global economy as it powers most of our industry and homes.

Despite significant advancements in renewable energy technologies, fossil fuels still constitute a major portion of the global energy mix. According to the latest data, fossil fuels account for approximately 82 - 84% of global energy production [19].

Fossil fuel consumption is continuously rising. The global energy consumption for oil, coal and gas in 2023 was: oil: 196 exajoules/yr (31.6%), coal 164 exajoule/yr (26.5%) and natural gas 1404 exajoule/yr (23.2%) compared to the combined wind, solar and hydropower consumption of 91 exajoules/yr (14.7%) [19], [20].

The Intergovernmental Panel on Climate Change (IPCC) of the United Nations reported in 2022 [21]. The largest constituent of the cause for greenhouse gasses is industry at 25% followed closely by electricity and heat at 23%. Agriculture is believed to contribute 22%, with the remaining 15%, 10% and 5,6% being caused by transport, other energy and buildings.

This leads to a further problem that the increased use of fossil fuels as source of energy is not sustainable [22]. According to British Petroluem company at the current trends for the usage of fossil fuels, coal is expected to last only 114 years, natural gas 52.8 years and oil 50.7 years [23]

Looking at more recent findings from 2019 [24], global energy consumption is still on the rise. Although renewable energies compensate for some of the increased demand

net carbon emissions are still on the rise. This is expected to continue until 2040. The report further illustrates that there is still growing demand for fossil energy sources like coal and natural gas to keep up with this increasing demand. Due to both the increasing demand in energy as well as the reliance on fossil fuels for the foreseeable future efficient use of energy should still be a priority.

2.2.2 Impact of fossil fuels on global warming

Since fossil fuels are the major source of energy globally, a very large part of the anthropogenic greenhouse gas (GHG) emissions are a direct result thereof [18], [25]. To keep up with global energy demand more fossil fuels were extracted and used. This leads to continually increasing the production of CO₂ and releasing it into the atmosphere [26].

From 1850 to 2012 there has been an increase in CO₂ of between 1.1% and 4.7% annually, with 1.1% being the lowest period from 1913 to 1945 and the highest being 4.7% from 1945 to 1973. From 2000 to 2012 there has been an increase of 3% yearly [27]. This leads us to believe that at current trends CO₂ will continue to increase [27], [28].

This increase has led global temperatures to be 1°C higher for the past decade and 1.3°C for 2016, than the Holocene (current geological epoch) temperature range [29]. The global temperature has increased with roughly +0.18°C more than the natural Holocene expected increase [29]. By 2040 it is expected for the arctic temperature range to have increased by 1.5 to 2.5 °C which will have a drastic impact on the global climate [28].

Although natural phenomena is also partly responsible for some of the increase in global temperature, it is plausible that warming caused by human activities can be accountable for an increase in roughly 1 °C from pre-industrial era until 2017. This increase is believed to be increasing at approximately 0.2 °C per year. This 20% it is a significant increase that can and will have significant impact on the environment [30].

The more recent International Plant Protection Convention (IPPC) in 2022 further found that with current policies that are in place it appears as though greenhouse gasses will continue to rise [21]. This is expected to increase the global temperature between 2.4°C to 3.5°C by the year 2100 [21]. Therefore, it is clearly a goal of the IPPC to improve and implement policies to help mitigate this expected increase in CO₂, which in turn would mean adoption of new technologies as well as improving the efficiencies of current energy consuming systems.

2.2.3 Wasted energy in processes

In order to extend the availability of fossil fuels for use as a source of power and reduce the effect it has on global warming there are a couple of options available. Firstly, we can improve the efficiency of systems using fossil fuels. Another option is to look to alternative means of producing power that are more sustainable, like solar power, hydro power, wind power, etc. Currently approximately 26.5% of energy is produced using renewable sources. With the main contributor being hydropower at 16.4% [31]

Wasted power seems like a good place to start as most of the industries already has systems in place that produce waste energy of some sorts. This makes it reasonable to

improve the efficiency of certain processes, as the infrastructure is mostly there. This means that a system can be altered in such a manner that improves the efficiency in such a manner that is more cost effective than to build alternative means of power. First it must be determined where energy is lost, before it can be attempted to improve efficiency.

One such an example is the internal combustion engine (ICE). ICEs are mostly powered by either diesel or petrol that is produced from crude oil. The process of internal combustion produces a significant amount of waste heat that is lost through the cooling system, the exhaust gasses and radiation [32].

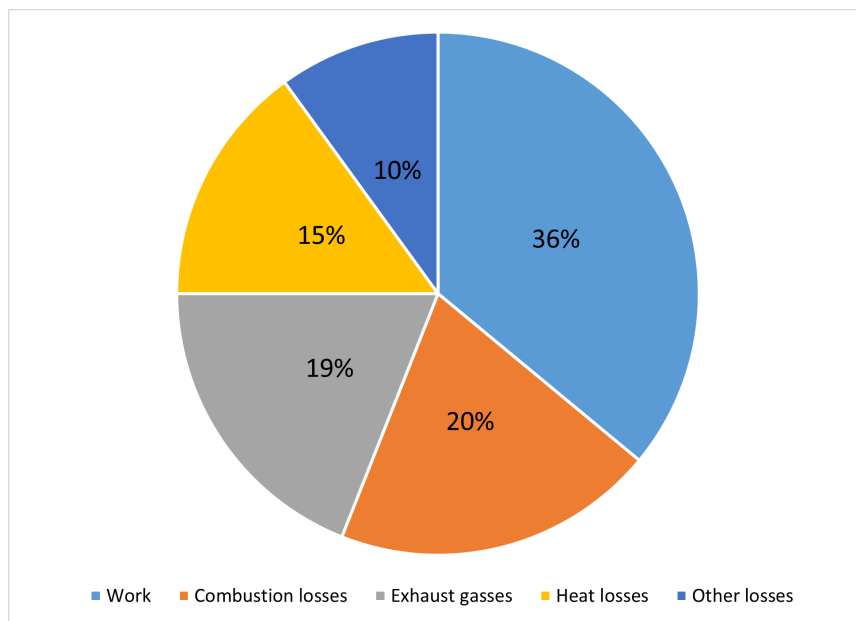


FIGURE 2.1: Energy lost in an ICE [33]

The split of energy used and lost in the combustion process is depicted in figure 2.1 [34]. A significant portion of heat-energy expelled from the exhaust gasses is currently not utilised. This wasted heat-energy is a potential source of energy to increase performance and efficiency.

A more in depth and recent study on the efficiency of combustion engines by Albatayneh et. al. [35] shows that for gasoline, diesel and gas powered vehicles the efficiency from the source to the wheel is 14-33%, 25-37% and 12 -22% respectively. This further emphasises the opportunity to harvest lost energy in these processes. In numerous industrial processes hot gasses are also produced and extracted. These gasses are then either released to the environment or cooled down. This waste heat also has the potential to be used for work or heating [36], [37].

2.3 Improving efficiency to reduce consumption of fossil fuels

2.3.1 Methods to utilize waste heat from exhaust gasses

Currently there are two attractive methods to improve efficiency for current fossil fuelled power generation. These two methods are co-generation and combined heat and power (CHP). Most power plants and various other industrial processes produce energy while producing heat thus being classified as co-generation [38]. A significant amount of this heat that is produced is released into the atmosphere and the energy is lost [38].

CHP processes can be used to recover the wasted heat from exhaust gasses. By utilising the waste heat in a CHP process to generate electrical energy, the overall system efficiency is improved [39]. Therefore, the CO₂ produced is reduced significantly for the same amount of energy that is produced as less is wasted. [39].

In order to recover this heat and use to recover energy from it, the heat needs to be at a high enough grade (high temperature) to use conventional cycles like the Breyton or Rankine cycles [1]. Unfortunately, as most of the time the wasted heat comes from a similar cycle the exhaust temperatures are a lower grade heat. That makes that these conventional systems will function inefficiently or not at all. This is where an Organic Rankine Cycle (ORC) becomes an attractive option [1], [37].

There are generally considered to be three qualities of waste heat that is available: above 650°C which is high grade, 232 - 649°C which is medium grade and lower than 232°C which is low grade. S. O. Oyedepo and B. A. Fakeye [40] explored some of these options and came to the conclusion the ORCs are an attractive option to recover lower grades of waste heat. This is due to these systems typically being relatively simple in structure, reliable as well as suitable for a wide variety of heat sources due to the vast selection of organic working fluids available. These systems are typically used where there is lower grade waste heat available that ranges between 80 and 300°C.

There are various applications where an ORC system can be used to recover energy. These applications include wasted heat from power plants, industrial plants with sufficient waste heat, geothermal plants, solar thermal plants and ICEs [41]. This implies that there is ample opportunity to apply ORC systems to utilise waste heat from exhaust gasses [42].

2.3.2 Organic Rankine Cycle to recover wasted exhaust heat

According to Sprouse [43]. The ORC is a closed-loop cycle that functions similarly to the conventional Rankine cycle with the main difference being the working fluid. The standard working fluid in a Rankine cycle is water, where in the ORC organic fluids are used instead. This is to utilise the lower grade heat that is supplied in the exhaust gasses, where water would not be effective or viable.

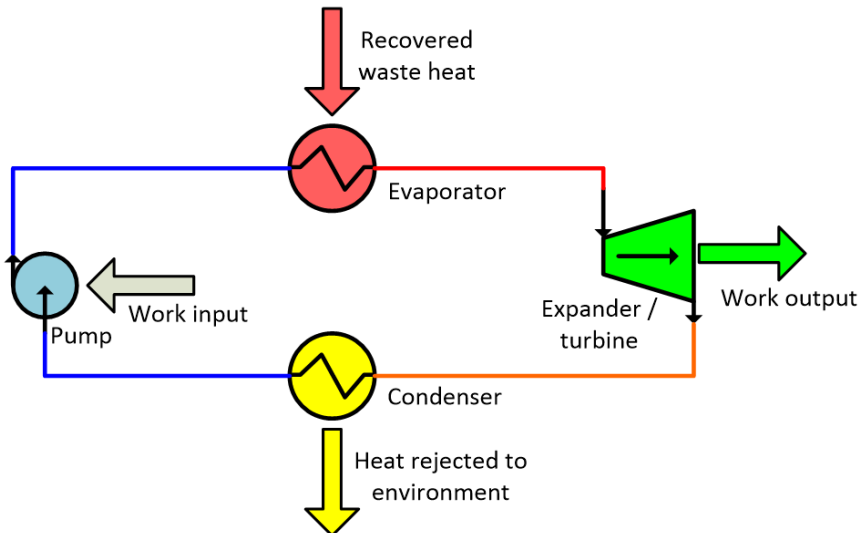


FIGURE 2.2: Organic Rankine Cycle: Process flow (Adapted from[44])

The operation of such a system starts with a heat exchanger using the waste heat from the exhaust gasses to heat the working fluid [1], [45]. This working fluid is then expanded through an expander/turbine. The turbine drives the ORC feed pump as well as an electrical generator. The feed pump pressurises the fluid to its original working state after it has been cooled down by a condenser that removes the excess heat [1], [45] [46].

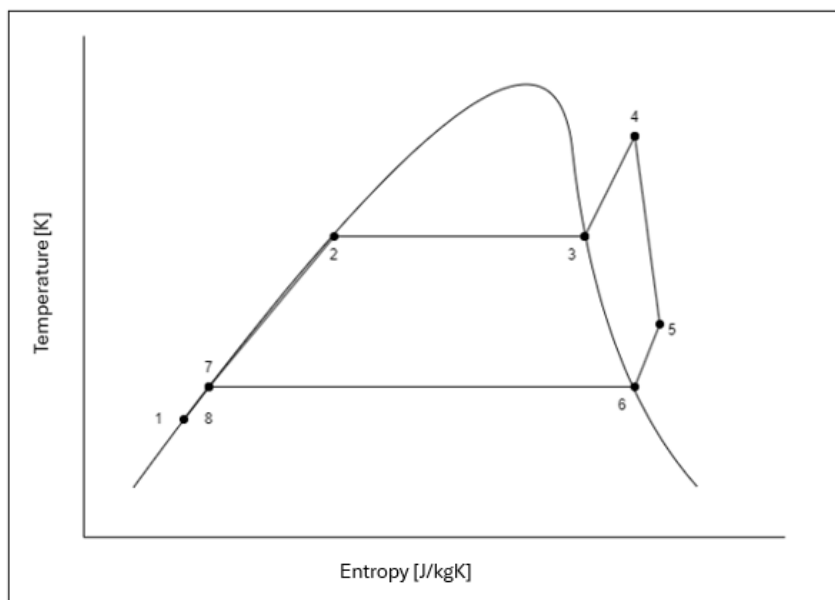


FIGURE 2.3: Temperature - entropy diagram of an organic Rankine cycle [47]

Figure 2.3 shows a temperature vs entropy diagram and figure 2.4 shows a pressure - enthalpy diagram of an ideal ORC. The liquid working fluid is heated to its wet saturated condition in process 1- 2. In process 2 - 3 the working fluid is heated and vaporised from its wet saturated to dry saturated state. It is then superheated in process 3 - 4. Therefore, there is a significant amount of energy available in this

system. After superheating the working fluid, it is expanded through a turbine to extract energy from the system from points 4 - 5. Once the energy is extracted it is then condensed to fluid thus moving the working fluid from position 5 - 6 - 7 - 8 so that it can be pumped to the heat exchanger. Then the cycle starts over and repeats [47].

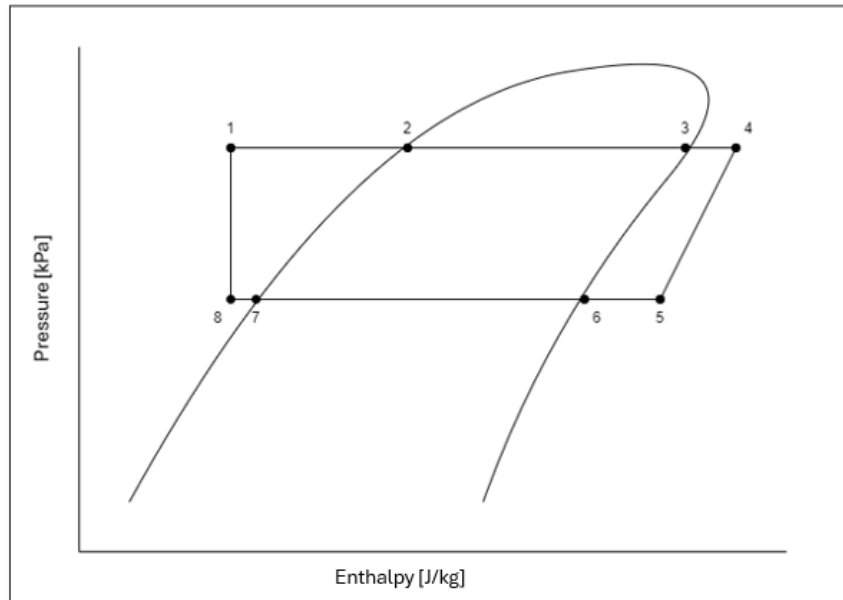


FIGURE 2.4: Pressure - enthalpy diagram of an organic Rankine cycle [47]

It is important to utilize the waste heat that is available from exhaust gasses effectively. Generally this waste heat is considered to be of low and medium grade, meaning under 500°C [3]. The optimal working fluid will depend greatly on the heat that is available. Therefore, there is not an ideal working fluid for an ORC, but rather a selection should be made based on the heat that is available.

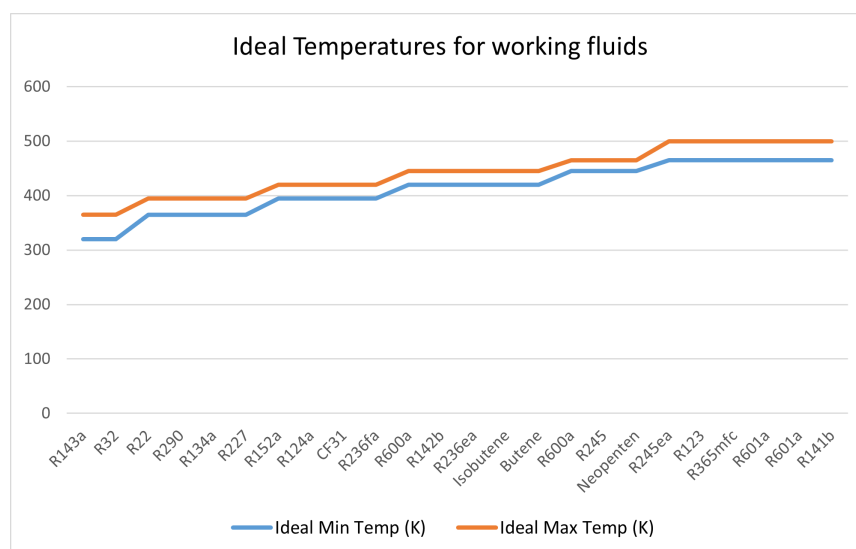


FIGURE 2.5: Chart of suitable refrigerants at various temperatures (Adapted from [3], [46])

Figure 2.5 depicts various working fluids that are most suited for a specific temperature range [3] [46].

When the temperature differences in the process are not sufficient, water (steam) becomes less effective because it is susceptible to losses such as shock losses [3]. An organic fluid has a higher expansion ratio for the same conditions and would lead to a more efficient turbine [3], [4].

Bao[3] further expands that on the ideal operating conditions for various organic working fluids, by considering parameters like the heat source temperature, application, evaporation and condensation temperatures. These parameters can then be used to select a suitable working fluid. Similarly Quoilin [48] shows that R134a is often used in ORCs that have lower grade heat available. Whereas popular choices for medium to low grade sources are R123 and R245fa. The study also highlighted some working fluids for very low-grade waste heat like R1234yf and highlights that water should generally only be used in higher grade heat systems.

2.4 Organic Rankine cycle expanders

In an Organic Rankine cycle, there are various types of expanders that can be used to extract energy from waste heat. These expanders are divided into two main categories, namely volumetric expanders and rotodynamic expanders. The general concept behind volumetric expanders is to extract work from expanding a working fluid, thus the work is generated by the change in volume. Whereas with rotodynamic expanders the work is extracted through the rotational movement of a turbine [49], [50].

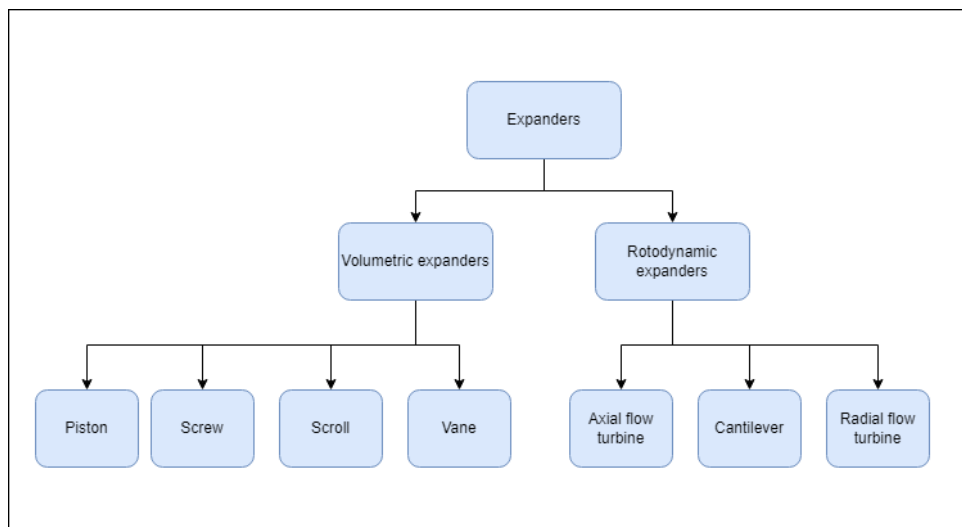


FIGURE 2.6: Types of expanders [49]

Figure 2.6 show some of the more popular types of expanders that are generally used.

Piston expanders:

Piston expanders use reciprocating motion to extract work that is transferred to a shaft. They can have high expansion efficiencies. However, they are complex and require a significant number of mechanical parts like bearings and valves to function efficiently. They are also generally more suited to higher expansion ratios [49], [51].

Screw expanders:

This expander continuously expands the working fluid across two inter meshing screws/rotors. This has the benefit of not needing any valves, however it does require the use of four bearings which does add complexity to the system and requires lubrication. These expanders generally function better at higher volumetric ratios and have reasonable efficiencies, approaching approximately 50% [49], [52].

Scroll expanders:

The scroll is a volumetric expansion device which is much simpler in design than piston expanders. It consists of two discs with inter meshing spiral grooves, one stationary and the second mounted on a motor driven shaft. These expanders are also not sensitive to the formation of droplets within the system. Generally it is used in lower energy applications and could also be an attractive option for this study [49], [50], [53].

Vane expanders:

Vane expanders are used when lower speed applications are considered. They generally have efficiencies below 50%. This is due to high friction losses in most applications. Generally these expanders also have considerable wear due to the vanes being in contact with the casing [49], [52].

Rotodynamic Expanders:

Weiss [49] compared volumetric expanders with rotodynamic expanders for use in small applications and found the rotodynamic expanders to be more suited to small scale applications. The criteria considered are the volumetric ratio, size, adaptability, wear, etc. He further compared impulse turbines against reaction turbines and found impulse turbines to be more ideal than reaction turbines for smaller applications. Therefore axial, radial inflow and cantilever inflow turbines are seen as ideal candidates for this study.

2.4.1 Turbine selection

To develop a turbine for an ORC system there are various turbine designs that are available for use. These turbines are generally categorised in two major groups namely: Axial flow turbines and radial flow turbines.

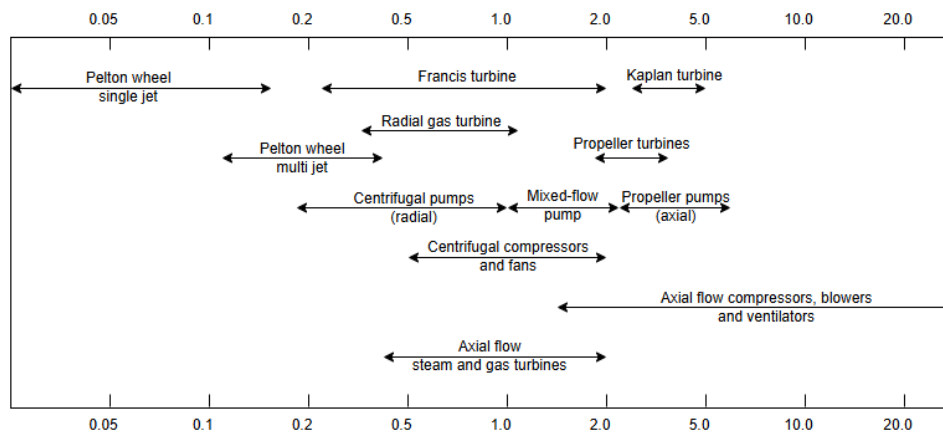


FIGURE 2.7: Ideal specific speeds of various turbomachines [54]

Figure 2.7 from Dixon chapter 2 [54], shows the specific speed of various different turbomachines. This is very helpful in selecting a suitable turbine as the expected specific speed can be used to select the appropriate turbine.

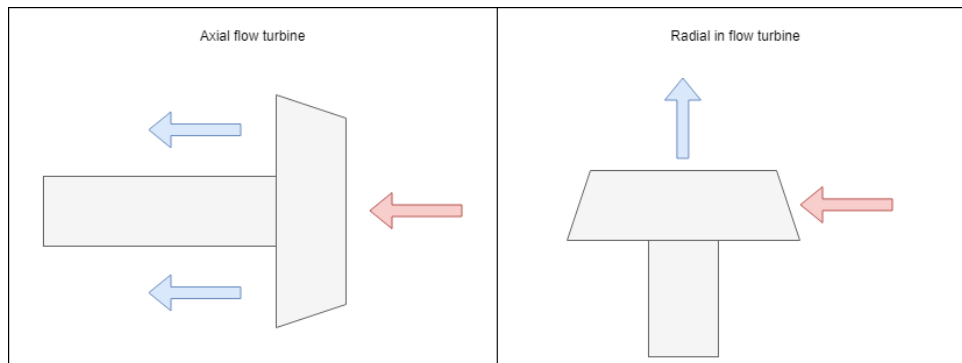


FIGURE 2.8: Turbine types

Axial flow turbines:

Axial flow turbines operate by expanding the fluid over a rotor parallel to the shaft. Both the inlet and exit of the working fluid is in the axial direction, as seen in figure 2.8. Axial flow turbines are usually used when the specific speed is between 0.4 and 2 [54]. Generally axial flow turbines performs better with higher volumes of working fluid with lower pressure ratios than compared to radial in flow turbines [55]. Alexander et. al. [56] compared the differences in head when using axial flow turbines for micro hydro system. These turbines were tested up to 40m of head which is roughly 3.9 bar at the higher end of the pressures the turbines started to lose efficiency. Kolasinski, P et. al. [57] looked at turbines for micro ORC systems and found that axial flow turbines were ideal for specific speed for these turbines to be between 0.05 and 0.1. This was dependant on the expansion ratio of the turbine.

Radial inflow turbines:

Radial inflow turbines functions similarly to axial flow turbines by expanding the working fluid over a rotor [58], [59]. However as seen in figure 2.8 the working fluid enters the turbine perpendicular to the shaft and is directed axially by the rotor before leaving the turbine axially. Kiyarash et. al. [60] investigated small scale radial flow turbines specifically for the use in ORCs. The results found indicated efficiencies between 82% and 84%. The turbines were tested using various refrigerants including inter alia R245a, R134a and R123. Most of these turbines achieved high efficiencies when used in a suitable application, which was mostly determined by the temperature of the application [61], [62]. This give the turbine a significant amount of flexibility to be adapted pending on the available conditions and made it an attractive option for this study.

The radial inflow turbine is a very good turbine to use for such a system, this is because it can be easily adapted for different working fluids [5]. It is not as sensitive as other turbines to the profile of the blades, thus turbines with simpler blade profiles can be created [6]. The blade profiles can also be made more robust and therefore be run at higher speeds. Radial flow turbines are simpler to manufacture when compared to axial flow turbines [7].

2.4.2 Small scale radial turbines

Small radial inflow turbines work most effectively at the following conditions [7], [54]:

- Head coefficient greater than 1.0 and less than 6.0.
- Flow coefficient greater than 0.095 less than 0.8
- Specific speed greater than 0.2 and less than 1.0

As the size of the turbines increase, the power that is produced increase as well. At a certain point increasing the diameter of a single stage radial inflow turbine becomes less effective. When the inlet area to the rotor is between 6 cm and 16 cm it seems it is most effective while using an organic fluid [7].

Determining pressure losses will be crucial in determining the efficiency of the turbine. Factors that typically contribute to the pressure losses are [5]:

- Losses from the tip clearance
- Loss of kinetic energy at the exhaust
- Impeller losses
- Losses due to friction
- Turbulence losses in the rotor
- Other forms of losses like leakage, flow and bearing losses

All of these losses will have to be taken into account when calculating the mechanical and system efficiencies for the turbine [63].

2.5 Production methods

In order to manufacture small radial inflow turbines there are a few options that can be considered. These can be grouped into two different categories which are conventional/traditionally manufactured and additively manufactured. Traditional manufacturing is a subtractive method whereby one starts with a large amount of material and machine away the material in order to produce a part or component. Whereas AM consist of adding material sequentially in order to produce a product [64].

2.5.1 Conventional production of radial inflow gas turbines

When considering conventional manufacturing of radial inflow turbines, especially the rotors, turbochargers is a good starting point. Generally, turbochargers have very similar rotors and have been manufactured for a significant amount of time. Casting is an attractive option that results in minimal machining after producing the rotor. The shaft is generally machined on a lathe/CNC machine which is then friction welded to the turbine [65]. The shaft can also be manufactured in such a manner as to pass through the turbine to be attached to the rotor using a fastener on the other end.

This requires a significant amount of time in post processing to ensure that the turbine and shaft is balanced during the manufacturing process [66]. Klink et. al. [67] investigated various method for manufacturing turbocharger impellers like wire-EDM and wire-ECM, which can also be attractive methods to manufacture smaller turbines.

2.5.2 Additive manufacturing methods

Additive manufacturing is quite a broad field and there are various additive manufacturing methods available [68], [69].

- Laser based melting
- Thermal extrusion
- Material Jetting
- Material adhesion
- Electron beam melting

Lazer based AM processes consists of 2 major categories i.e., laser melting and laser polymerization, the latter which is used for resins. Laser melting comprisses of two processes which either uses a powder bed for methods like selective laser sintering (SLS), selective laser melting (SLM) and direct metal laser sintering (DMLS) whereby powder is deposited onto a bed and a laser is used to melt the material and for a structure. Whereas methods like DMD, LENS, SLC, etc uses a powder supply that feeds directly in front of the laser to to produce the parts. [68], [69]

SLM can be used with various different alloys. The process uses a powder, usually metallic that is melted with a laser as follows: The entire bed is covered with a layer of powder after which, a laser selectively melts the powder in the desired form. Once a layer has been completed another layer of powder is deposited after which the excess powder is scraped away, and the laser can melt the next layer of the part. This fuses the first and second layer together. The process is repeated until the part is fully completed [68], [69].

Direct metal deposition (DMD) is the a process that is very similar to SLM as discussed above, however the major difference being that the entire bed does not get covered with the powder/wire. The material is only deposited directly in front of the laser in the shape of the part that is being produced. This powder or wire is then melted with the laser and the next layer can be started until the part has been completed [68], [69].

2.5.3 Additive manufacturing of turbines

The method of AM chosen for this study was SLM due to the availability of the manufacturing method as well as to specifically test whether this method would provide useable components.

AM is an attractive method to produce intricate and small components, where conventional machining would be too expensive or impractical [70]–[72]. It can use a wide variety of materials and alloys to create components with high accuracy. AM, especially laser sintering, allows for very precise control over geometry, mechanical

and thermal properties, by altering the laser conditions; where the relative density of the produced part can be as high as 99,5% of that of the mother material [73].

By using AM to create a small radial turbine, this study can further explore and determine possible applications or situations for using AM. By determining the efficiencies on the turbines, insight can be gained on the expected performance of turbines operated at, or closer to design conditions, as well as with organic working fluids. Another benefit of using AM is in the reduction of parts needed to be produced as well as a reduction in weight by being able to print more complex structures [70].

2.5.4 Materials for Small scale turbines produced by AM

AM manufacturing can be used with a wide variety of metals such as aluminium, copper, steel, stainless steel, titanium, etc. as well as various alloys of these materials [74]. Depending on the material there are various AM manufacturing process that can be used. Materials such as aluminium are not generally used with SLS and SLM due to the properties of the materials whereas titanium and steels are ideal. However, there has been some recent advances in the use of aluminium powder by companies like EOS [75]. The type of material will influence various factors such as the build rate, surface finish and density to name a few [74].

Titanium (Ti) has a low thermal conductivity and a high resistance to corrosion. The elastic modulus and specific strength is high compared to most metals [11], [76]. Ti-6Al-4V is used for compressor discs, blades and stators, due to it having a high ultimate tensile strength (UTS) and Young's Modulus at high temperature [11], [76]. The above-mentioned properties make Ti-6Al-4V an ideal candidate material to create turbines for the ORC, using AM.

Generally, materials such as Ti-6Al-4V are not traditionally used to produce turbines due to a high loss of the material from conventional machining. This is problematic as the material is expensive when compared to steels. AM have the benefit of not wasting as much of the material thus making this alloy more usable for applications like this [77]. Therefore, with the reduction in material required and the properties of an alloy such as Ti-6Al-4V it becomes an attractive option to manufacture smaller turbines in order to ensure high strength at various working temperatures.

2.6 Design of 90° IFR turbines

In order to design 90° IFR turbines there are various different design methodologies that can be used, which ranges from 0-D to 3-D (CFD methods). Generally 0-D or 1-D methods are used for the initial design in order to determine the basic geometry of the turbine of which a 3-D design can be created using those parameters [78].

0-D approach:

This comprises of a simplified model where the entire turbine is treated as one component. This model does not account for the actual geometry of the rotors and generally considers the inlet and outlet conditions at each stage of the turbine. Initial conditions were specified, which made for a starting point along with key design parameters like the required power, condition of the working fluid (pressure and temperature) and

size of the turbine to name a few [54], [78].

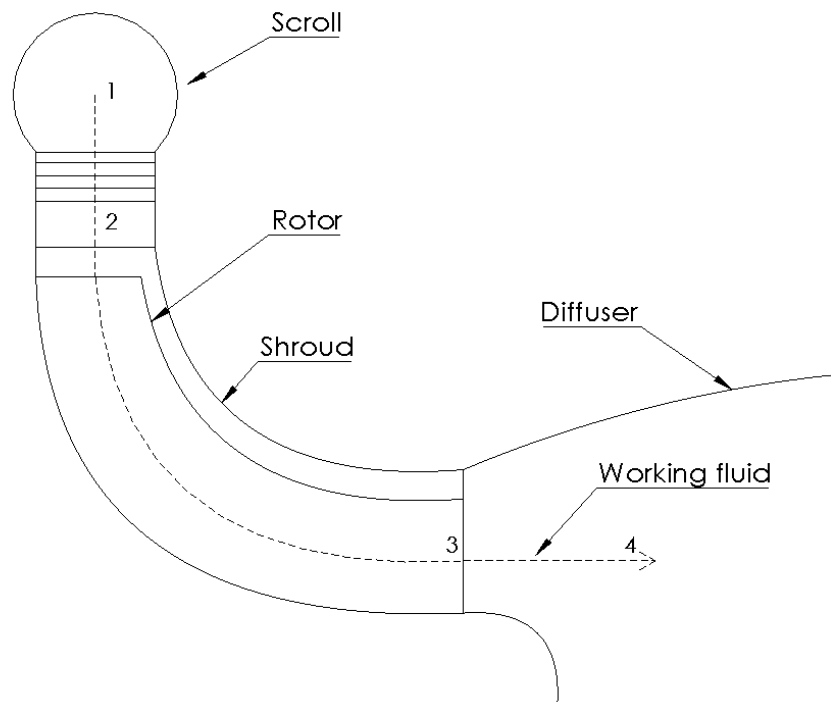


FIGURE 2.9: Diagram of a 90 degree inflow radial turbine [54]

In figure 2.9 these points are displayed as generally used in practice. The working fluid enters the turbine at point 1 which is the scroll. This directs the working fluid to the inlet of the rotor which is position 2. The working fluid then expands over the rotor blade providing rotational energy to point 3. Finally the working fluid leaves the turbine via the a diffuser at position 4 [54].

The conditions at each of these points can then be used to design and predict the performance of a 90° radial inflow turbine, by determining the energy at each of the four points. The process to determine the energy at each of these points are thoroughly described by Dixon [54] in chapter 8. Whereby the expected power can be determined using the enthalpy at position two and three.

Similarly, the inverse can be done to determine the design of a turbine. By using the power required and the energy available in the working fluid this method can be used to determine the geometry of the turbine. Whereby the specific work out of the system will be equal to the velocity of the turbine rotor. The sprouting velocity can then be used with this relation to determine the rotational velocity of the rotor. Specifically for a 90° radial inflow turbine the ideal ratio between the sprouting velocity and rotational velocity is between 0.68 to 0.71 [54]. In chapter 3 the 0-D design methodology will be discussed and applied to the design of the 90-degree inflow additively manufactured turbine.

3-D (CFD) approach:

Three-dimensional design is a much more detailed design approach that take into account the entire profile of the turbine. This includes the actual geometry of all the components which are modelled. Usually this involves the use of computational fluid dynamic (CFD) simulations. Usually, this method is used to improve an existing design or improve on a design that was initially created using a simpler method. The geometry of the part being designed is split into a mesh, subdividing the profile in small cells which is the analysed using a numerical model. The density of this mesh plays a major factor in the accuracy of these calculations. However, making a mesh that is too fine results in simulations that can be extremely hardware intensive and requires extreme amounts of time to compute [78].

CFD was not used for this study due to the time and resources available, as well as being an initial design of the turbine. In future work this may be a useful method to further optimize the turbine.

2.7 Summary and conclusion

AM provides a flexible way to produce 90° radial inflow turbine prototypes for use in ORCs. Geometrically complex design can be manufactured easily, especially when creating smaller sized turbines. AM also significantly reduces the wasted material allowing the use of an expensive alloy such as Ti-6Al-4V, which has favourable properties for turbines used in such systems. These properties include the high relative strength and corrosion resistance. A 1-D design methodology is sufficient for the design of the turbines for this study as the turbines will be manufactured and tested. This is due to the turbines being an initial design and this study is not focusing on the optimisation of these turbines for which methods like CFD would be more suited.

Chapter 3

Turbine design methodology and design

3.1 Introduction

To ensure a well-functioning radial-inflow turbine (90° IFR turbine) it was important to start with a theoretical base. Firstly, a design methodology was implemented that summarised the order and method in which components were to be designed. This was done in a logical and sequential manner as to provide an effective workflow.

Some initial assumptions as well as initial design parameters guided the initial part of the design. This was done to determine the basic design geometry as well as some ideal estimates of the performance that is to be expected. Thereafter the applicable dimensions were selected to be as close to the theoretical ideal as possible. The changes to the ideal dimensions were driven by the following:

- Time and cost of manufacturing
- Parts count
- Assembly time
- Ease of use

The above considerations led to 40 mm being selected for the rotor diameter. The rest of the turbine was designed to be as efficient as possible, while staying within this limit. There are some exceptions such as the inlet and exit angles of the rotor blades as well as the clearances. Three different rotor and inlet guide vane configurations were selected. This was done to understand the influence of these configurations on the performance of the turbine.

Engineering Equation Solver (EES) software was used to create the theoretical model of the turbines. EES is software used to solve systems of non-linear equations iteratively. It also has a database of the thermodynamic properties of various working fluids. It calls the required thermodynamic properties by an iterative solution. It also has parametric tables that can be used to compare variables and generate graphs and plots, which is useful to characterise a turbine.

3.1.1 Initial design specification

The driving design specification for the turbine was to be as small as possible, while also being able to produce a usable amount of energy. The target design point power output was 500 W.

3.2 Theoretical design of turbine

3.2.1 Turbine rotor design

The first step was to design the rotor blades of the turbine. To do this the working points had to be specified. These points acted as reference points for the calculations to follow.

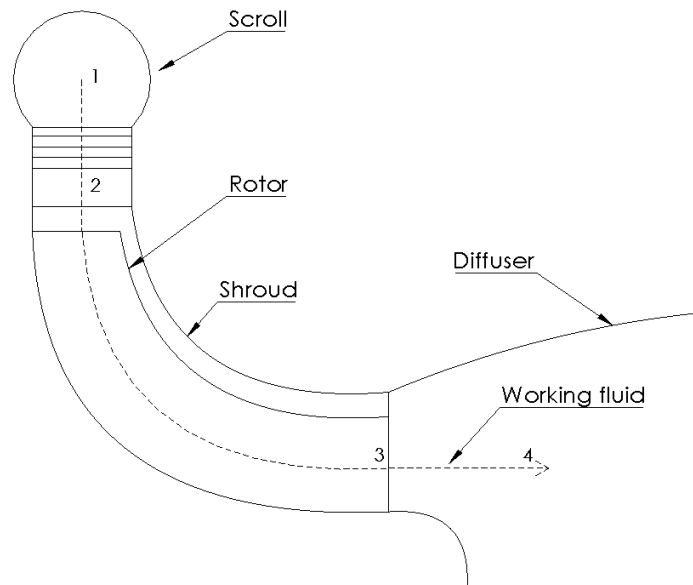


FIGURE 3.1: Diagram of a Turbine [54]

The points selected in figure 3.1 was:

1. Inlet to the nozzles/stator blades
2. Exit of the nozzle/stator blades and Inlet to the rotor blades
3. Exit from the rotor blades
4. Diffuser

Before the geometry of the rotors were designed, vector triangles were calculated. These give the values for the flow rates and velocities of the working fluid at certain points in the rotor. The power is determined by the amount of working fluid that is required at a certain velocity. This influences the design as you can use a smaller flow rate with a higher velocity or a larger flow rate at a lower velocity to produce a similar amount of power.

The flow rate and velocity have a large impact on the dimensions of the rotor. A higher flow rate and smaller velocity results in a larger turbine rotating at a smaller angular velocity. Whereas the opposite is true for a small flow rate with a higher velocity. The desired size of the turbine and limitations of the components and materials were determined by the initial design.

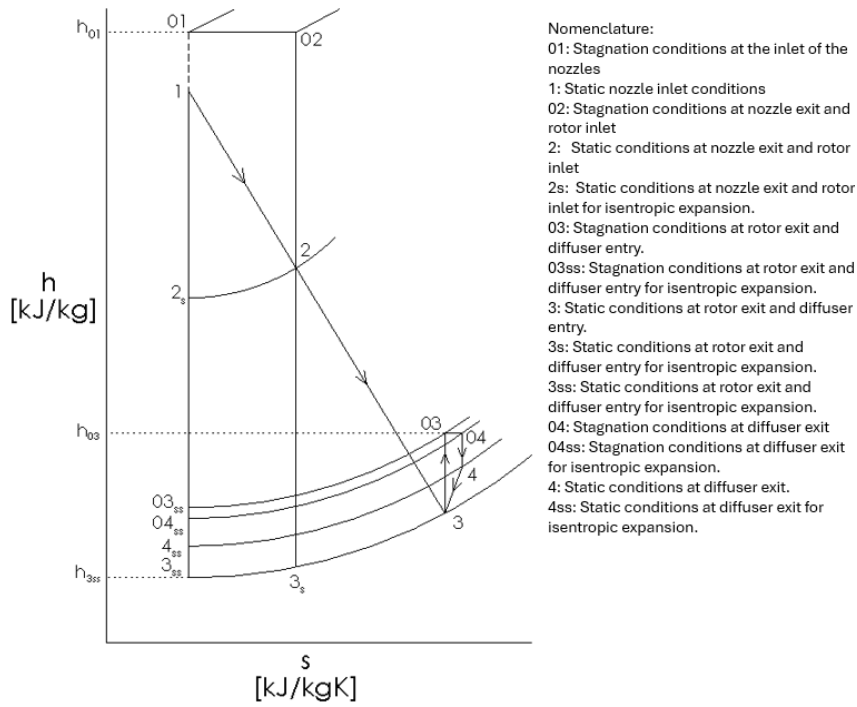


FIGURE 3.2: Mollier diagram of a 90 IFR turbine [54]

Figure 3.2 is a typical Mollier diagram for a turbine, which shows the rise in entropy as the enthalpy drops over a certain pressure ratio. When the entropy rise strives to zero then the turbine will be considered to expand the gas through an isentropic process. This means that it is an ideal turbine, which is practically not possible. However, the isentropic expansion can be used as a tool to compare the ideal, theoretical, and practical performance of the turbines.

In Figure 3.2 the enthalpy is high at point 1 which represents the maximum available energy at the turbine inlet. This is due to a higher temperature and pressure compared to points lower on the graph. As the gas expands through the turbine the enthalpy drops. The pressure and temperature also drop. The enthalpy drop is due to the energy being extracted from the turbine by expanding the gas through the turbine. These points can then be used to calculate the expected enthalpy at certain points in the turbine to calculate the efficiency and expected power delivery of the turbine. The rise in entropy during this process is a direct indication of inefficiencies in the turbine.

$$\Delta W = h_{01} - h_{03} \quad (3.1)$$

The theoretical maximum work done by a turbine can be calculated by subtracting the enthalpy of the working fluid at exit from the enthalpy of the working fluid at inlet as seen in equation (3.1). This determines the amount of energy lost from the working fluid and is the absolute thermodynamic maximum work that the turbine can produce [54].

$$\Delta W = h_{01} - h_{02} = h_2 - h_3 + 1/2(c_2^2 - c_3^2) \quad (3.2)$$

From (3.2) the power can also be determined by using the enthalpy at the selected points and adding the kinetic energy of the working fluid at each point. This is useful as enthalpy is dependent on the temperature as well as the pressure of the working fluid. Both of which can be easily measured and recorded. By adding the kinetic energy, the stagnation enthalpies can be calculated.

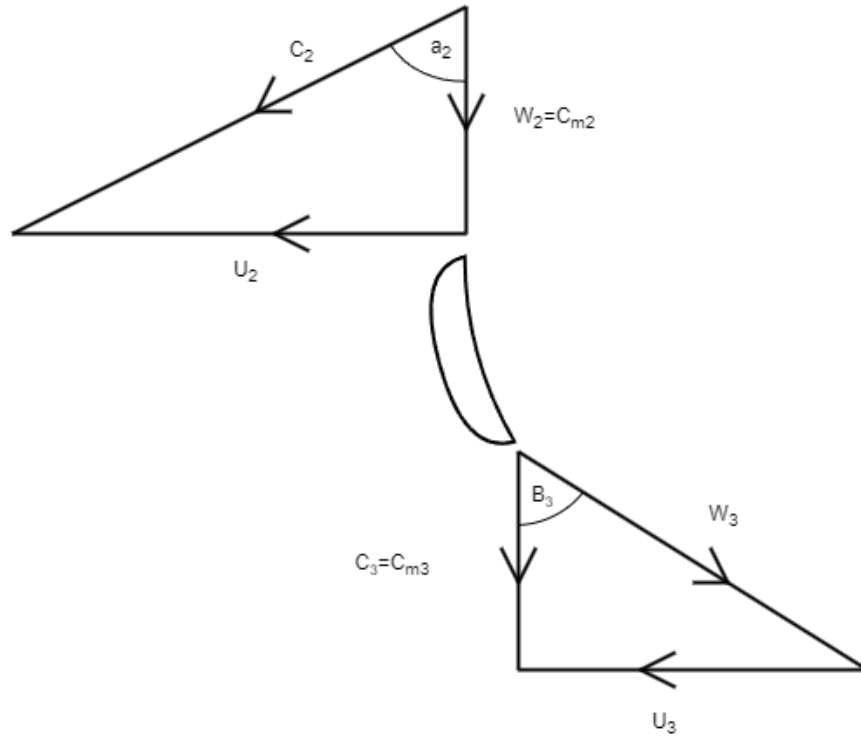


FIGURE 3.3: Velocity diagram for the turbines

Figure 3.3 depicts the velocity triangle of a turbine, which describes the motion of the rotor relative to that of the working fluid. For the purpose of optimisation, the turbine will be designed for nominal conditions. This means that at inlet the whirl velocity is equal to the relative velocity and the exit velocity is equal to the relative velocity [54].

$$1/2(C_0)^2 = h_{01} - h_{03ss} \quad (3.3)$$

In (3.3) the sprouting velocity is calculated. This is the amount of kinetic energy in the working fluid that is equal to that of the enthalpy drop from the turbine inlet to the stagnation pressure. The sprouting velocity relative to the velocity of the turbine rotors, can be used to determine the velocity ratio of the turbine [54].

$$U_2/c_0 = 0.707 \quad (3.4)$$

Equation (3.4) depicts the ideal velocity ratio for an actual frictional IFR turbine. This value is generally between 0.68 and 0.71. A value between those borders is selected for the theoretical model of the turbine.

For the theoretical model certain losses also need to be brought into account. These losses included but are not limited to the nozzle and rotor losses [54].

$$\zeta_N = (h_2 - h_{2s}) / (1/2c_2^2) \quad (3.5)$$

$$\zeta_R = (h_3 - h_{3s}) / (1/2\omega_3^2) \quad (3.6)$$

Equations (3.5) and (3.6) are used to calculate both the expected stator and rotor losses. By calculating the theoretical expected losses from the stator and rotor, the theoretical model should more accurately represent an actual turbine [54].

$$Z = \pi/30(110 - \alpha_2)\tan\alpha_2 \quad (3.7)$$

For the selection of the number of blades, there are three generally accepted methods which are [54]:

- Jamieson equation
- Glassman equation
- Whitfield equation

Glassman's equation as shown in equation (3.7) use the relation between the number of blades and the flow angle of the working fluid to determine an ideal number of rotor blades. Glassman's method was selected as the others result in the rotors having too many vanes. Due to the size constraint a smaller number of vanes is preferable. Similarly using the design philosophy depicted in Dixon [54]. Reasonable inlet and outlet angles for the turbine will be selected.

3.2.2 General turbine design

Three turbines were designed in order to study the influence of configuration differences on the actual performance of the turbine. It would also assist in validating the theoretical design methodology. Ideal graphs were created using the data from the 3 turbines and their various configurations. This data was used to compare to the experimental counterparts to calculate the losses of the turbines.

The losses due to the surface roughness was important to determine, as it showed the influence of the production method. This was important to note as it is one of the largest unknowns of the effect that AM has on the performance of a small 90° IFR turbine. These losses can be determined in future studies by comparing similar turbines produced using AM and conventional tooling.

The theoretical design consisted of:

- Velocity triangle design
- Geometry design
- Blade profile design
- Calculation of clearances and determination of losses

The velocity triangle design was used to find the approximate values that was needed to design the geometry of the turbine rotors. From the geometric values the theoretical performance curves of the turbines were calculated. This was done for all of the configurations that would be tested experimentally. The calculations also considered relevant losses to ensure that the data was compared accurately. See appendix A for the EES calculations.

The following design specifications were selected for the turbines:

- It must be a small 90° IFR turbine, therefore 40 mm rotor diameter was selected.
- The method to produce the turbine was selected laser melting, one of the AM technologies.
- The material that was used was Ti-6Al-4V.

For each of the rotors' blades the exit angle was changed. Where the guide vane inserts were designed with 2 different exit angles (gas inlet angle to the rotor) and two different clearances. Meaning that two guide vane inserts had the same exit angle, but different clearance between rotor tip and guide vane inserts and two had the same clearance, but different exit angles (cf. Table 3.1).

Compressed air was used as the working fluid to characterise the turbine. In future the turbine should also be characterised with organic fluids.

Assuming an inlet velocity and an inlet angle, the inlet velocity triangle can be completed. The inlet velocity is selected such that all the velocities in the rotor is lower than Mach 1. This is to reduce losses that occurs at Mach numbers greater than 1, as well as to prevent unnecessary shock in the system.

A value for the blade loss coefficient is then selected to calculate the relative exit velocity of the working fluid. The radius ratio (ratio between the outside and inside radius of the rotor blade) also must be assumed at an appropriate value in order to calculate the average velocity of the exit radius of the rotor. With both those values known the exit velocity triangle can be completed.

With both velocity triangles completed all of the enthalpies can be calculated. The theoretical enthalpy drop can then be compared to the isentropic enthalpy drop. The closer the theoretical enthalpy drop is to the isentropic, the more efficient is the rotor.

For the initial design certain assumptions were made in order to get a starting point. This also included the necessary constants needed for the calculations.

TABLE 3.1: Initial assumptions and selected values

Selected values and assumptions			
Description	Symbol	Value	Units
Pressure coefficient	c_p	1	kJ/kgK
Volume coefficient	c_v	0.8116	kJ/kgK
Gas constant	R	0.287	kJ/kgK
Inlet angle	α_2	65	°
Relative Mach number	M_{w2}	0.11	N/A
Power	$\dot{W}$	0.5	kW
Radius ratio	r_3/r_2	1.5	N/A
Relative inlet fluid temperature	T_{w02}	300	K
Flow coefficient	θ	0.3	N/A
Rotor inlet pressure	P_2	500	kPa
flow rate of air	m_{air}	28.97	kg/kmol
Isentropic efficiency	η_{is}	0.75	N/A
Inlet temperature	T_1	310	K
Turbine inlet pressure	P_1	600	kPa
Pressure ratio	P_{r2}/P_{r3}	1.2	N/A

Table 3.1 list the values that was selected as initial values for the design. It also shows some of the constants and assumptions that were made for the initial design of the turbine.

These values were used as the starting point of the design of the turbine rotors.

3.2.3 Guide vane inserts design

3.2.4 Casing design

The casing is an important part of the turbine to consider when designing.

The flow passage between the casing wall and the inlet to the guide vane insert assembly was made large to ensure equal flow on to the rotor blades from all directions.

The casing was also designed as a split casing as seen in ref figure 3.1. This configuration enabled for easy removal of the turbine rotors and shaft. A split casing design allowed for the shaft to be press-fitted into the bearings and still allow for easy removal or changing of components.

3.2.5 Geometry design of turbine rotors and guide vanes

The design process is iterative and needs to be adjusted in order for the rotor to be effective and correctly designed. The geometry of the rotor will be calculated using the values from the velocity triangles. If the geometry of the rotor is within the size and velocity constraints then the design can be theoretically characterised, by calculating the new velocity triangles as well as changes in enthalpy and entropy.

TABLE 3.2: Calculated turbine rotor geometry

Calculated geometry from velocity diagrams					
Description	r_2	r_{3s}	r_{3h}	t	s
Value	19.753mm	14.268mm	4.993mm	0.5mm	2.5mm

Table 3.2 shows the results for the final geometry for the turbines. These results will be slightly altered in order to aid the design of the experimental turbine. The changes to the geometry will then be used to determine the performance of the turbines. The blade inlet and exit angles are listed in Table 3.4.

TABLE 3.3: Practical geometry for turbine rotors

Practical geometry for turbine rotors					
Description	r_2	r_{3s}	r_{3h}	t	s
Value	20mm	16mm	6mm	0.5mm	2.5mm

In Table 3.3 lists the values for the geometry of the turbine rotor that is adjusted to fit the preferred geometry for the design. The r_2 was rounded up to 20 mm and both r_3 values adjusted accordingly. These values can be used to calculate the theoretical performance of the turbine.

The cross-sectional profile of the blades on the turbine was flat, with an incidence angle of 5 degrees. This does however not play as big a role on a turbine as with a compressor. A flat profile was chosen to keep the turbines as close to a baseline model as possible and to introduce the smallest number of variables.

3.2.6 Influence of clearances and losses

Losses in the turbine will also greatly impact the performance of the turbine. These need to be added to the theoretical calculations. This needs to be done to accurately compare the theoretical with the experimental results. The primary losses to consider are nozzle-, rotor-, clearance- and windage-losses [78].

Nozzles that are designed well, usually has a flow coefficient ϕ_n between 0.9 and 0.97 [54]. The rotor losses ϕ_R can be expected to be between 0.7 and 0.85 [54]. These losses were used when theoretically characterising the turbine.

Windage and clearance losses are greatly dependant on the clearances between the rotor and the turbine casing, as well as the specific speed of the rotor. The smaller the clearance the less clearance losses can be expected. Care must be taken to avoid damage between the rotor and casing. If the clearance is too small vibration or thermal expansion can lead to turbine damage.

3.3 Turbine Specifications

3.3.1 Turbine rotor Specifications

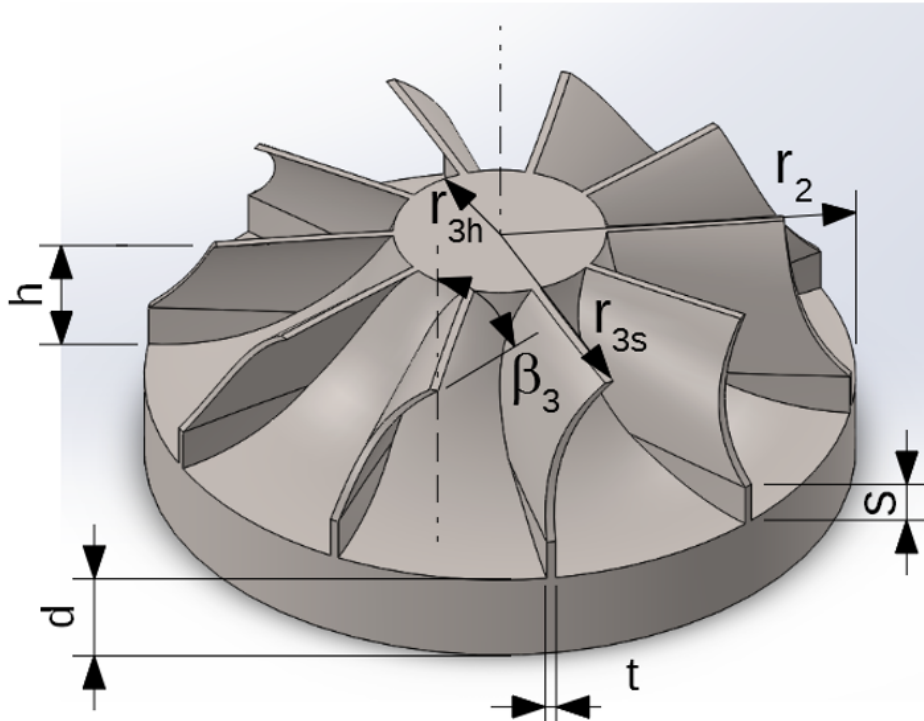


FIGURE 3.4: Rotor dimensions

TABLE 3.4: Turbine rotor specification

Description	Turbine rotor1	Turbine rotor2	Turbine rotor3
r_2	20mm	20mm	20mm
r_{3s}	16mm	16mm	16mm
r_{3h}	6mm	6mm	6mm
Blade height	10mm	10mm	10mm
Z	10	10	10
i	5°	5°	5°
β_3	50°	25°	0°
t	0.5mm	0.5mm	0.5mm
s	2.5mm	2.5mm	2.5mm
Base plate thickness	5.2mm	5.2mm	5.2mm
Material	Ti-6Al-4V	Ti-6Al-4V	Ti-6Al-4

Table 3.4 shows the design of the three different rotors. The only significant difference between them being the blade exit angle as the inlet angle is determined by the guide vane for this turbine.

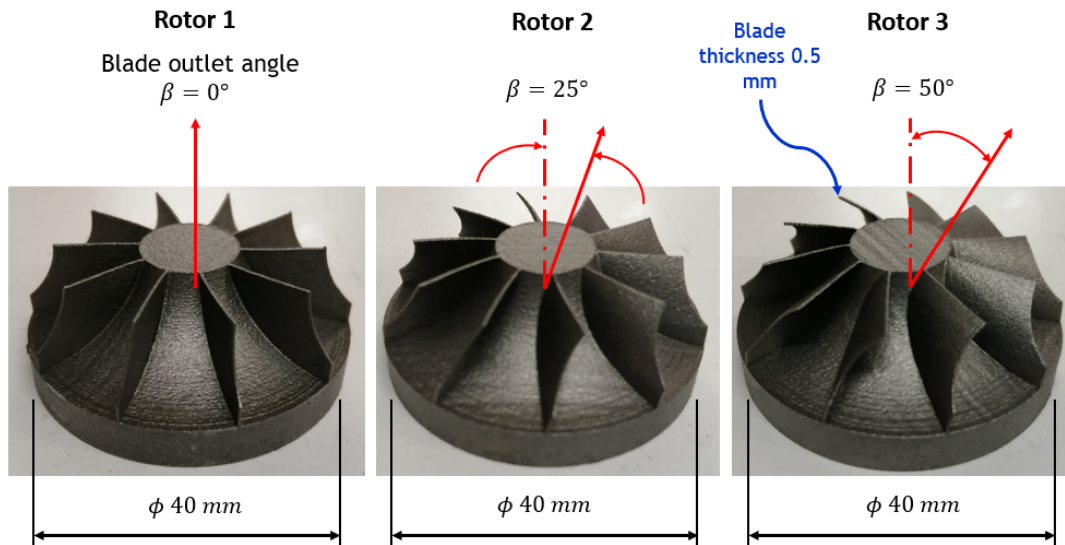


FIGURE 3.5: Photograph of the three manufactured turbine rotors indicating blade outlet angles of each.

In figure 3.5 the differences can be seen between the three different rotors. With rotor one having no exit angle, thus $\beta_3 = 0^\circ$. Rotor two has a blade exit angle of $\beta_3 = 25^\circ$ and rotor three $\beta_3 = 50^\circ$

Rotor 1 is the most basic design of the three turbine rotors. The blades were designed to have an inlet angle determined by the guide vanes and an exit angle of 0° . This was the simplest configuration and was expected to perform the worst of the three turbines. This is due to the exit of the turbine being parallel to the exit angle of the working fluid, which results in more of the energy in the working fluid not being utilised.

Rotor 2 has some curve at the exit of the rotor blades, with a blade exit angle of 25° . This was selected at a midway point between the first and the third turbine. This turbine rotor was expected to perform somewhere between rotor 1 and 3. This is due to the theoretical performance increasing as the exit angle increased up until approximately 65° .

Turbine rotor 3 is the turbine rotor with the largest exit angle of the three turbine rotors. This turbine had an exit angle of 50° . This was the rotor designed closest to the theoretical ideal. It was also the turbine expected to perform the best out of the three turbines.

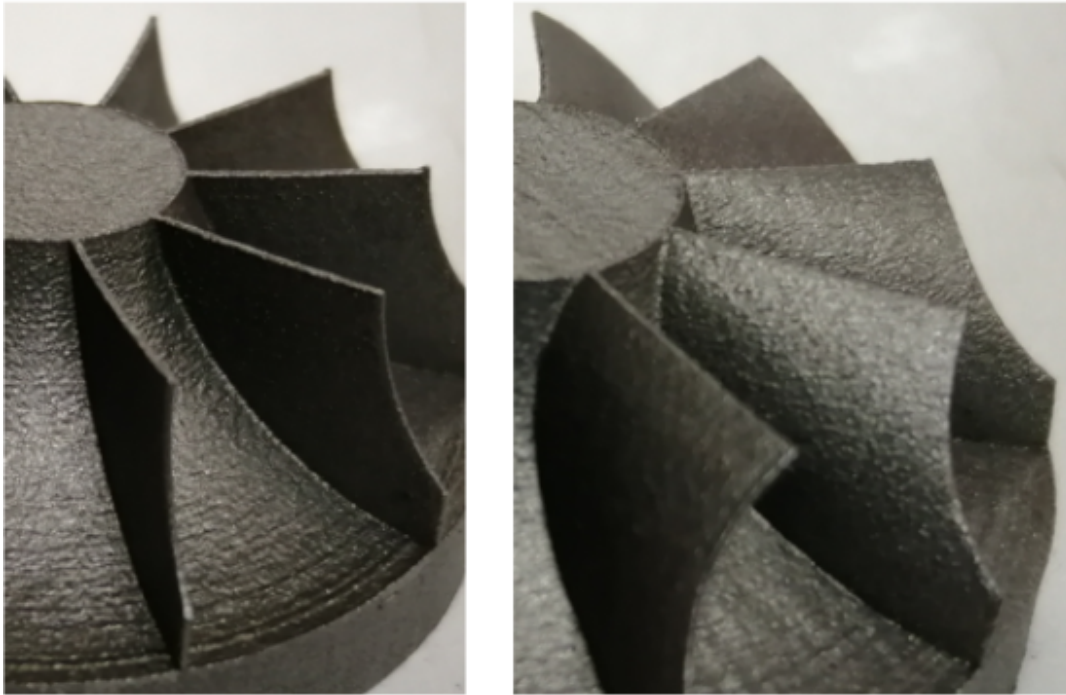


FIGURE 3.6: Closeup photos of rotors 1 and 3 to show the coarse surface roughness.

As can be seen from all figure 3.6, the surface of the rotors has a rough finish. This is due to the nature of the SLM process and will impact the performance of the rotors. This is expected to negatively influence the performance of the rotors, due to the rough surface finish and increased friction as a result.

3.3.2 Guide vane insert specifications

The guide vanes were designed in a similar fashion to the rotors. Three versions of guide vanes were designed and manufactured, changing certain variable in order to determine the performance of the turbine over a broader spectrum of configurations. The guide vanes were also manufactured using SLM and Ti-6Al-4V as the material. This was done so the rotors and the guide vanes could be produced in the same batch. This reduces production time as well as saving costs. This also meant that the guide vanes are strong as well as resistant to heat and corrosion, even at elevated temperatures. This should ensure a good life cycle from the guide vanes.

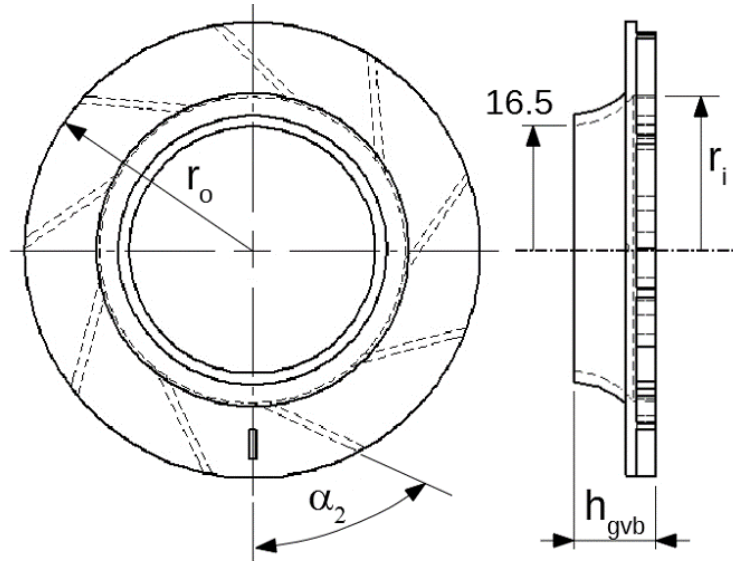


FIGURE 3.7: Guide vane dimensions

TABLE 3.5: Guide vane specification

Description	Guide vane 1	Guide vane 2	Guide vane 3
A_{inlet}	$40 \times 10^{-6} m^2$	$40 \times 10^{-6} m^2$	$40 \times 10^{-6} m^2$
A_{exit}	$25 \times 10^{-6} m^2$	$25 \times 10^{-6} m^2$	$25 \times 10^{-6} m^2$
$r_{outside}$	61mm	61	61mm
r_{inside}	41mm	41.5mm	41mm
Clearance	2.44%	3.61%	2.44%
r_{top}	33.55mm	33.55mm	33.55mm
α_2	65°	65°	55°
Material	Ti – 6Al – 4V	Ti – 6Al – 4V	Ti – 6Al – 4

Table 3.5 shows the differences between the three guide vanes that were manufactured. The inlet and exit areas were kept constant in all three guide vanes. The main differences between the guide vanes were the exit angle (α_2) and the clearance between the rotor and the guide vane. These values were changed in order to determine the influence these variables had on the performance of the turbines.

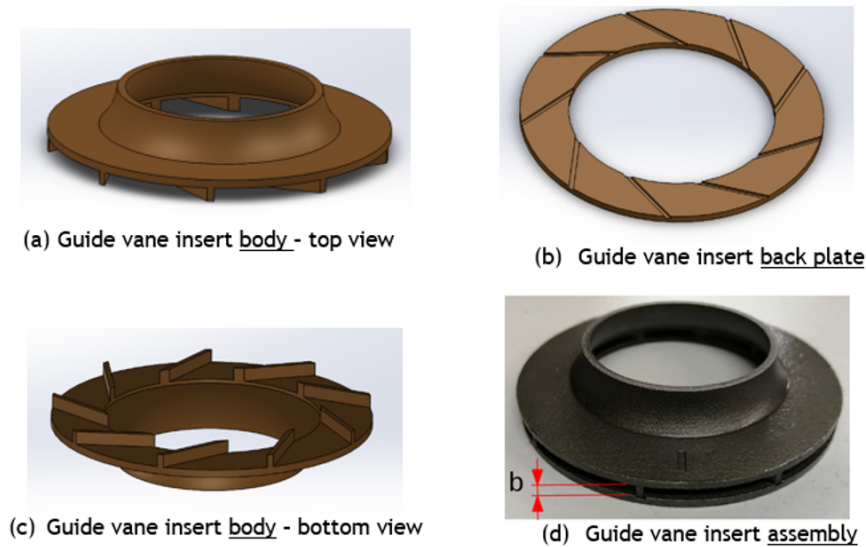


FIGURE 3.8: Guide vane parts

Figure 3.8 shows one of the manufactured guide vanes. These guide vanes were created in two parts, the top and the bottom, which were then pressed together to form one part. This was done, due to a restriction in the manufacturing process. By creating the part with additive manufacturing large overhangs cannot be made without adding supporting structures. These supporting structures would influence the flow of the working fluid and due to the gap only being 2 mm would be difficult to remove. The simplest solution was to manufacture the top part separately which meant that the supporting structures could easily be removed.

3.3.3 Turbine casing Specifications

One turbine casing was designed to accommodate all rotor and guide vane insert configurations. The main requirements for the turbine casing were:

- Accommodate ease of changeover between configurations.
- Even flow of the working fluid to the guide vanes.
- Should not restrict the flow.
- Easy connection of the inlet and outlet pipes.
- Must be leaktight.

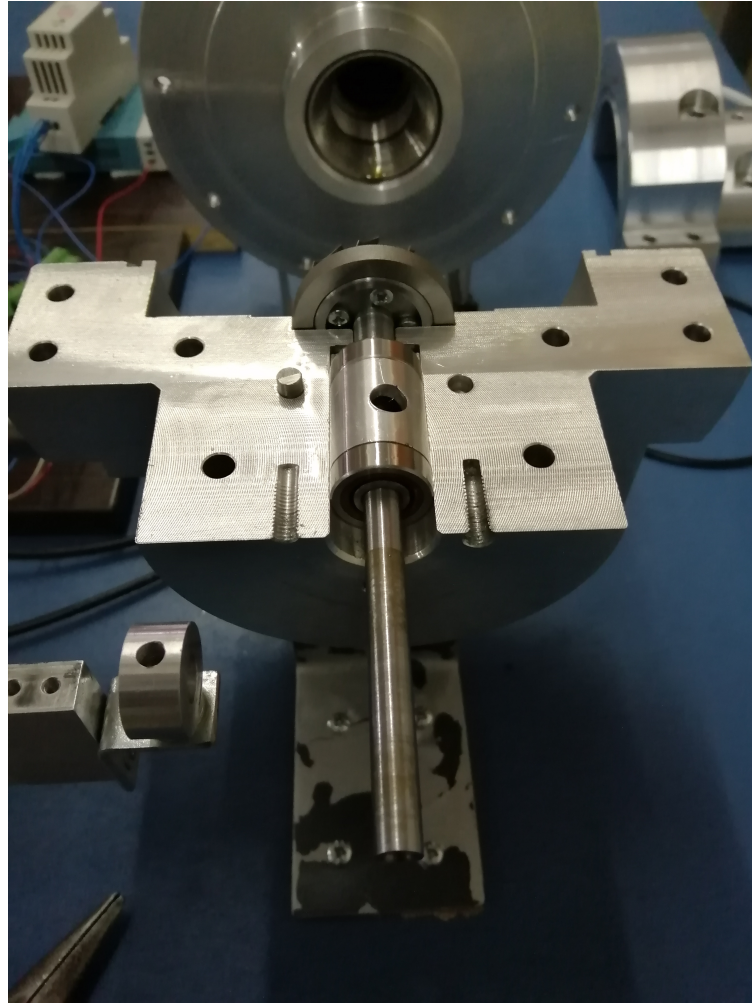


FIGURE 3.9: Turbine casing

In order to ensure that the setup could be switched easily between configurations a split casing design was used where the bottom and the top of the casing could be separated. The casing also had a cap on the front so that the guide vanes could easily be changed during test runs. Figure 3.10 shows the bottom half of the casing. The two halves were machined accurately and then bolted together to ensure a leaktight seal. The turbine was fitted onto a bearing casing that consisted of two bearings with a spacer in between in order to position the turbine inside the casing.

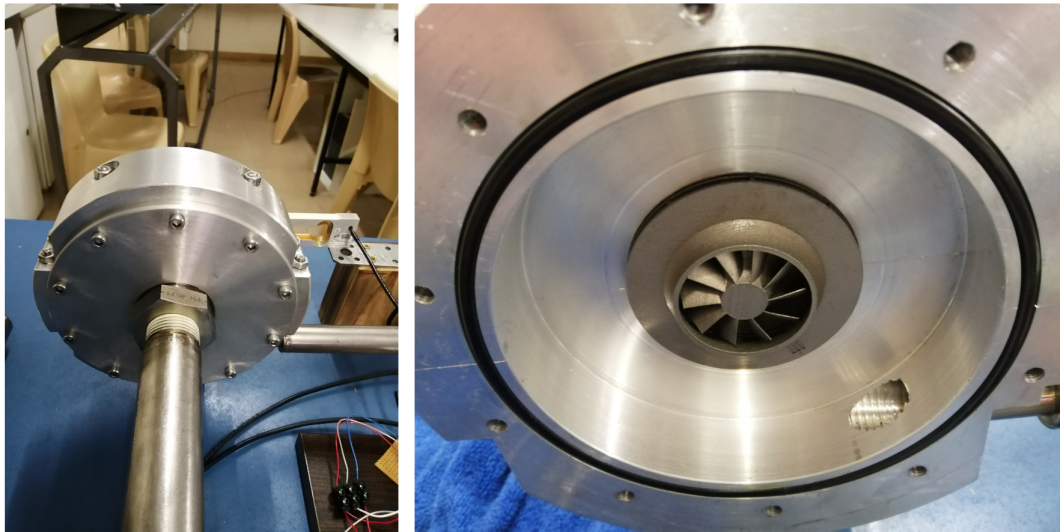


FIGURE 3.10: Turbine casing front

Figure 3.10 shows the front of the casing. A total of 10 screws were used to bolt the front of the casing tightly against the body. To achieve sealing between the casing front cover and the casing body an O-ring was fitted in an O-ring groove. From this figure it can also be seen that there is a rather large open volume around the turbine. This was done to ensure a uniform pressure distribution at the guide vane inlets, thus ensuring even flow through the turbine.

Chapter 4

Experimental investigation

4.1 Introduction

4.2 Design of experiment

4.2.1 Measured parameters

Certain parameters needed to be obtained in order to determine the performance of the turbine. The parameters that were selected to measure was based on the performance curves that was used to characterise the turbines. These curves formed the basis that was used to compare the theoretical and practical turbines.

The performance curves used to describe the turbine performance were as follows:

- Total to static efficiency versus the pressure ratio at different temperatures[79]
- Normalised flow rate versus the pressure ratio [54]
- Total efficiency versus the pressure ratio at different temperatures [54]
- Total to static efficiency versus the specific speed [54]

Important parameters that needed to be measured for the performance curves were pressure and temperature before and after the turbine, mass flow rate, rotational speed of the turbine and the torque developed by the turbine.

TABLE 4.1: Measured parameters

Inlet		Turbine		Outlet	
Description	Unit	Description	Unit	Description	Unit
Pressure	kPa	Rotational Speed	r.p.m.	Pressure	kPa
Temperature	K	Torque	Nm	Temperature	K
				Flow rate	Nm ³ /h

Table 4.1 lists the parameters measured during experimental testing of the turbine. Temperature and pressure were measured in the turbine inlet piping. This is done to determine the energy in the working fluid before it has done any work. The turbine was tested at steady state conditions. The mass flow rate is constant throughout the process. The parameters measured for the turbine was the rotational speed and the torque. These values were used to calculate the performance of the turbines at the experimental working conditions. Finally, the outlet temperature and pressure was measure at the outlet of the turbine to capture the remaining energy left in the working fluid. Volumetric flow rate was measured in the discharge pipe and converted to

mass flow rate.

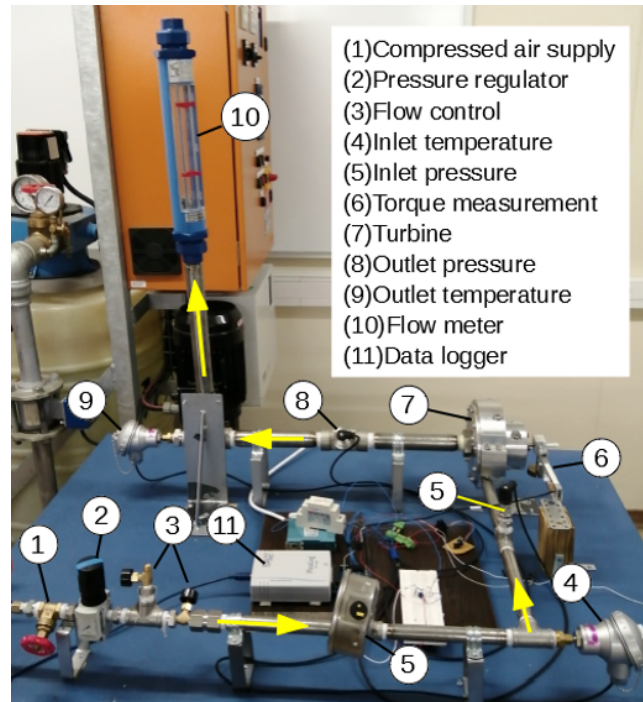


FIGURE 4.1: Experimental setup

Figure 4.1 shows the experimental setup used to test the turbines. The air from the compressed air supply (1) enters and passes through a pressure regulator (2). Thereafter passes some flow control valves (3), before the temperature (4) and pressure (5) are measured. The air then passes through the turbine (7) where a prony brake (6) is used to measure the torque, with a torque arm and load cell arrangement. Turbine exit pressure and temperature are measured at (8) and (9) respectively. Lastly the air flows through a flow meter (10) before exiting to the atmosphere. All data is captured on a data logger (11).

4.2.2 Instrumentation used to measure parameters

The inlet and outlet temperatures were measured with a resistance temperature detector (RTD) probe. For this experiment the platinum resistance PT100, 3-wire, class B resistance thermometer with KNE transmitter was selected. The reason being that RTDs generally are more accurate than thermocouples.

For the measurement of pressure, the Gems 3000 series pressure sensor was used with the DIN 43650 GDS 3-wire connection. This provided a 4 to 20mA current output that is compatible with the data acquisition device (DAQ).

For the flow measurement a rotameter was used suitable for the flow rate and the temperature of the experiment. No data needed to be recorded for the flow meter, as the system was set to run at steady state and a pre-set flow rate. The flow rate was manually adjusted. The selected flowmeter was a glass tube variable area flowmeter series 6001/6002 from Techfluid.

Torque on the turbine output shaft was measured with a Prony brake and lever mechanism. Lever force was measured with a load cell and converted to torque. The rotational velocity was measured using a tachometer. With both of these values the power can be calculated at various pressure ratios and flow rates and from that the performance curves of the turbines can be created.

The data was recorded with a 1216 PicoLog DAQ system used with a external terminal board. The external terminal board was used to ensure that the sensors provided the signal in the correct voltage and ampere range for the system.

TABLE 4.2: Instrumentation used to record data

Instrument Specification	
Temperature Sensors	
Model used	KNE PT100 RTD
Temperature range	0 - 200°C
Accuracy	IEC 751 Class B
Signal transmitted	4 - 20mA
Pressure Sensor	
Model used	GEMS 3000 Pressure sensor, DIN 43650C GDS connection
Pressure range	0 - 10 Bar (gauge)
Accuracy	1% of full scale
Signal Transmitted	4 - 20mA
Rotameter	
Model used	Techflow series C313-06301
Flow range	1.8 - 18 Nm ³ /h
Accuracy	Class 1,6 according to VDI / VDE 3513,
Measurement	Manual visual
Load Cell	
Model used	Vishay Transducers Model 1004
Weight range	0.3 - 3 kg
Accuracy	0.0067% of full scale
Signal Transmitted	0.9+-0.1 mV/V
Tachometer	
Model used	RS Pro Tachometer
Rotational range	0 - 99 999 rpm
Accuracy	0.05% of full scale
Measurement	LCD display
Data Acquisition	
Model used	PicoLog 1216
Resolution	12 bits
Accuracy	0.5% at 12 bits
Signal Recieved	0 - 2.5 V

TABLE 4.3: Uncertainty in measurements

Measurement	Working Range	Level of uncertainty
Inlet temperature	0 - 200°C	$\pm 1.73^{\circ}\text{C}$ ($\pm 0.7726\%$ of FS)
Outlet temperature	0 - 200°C	$\pm 1.73^{\circ}\text{C}$ ($\pm 0.7726\%$ of FS)
Inlet pressure	0 - 1000 kPa	± 11.49 kPa g ($\pm 1.153\%$ of FS)
Outlet pressure	0 - 1000 kPa	± 11.49 kPa g ($\pm 1.153\%$ of FS)
Mass flow rate	1.8 - 18 Nm ³ /h	± 0.29 Nm ³ /h ($\pm 1.6\%$ of FS)
Rotational speed	0 - 99999 rpm	± 50.0 rpm ($\pm 0.05\%$ of FS)
Force applied by brake	0.3 - 3 kg	± 0.039 kg ($\pm 1.118\%$ of FS)

95% confidence level using a coverage factor of 2 for the expanded uncertainty.
(FS: full-scale)

The experimental data gathered in the case of this study are classified as scarce, therefore a Type B uncertainty analysis according to ISO [80] was conducted. This analysis included the resolution and accuracy of the data logger, the accuracy of all of the measurement equipment used as well as the effect that the shunt resistor circuit might have had on the readings. Table 4.3 summarises the uncertainty of the measuring equipment used. In this study's case an expanded uncertainty was used with a factor of two, that provided a confidence level of 95%. See appendix B containing the MS Excel sheet with the calculations.

4.2.3 Experimental setup

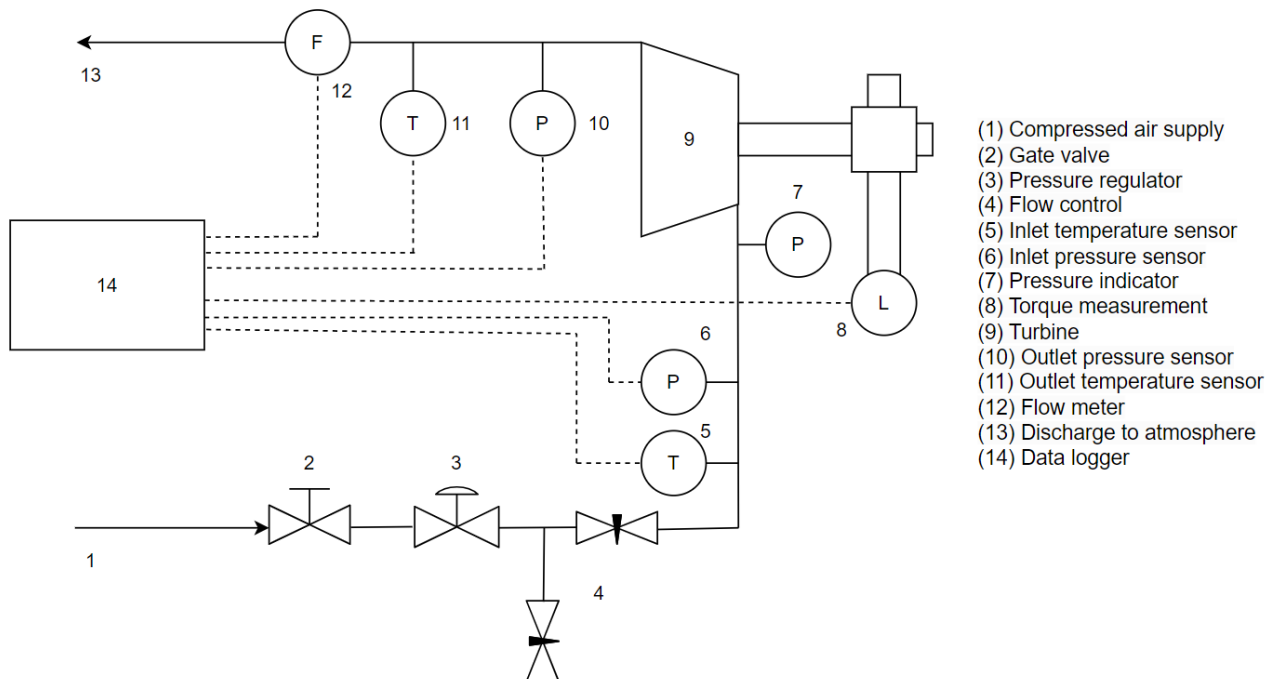


FIGURE 4.2: Process flow diagram of experimental setup

Figure 4.2 shows the process flow diagram (PFD) of the experimental setup. Compressed air at approximately 100 kPa g was used to characterise the turbine. Higher pressure and flow rates were not possible due to laboratory compressed air supply

limitations.

A gate valve (2) was used to open or close the experimental setup to the compressed air supply (1). This was done so that the experiment could be setup safely and no air would be passing through the experiment when it was not required.

From the gate valve (2) the air flowed to a pressure regulator (3). The pressure regulator (3) was able to withstand a maximum pressure of 10 bar. The pressure regulator (3) also provided coarse flow control. Needle valves were used as flow control valves (4), which were arranged in such a manner as to ensure accurate flow control.

After the needle valves (4) the air passed some instrumentation that recorded the pressure (6) and temperature (5), as well as a pressure indicator (7), before entering the turbine (9). This instrumentation was used to capture the data and the pressure indicator served as a visual indicator of the pressure and ensure a safe working pressure and to serve as a visual indicator when making adjustments. Turbine outlet pressure and temperature were measured at (10) and (11) respectively. All pressure and temperature sensors were connected to the DAQ (14).

A flow meter (12) just before the air exited to the atmosphere (13), was used to determine the flow rate of the air through the system. A rotameter was used, due to its affordability.

A prony type brake (8) was connected to the shaft of the turbine to measure the torque (8) produced by the turbine. The force exerted by the prony brake arm was measured using a load cell. This load cell also fed its data to the DAQ (14).

In order to determine the power produced by the turbine, the rotational velocity of the shaft is needed as well. To determine this a handheld tachometer was used. The end of the shaft was coated with a non-reflective tape. A small reflector strip was placed on the non-reflective tape in order for the tachometer to obtain accurate measurements.

The DAQ then recorded all of the results from the instrumentation over a certain period of time after steady state conditions were achieved. Then an average off all measurements were taken to determine the final values used for the calculations.

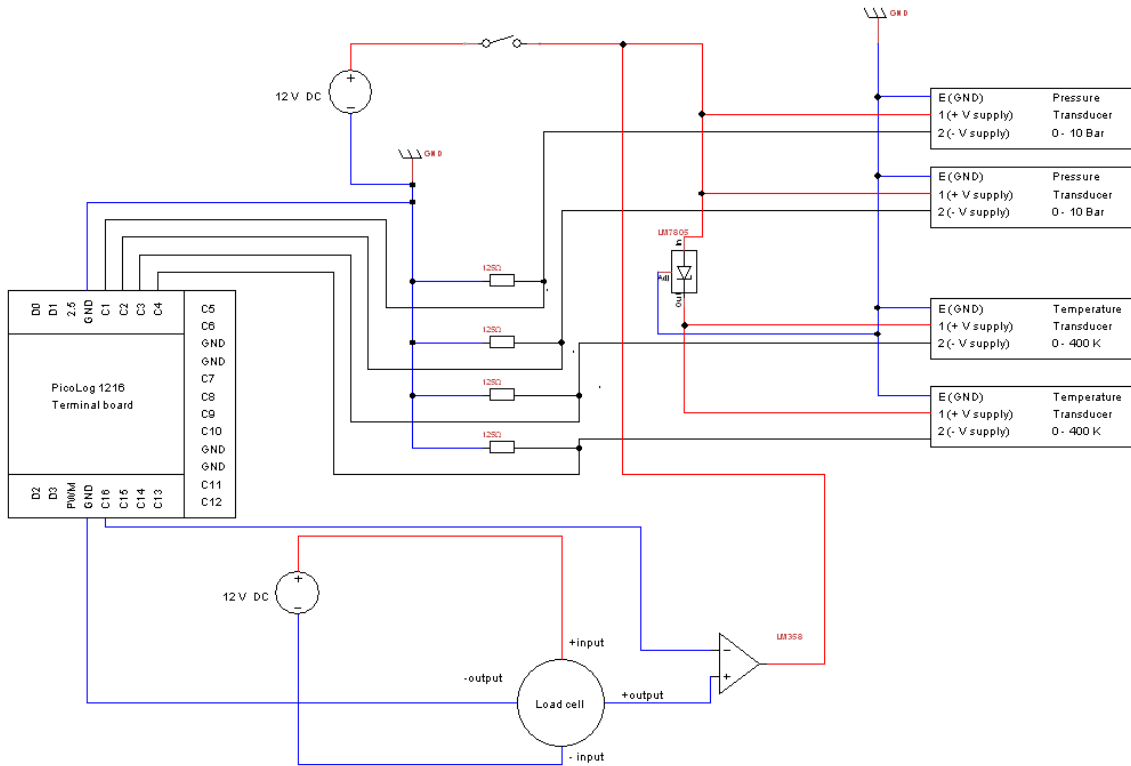


FIGURE 4.3: Wiring diagram of the experimental setup

The instrumentation was connected as per figure 4.3. A 12-volt direct current (DC) power supply powered the pressure sensors as well as the 5-volt power regulator. The 5-volt, voltage regulator in turn powered the RTDs. An LM7805 voltage regulator was used in order to reduce the voltage from 12 to 5 volts for the RTDs.

From the pressure sensors the signal went through 125 ohm resistors to convert the signal from 4 - 20 mA to 0 - 2.5V for the data acquisition system. The signal from the RTDs were also converted using resistors for the DAQ.

The load cell was powered with a separate 12 volt DC power supply. This was done to ensure that the load cell does not cause a short circuit with the other equipment. The load cell's signal was amplified with a LM 358 operational amplifier in order to ensure a signal that the DAQ can accurately measure.

The DAQ was connected and powered via USB to a laptop computer on which the data was recorded and exported to Microsoft Excel.

4.3 Experimental methodology

4.3.1 Setup of experiment

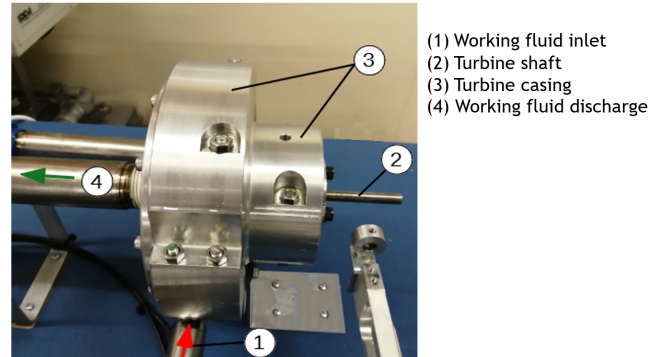


FIGURE 4.4: Turbine Assembly

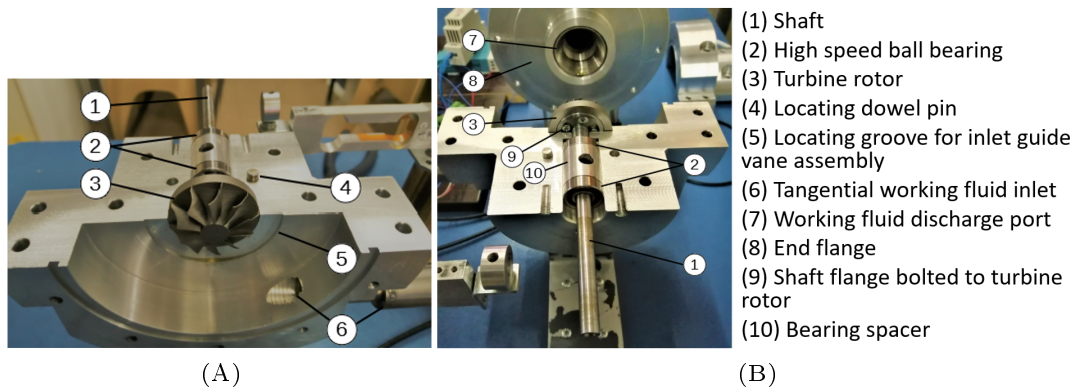


FIGURE 4.5: Turbine assembly with top split casing removed.

Figure 4.4, 4.5A and 4.5B shows the setup used for the experiment. Before the start of each run there was a specific startup procedure. The procedure was as follows:

- Insert the correct rotor and guide vane combination, as well as the correct spacers.
- Close the split casing and ensure that all seals, nuts and bolts are correctly in place.
- Place brake onto shaft and ensure that the brake arm is pressing correctly against the load cell. Also ensure that the correct mass is applied to the load cell to ensure consistent results.
- Use syringe to add oil to the bearing to ensure sufficient lubrication.
- Ensure that all valves and the pressure regulator are closed.
- Connect the DAQ to the laptop and switch on the power to the instrumentation. Ensure that all channels are displaying correctly on the DAQ window.
- Connect the system to the compressed air supply.

- Preload the loadcell in order to be at a better working range for accuracy and record 100 samples of the load applied.
- Open the gate valve completely and then slowly open the pressure regulator, while monitoring the flow meter.
- Adjust the flow rate using the pressure regulator, as well as the needle valves until the desired flow is achieved.
- Let the experiment run until steady state conditions are achieved.
- Record 100 samples of inlet pressure and temperature, outlet temperature and pressure and force on the load cell.
- Use the manual tachometer to record the rotational speed of the shaft.
- Ensure that the data was captured, then close the pressure regulator.
- Close all valves and disconnect the system from the compressor.
- Export data to MS Excel and save.
- Repeat cycle as many times as needed to record all data sets.

All the raw data captured were exported to MS Excel. From the raw data, the data was then summarized as seen in appendix C. Consequently, data reduction and calculations were carried out in MS Excel and Engineering Equation Solver (EES).

4.3.2 Experimental conditions

For the most part, conditions inside the laboratory was kept as stable as possible, however there was some slight variation in the room temperature and atmospheric pressure. The atmospheric pressure was determined by using three different weather stations' recordings and taking an average. The variation in ambient temperature was not a problem as it had little influence and was recorded by the RTDs in real time during the experiment.

The temperature of the compressed air supply varied slightly from test to test; this was dependant on the demand on the compressor as well as the temperature of the air stored in the reservoir. This variation was however recorded using the RTDs. After steady state conditions were achieved 100 values were recorded and an average temperature was used.

4.3.3 Recording of results

The data was recorded using a Picolog 1216 data logger. A summary of the raw data can be found in appendix C.

Test No	0
Date	2019/05/21
Settings and configuration	
Desired Air Flow Rate	40
Rotor Blade Inlet Angle	1
Rotor Blade Outlet Angle	1
Working Fluid	Air
Loadcell Pre Load	400g
Friction weight	100g
Checks: Pre-start	
Turbine number	1
Guide vane number	x
Gate valve closed	x
Pressure regulator closed	x
Sufficient oil	x
Data acquisition	x
Working fluid supply	x
DAQ	
Temperature probe in	x
Pressure sensor in	x
Pressure sensor out	x
Temperature probe out	x
Load cell	x
Tachometer	x
Start-up & Test	
Open gate valve	x
Open pressure regulator	x
Adjust flow and pressure	x
Achieve steady state	x
Temperature in (°C)	17,07414
Pressure in (Pa)	6092,78741
Pressure out (Pa)	2119,96604
Temperature out (°C)	18,11914
Flow rate (m ³ /H ₂ O)	40
Shaft speed (rev/min)	1997
Rotational load (g)	368,891
Static load (g)	354,236
Net load (g)	14,655
Checks: Stop	
Gate valve closed	x
Pressure regulator closed	x
Working fluid disconnected	x
Data saved	x
Check/replace/drain oil	x

FIGURE 4.6: Data recording form

Figure 4.6 shows an example of the data recording form that was used to record the data. This data recording form also included a checklist of the necessary steps required before commencement of the experiment. This was done as a control to ensure that conditions were as consistent as possible and that the experiment was carried out safely whilst minimizing experiment execution errors.

The data on this form was used for all of the experimental calculations. The inlet temperature and pressure as well as the outlet temperature and pressure were used to calculate the theoretical performance of the turbines.

Chapter 5

Results and discussion

5.1 Introduction

To ensure repeatability of measurement results, five tests were conducted per configuration. Although the pressure ratios were less than the design pressure ratios it was still deemed sufficient for characterisation of the turbine configurations at low pressure ratio. The laboratory compressed air supply could not provide the pressure at the flow rate required to achieve pressure ratios that were near the ideal situation for the turbines. This was due to the fact that the turbines were tested in open conditions and not a closed loop as it would function in practise. For this reason, comparing the theoretical ideals would not be applicable to what was experimentally achieved, as the efficiencies of the turbines would be very low.

TABLE 5.1: Example of data recording sheet.

Test no	12401	12402	12403	12404	12405
Date	2019/05/17	2019/05/20	2019/05/21	2019/05/22	2019/05/23
Rotor blade number	1	1	1	1	1
Guide vane number	2	2	2	2	2
Rotor blade inlet angle ($^{\circ}$)	65	65	65	65	65
Rotor blade exit angle ($^{\circ}$)	0	0	0	0	0
Load cell pre load (g)	400	400	400	400	400
Brake arm length (mm)	80	80	80	80	80
Friction weight (g)	100	100	100	100	100
Temperature in ($^{\circ}C$)	18.87	16.522	16.067	16.862	17.216
Temperature out ($^{\circ}C$)	18.805	18.566	18.002	18.668	19.235
Atmospheric pressure (Pa)	86550	86450	86650	86650	86750
Gauge pressure in (Pa)	6024.11	5079.72	5634.92	5714.29	5636.45
Gauge pressure out (Pa)	2109.2	2026.8	1938.3	1756.7	1730.7
Absolute pressure in (Pa)	92574	91529	92284	923643	92386
Absolute pressure out (Pa)	88659	88476	88588	88406	88480
Flow meter calibration density (kg/m^3)	8.319	8.319	8.319	8.319	8.319
Working fluid density (kg/m^3)	1.0579	1.0566	1.0599	1.0554	1.0542
Flow rate (kg/h at 7 bar)	40	40	40	40	40
Flow rate (kg/s)	0.00111	0.00111	0.00111	0.00111	0.00111
Shaft speed (RPM)	1915	1993	1977	1951	1963
Operational weight (g)	373.061	376.808	373.353	370.659	374.353
Static weight on load cell (g)	354.598	358.401	353.721	351.088	355.768
Net load on load cell (g)	18.463	18.407	19.632	19.571	18.585
Power generated (W)	2.906	3.015	3.190	3.138	2.998

Table 5.1 is an example of the sheets that were used to record the data and have it in a usable form. Firstly, the test number was recorded to keep track of each test. The first numeral refers to the rotor number. The second to the guide vane number. The next two numerals refer to the target flow rate and the last numeral in the test

number of the specific rotor and guide vane combination. For example: Test number 12403

- 1 - Rotor number 1
- 2 - Guide vane number 2
- 40 - Flow rate displaying on flow meter of 40kg/hour
- 3 - Test number 3 using rotor 1, guide vane 2 and a flow rate of 40kg/hour

The load cell was pre-loaded with 400 g of weight. This was done to operate the load cell at a better range to improve the accuracy of the measurements. The load cell was rated to measure from 300 g to 3 kg. 100 g of mass was also added to the brake in order to increase the friction between the shaft and the prony brake. This was done to slow the shaft down using friction. This friction was then used to calculate the torque with:

$$P = (2\pi NT)/60 \quad (5.1)$$

where,

$$T = FR \quad (5.2)$$

5.2 Results and discussion

In order to determine the stagnation enthalpy, compressible flow equations from [54] were used to determine the Mach number at inlet and outlet using:

$$M = c/a = c/\sqrt{\gamma RT} \quad (5.3)$$

From there the inlet and exit temperatures were calculated as follows:

$$T_0/T = 1 + ((\gamma - 1)/2)M^2 \quad (5.4)$$

The inlet and exit pressure were calculated similarly using the following:

$$p_0/p = (1 + ((\gamma - 1)/2)M^2)^{\gamma/(\gamma-1)} \quad (5.5)$$

These equations were used in an Excel spread sheet to calculate the stagnation temperatures and pressure for each test run. Using Engineering Equation Solver (EES), this table was used as an input to calculate the enthalpy at each of these points. The last value required is the isentropic enthalpy at the exit which was also calculated in EES using the inlet entropy. These values can then be used in to determine the theoretical total to static efficiency of the turbine as follows:

$$\eta_{ts} = (h_{02} - h_{03})/(h_{02} - h_{03ss}) \quad (5.6)$$

During the testing a total of 225 tests were performed. The theoretical values in the following chapters is calculated using the inlet and exit conditions per run. Therefore, each run has a theoretical result and an actual result. The actual torque was the torque measured on the shaft. Whereas the theoretical torque was the expected theoretical result using the inlet and outlet conditions. See appendix D for the complete calculation. The enthalpy and entropy were calculated using EES, which can use a

parametric table from MS Excel with the relevant input values. See appendix E for the EES equations.

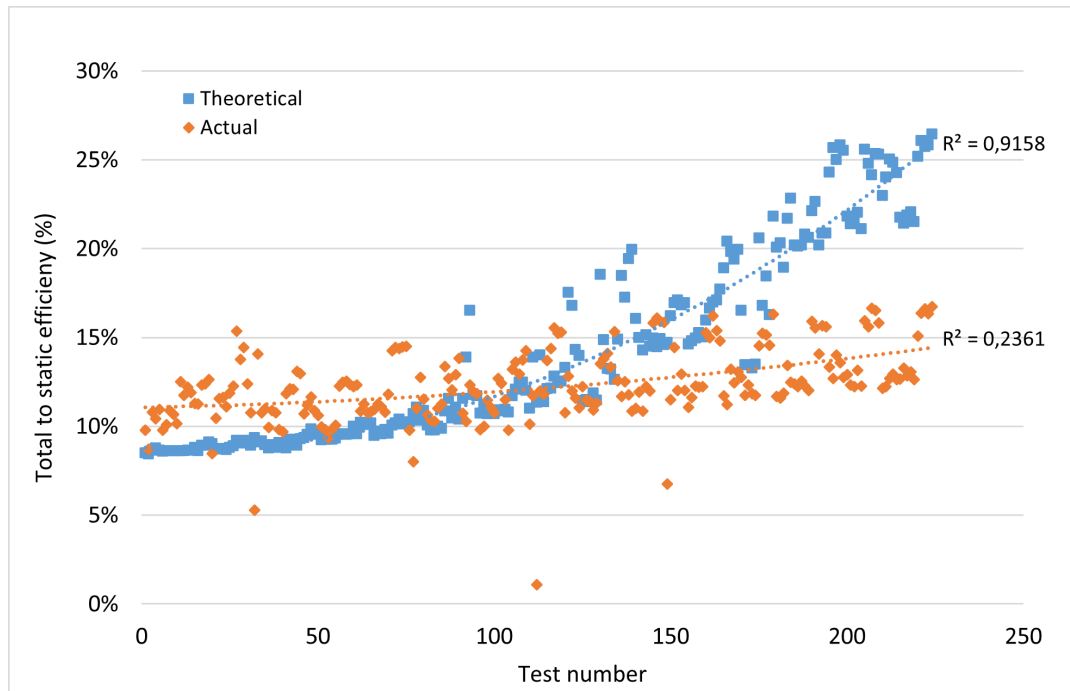


FIGURE 5.1: Theoretical and actual total-to-static efficiency for all tests

The theoretical efficiency in Figure 5.1 was calculated by assuming a constant mass flow through the system. The mass flow rate was then used to calculate the velocity of the working fluid at the inlet and exit. The rotational velocity of the turbine could be measured more accurately and therefore was used with the velocity to calculate the theoretical efficiency of this turbine. This method does not inherently calculate the theoretical losses. However, the result of the turbine speed includes the effect of them. The coefficient of determination (R-squared) value using this method is 0.9158, thus providing results which is usable to compare to the actual measured power. The actual measured power can then be compared with the theoretical power of the run, in order to determine the efficiency of the turbine.

The following results and figures describe the experimental results. This was done to determine what the actual differences between configurations were. The actual force measured on the brake together with the rotational velocity of the turbine were used to calculate the torque and power that was produced in order to depict these differences.

TABLE 5.2: Tested inlet guide vane and rotor configuration combinations.

Description	Nozzle inlet angle (°)	Rotor blade exit angle (°)	Clearance (%)
Config11	65	0	2.44
Config12	65	0	3.61
Config13	55	0	2.44
Config21	65	25	2.44
Config22	65	25	3.61
Config23	55	25	2.44
Config31	65	50	2.44
Config32	65	50	3.61
Config33	55	50	2.44

Table 5.2 shows a summary of the differences between the different experimental configurations. This table is useful for the graphs that follows.

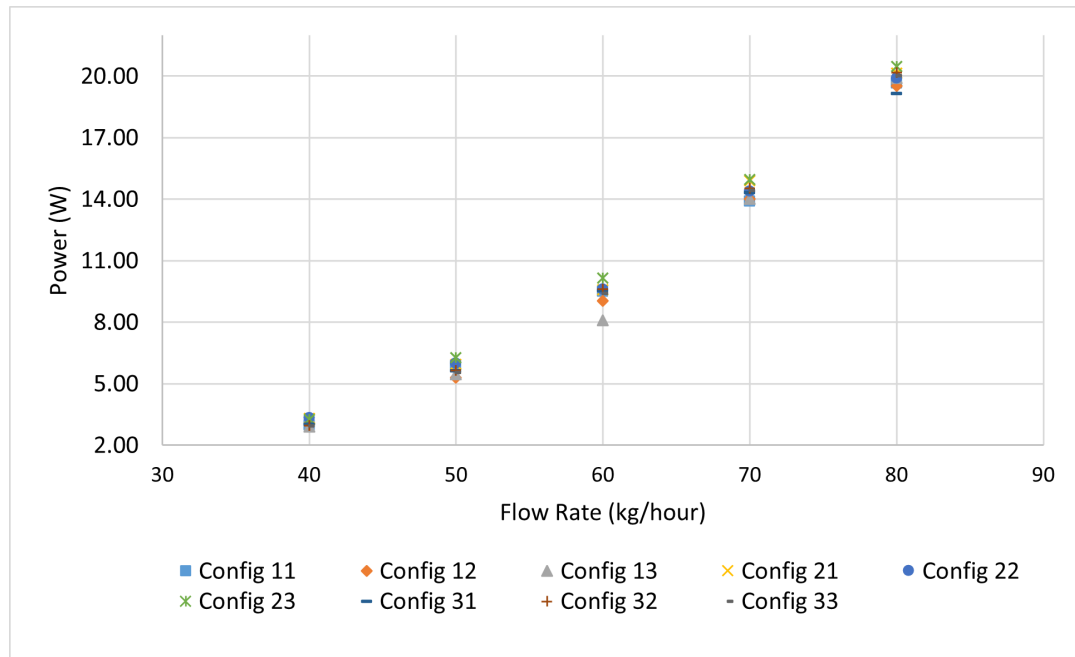


FIGURE 5.2: Measured average power of turbines

Figure 5.2 shows the average power produced by the turbines at the different flow rates used in this experiment. It is clear that as the flow is increased the effect that the inlet angle, exit angle and clearance has, also increases. Although the power measured was significantly lower than the initial design the quality of the data seems to be of a good consistent quality.

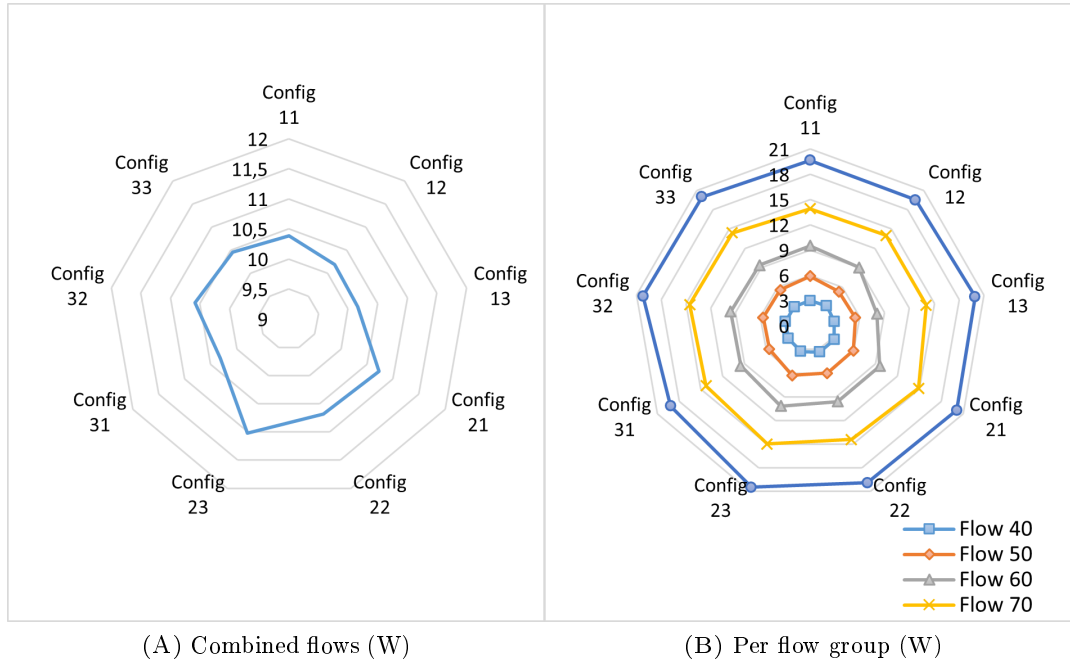


FIGURE 5.3: Average power measured per configuration.

Figure 5.3A shows the different average power generated when all the flow groups were combined. Whereas 5.3B shows the average power generated per configuration at the various flow rates. From figure 5.3A there seems to be four clear configurations that achieved a higher output of power. These groups were configurations 21, 22, 23 and 32. However, these differences appear to be somewhat smaller when looking at it per flow group.

Following the differences between the configurations will be investigated further. The total-to-static efficiencies were calculated for the entire turbine. The data used to compare the differences were sorted to compare test runs where the same guide vanes were used with the different rotors to determine the influence of the rotors. Whereas, to determine the influence of the guide vanes; similarly, the data was used where the same rotors were used.

In order to compare the efficiency the turbine total-to-static efficiencies were calculated theoretically using [54]:

$$\eta_{ts,theoretical} = P_{theoretical}/P_{available} \quad (5.7)$$

where:

$$P_{theoretical} = 0.5[(u_2^2 - u_3^2) - (w_2^2 - w_3^2) + (c_2^2 - c_3^2)] \quad (5.8)$$

With all the values calculated theoretically from the experimental conditions.

and:

$$P_{available} = 0.5[(u_2^2 - u_3^2) - (w_2^2 - w_3^2) + (c_2^2 - c_3^2)] \quad (5.9)$$

With all the values calculated using vector diagrams as discussed in chapter 3 and assuming ideal conditions.

The actual efficiency was calculated similarly [54]:

$$\eta_{ts,actual} = P_{actual}/P_{available} \quad (5.10)$$

Where:

$$P_{actual} = \tau n \quad (5.11)$$

With the data measured from the prony brake and the rotational speed of the shaft.

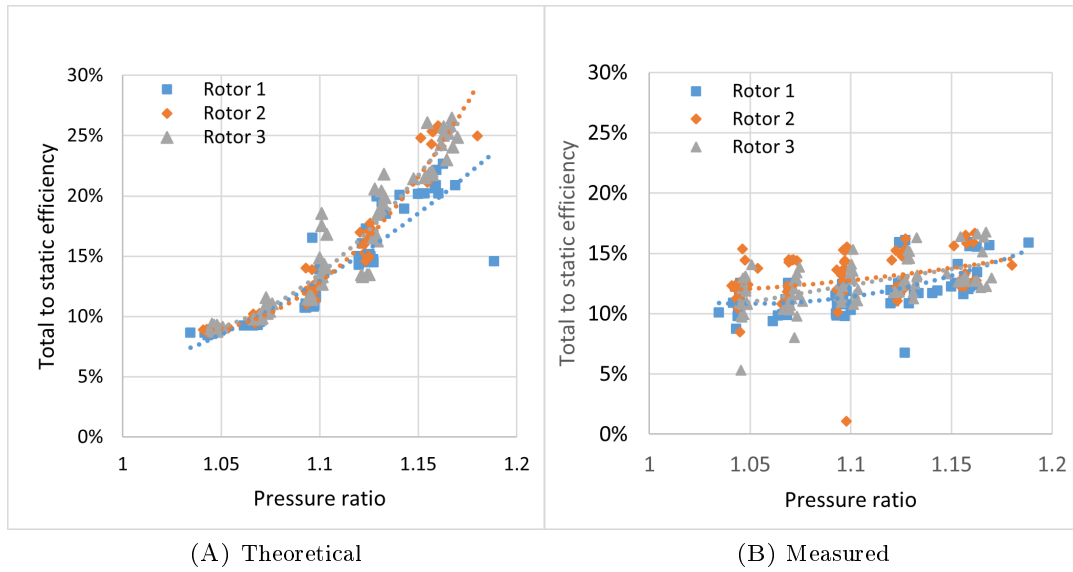


FIGURE 5.4: Total to static efficiency comparison between different rotors.

Figures 5.4(A) and 5.4(B) above depict the total to static efficiency at different pressure ratios for each of the rotor blades irrespective of the guide vane angle. The rotor blade exit angles for rotors 1, 2 and 3 are 0° , 25° and 50° respectively. In the case of the theoretical total-to-static (tts) efficiency the difference between rotor configurations increase with an increase in pressure ratio; noting that the theoretical tts efficiency was calculated from the measured turbine rotational speed. There is no significant theoretical tts efficiency difference between rotors 2 and 3.

In the case of the measured tts efficiencies, Rotor 2 has the best tts efficiency at low pressure ratios. The measured tts efficiencies of all three rotor configurations converge with an increase in pressure ratio. It seems clear that rotor 1 appears to be the least efficient, with the difference between 2 and 3 being very close to each other. Generally, the total-to-static efficiency increases with an increase in pressure ratio.

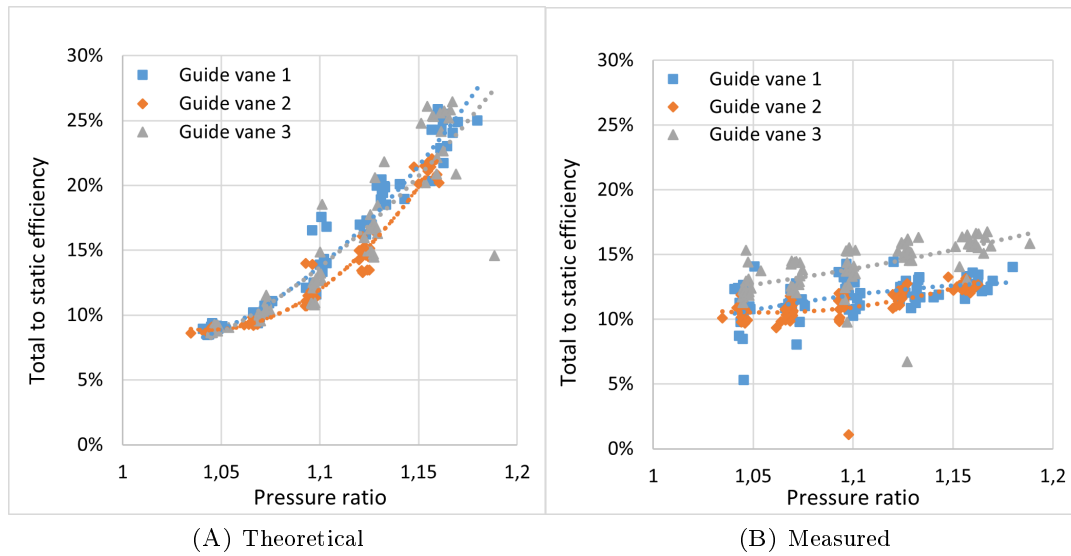


FIGURE 5.5: Turbine total-to-static efficiency when comparing of different rotor configurations

Figures 5.5(A) and 5.5(B) similarly shows the differences between the 3 different guide vanes. Guide vane 1 and 2 has exit angles (turbine inlet angles) of 65 degrees each and guide vane 3 has an angle of 55 degrees. Where guide vane 2 has a clearance of 3,61% while the other two have clearances of 2,44% each. This clearance is a percentage of the passage height and is the distance between the rotor inlet tip and the guide vane exit. In both the theoretical as well as the measured cases it appears as though the extra clearance does have a greater negative impact on the performance of the turbine. 5.5(B) shows that guide vane 3 is clearly the most efficient however guide vane 1 does seem to be improving as the pressure ratio was increased getting closer to the efficiency of guide vane 3.

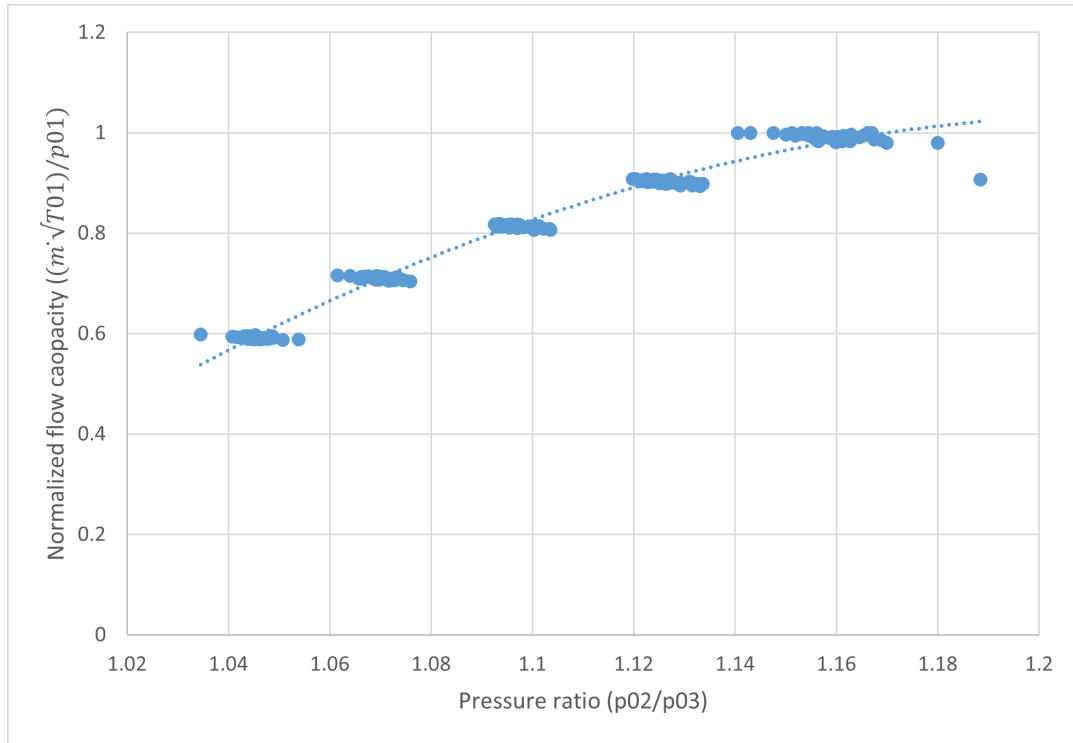


FIGURE 5.6: Normalized flow capacity ($\dot{m}\sqrt{T_{01}}/p_{01}$) vs measured pressure ratio.

Figure 5.6 above shows that as the pressure ratio is increased the flow rate through the turbine also increases. It also appears to be increasing less as the pressure ratio increases. This would indicate that there is a limit to the increase in flow rate by increasing the pressure ratio. At some point increasing the pressure ratio further would no longer significantly improve the flow rate through the turbine. Dixon [54] shows in chapter two that as the pressure ratio increases the normalized flow capacity also increases. However, at some point the flow starts to choke and no significant gains are further obtained.

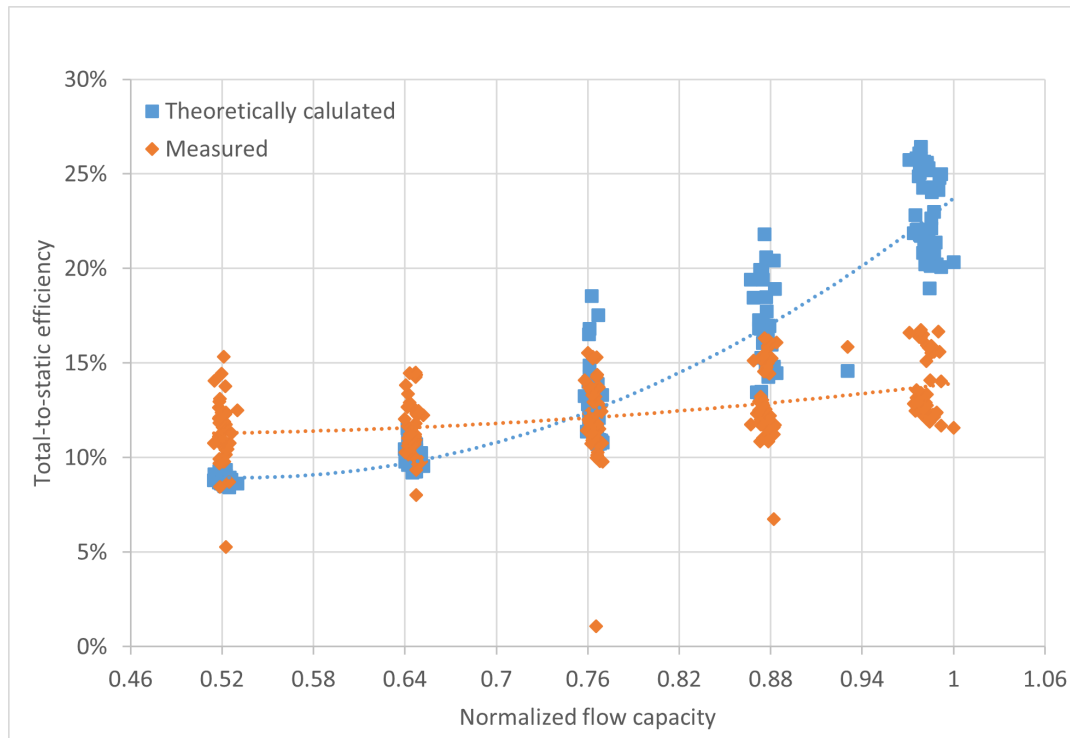


FIGURE 5.7: Total-to-static efficiency vs Normalized Flow Capacity

Theoretically the increasing flow rate through the turbine improves turbine total-to-static efficiency. There is a noticeable increase in the measured total-to-static efficiency, but not as significant as is the case for the theoretical case. The results suggest that these turbines should be operated at higher flow rates. The low total to static efficiency is due to the turbine operating far from its design point.

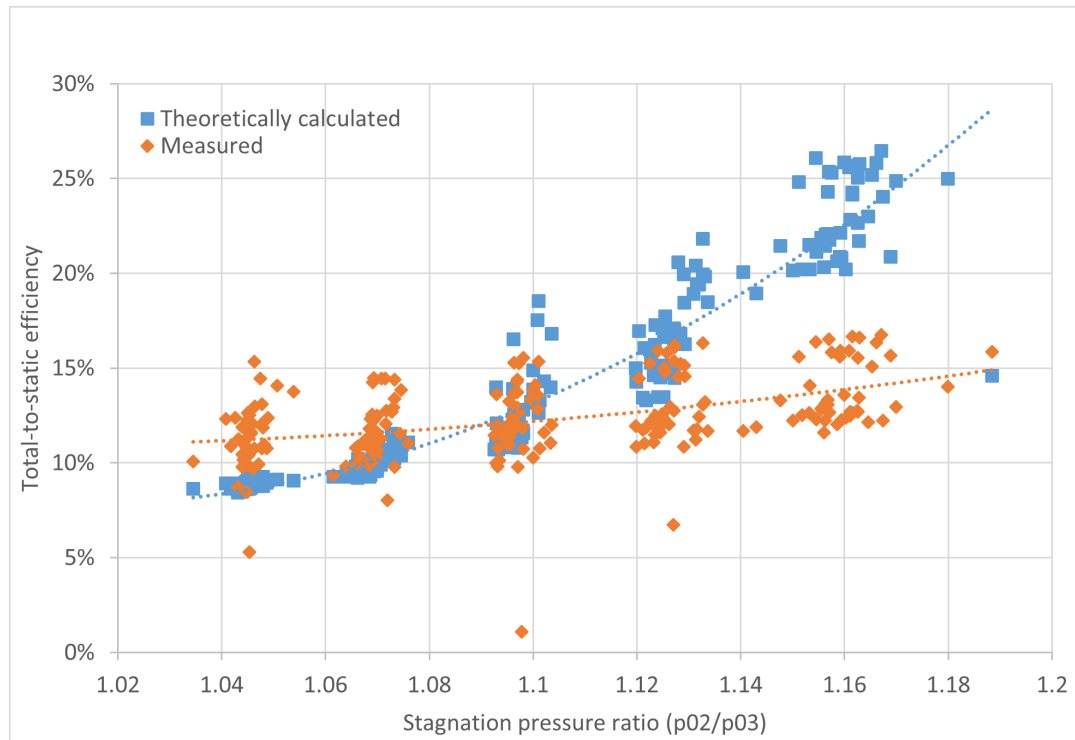


FIGURE 5.8: Total-to-static efficiency versus stagnation pressure ratio.

Figure 5.8 shows the effect that the stagnation pressure ratio has on the efficiency of the turbines. In general, as the stagnation pressure ratio is increased the total-to-static efficiency increases for both the theoretical and measured cases. In general turbine performance increases with an increase in pressure ratio and flow rate until choking conditions are reached after which further increase occurs [54].

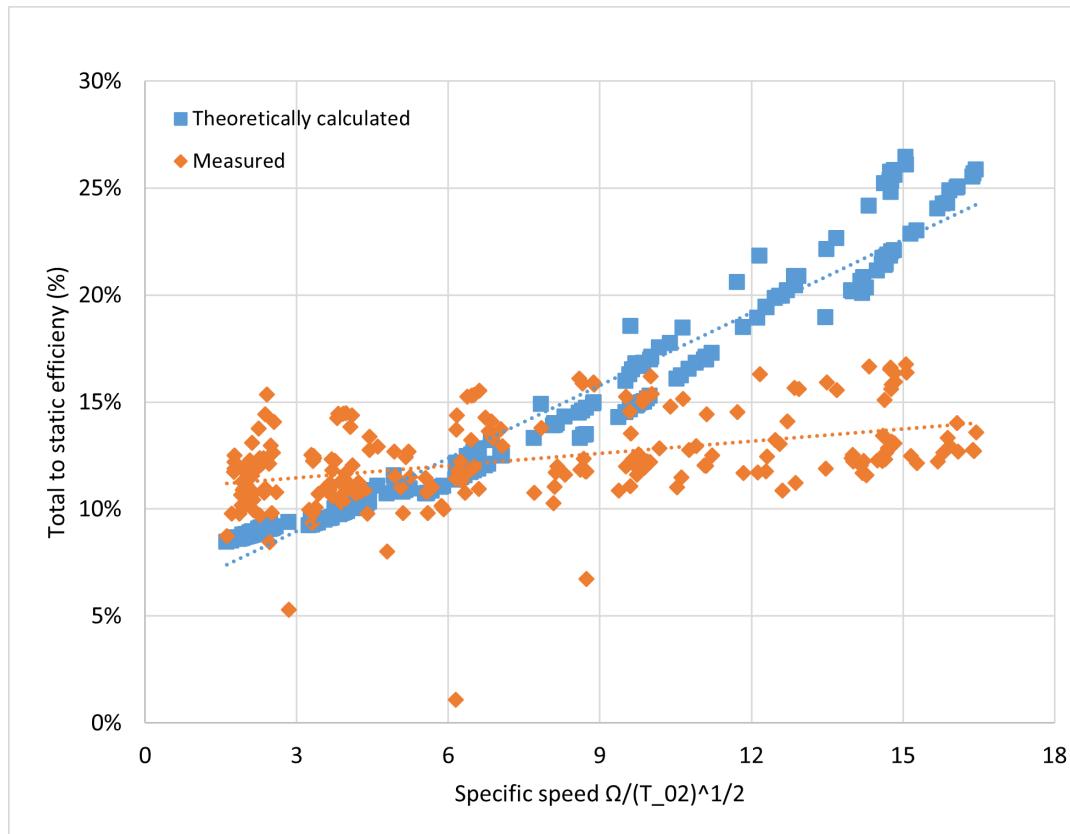


FIGURE 5.9: Efficiency vs Specific Speed

Figure 5.9 in a very similar trend shows that there is more performance available, if operated under better conditions. These turbines appear to operate better at a higher specific speed. The actual increase in this value seems to be quite linear. It would be interesting to see what would occur at higher temperature differences as well as when using different working fluids. From the results presented above it is clear that the turbine efficiencies were low, which is a result of the turbines being tested at off-design conditions far from the design point. It is expected that these turbines should be much more efficient at more ideal operating conditions.

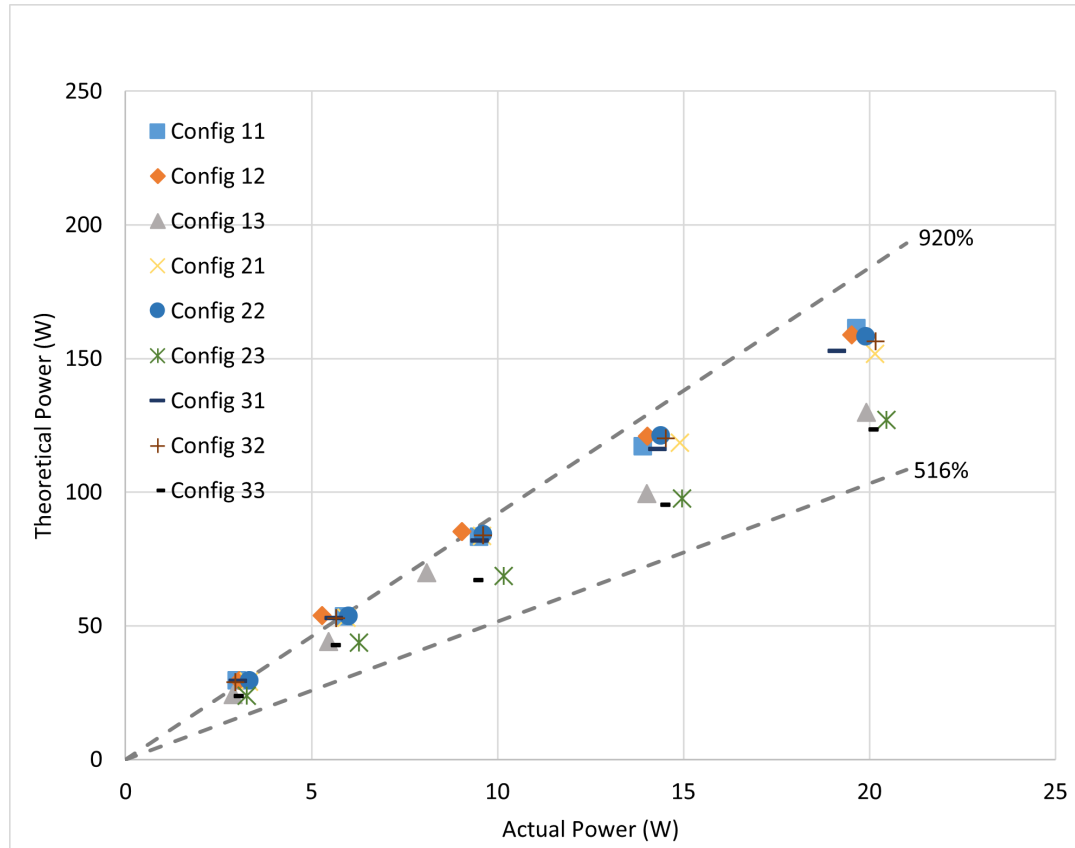


FIGURE 5.10: Theoretical versus actual power for all turbine and guide vane configurations.

The theoretical power vs the actual power measured of the turbine show how far off the ideal conditions the experiment operated, with an error between 500% to 900%. The error appears to be reduced as the power (pressure ratio) is increased which which is to be expected as the turbines were designed to operate at higher temperatures and pressures.

However, with this large error it would have to be investigated further and the turbine need to be tested at its design point. Ideally it is recommended that testing is done with a closed looped system that can be operated at a broad range of pressure ratios. So that the turbine can be characterized over a broad operating range.

Another factor that contributes to the difference between the theoretical and experimental results is the losses that occurs during operation. As discussed in chapter 3.2.5, there are different factors that will reduce the efficiency of the turbines. In these calculations these losses were not accounted for due to the following two reasons. Firstly, to compare the ideal turbine with the actual experimental results. Secondly, due to the turbine having been tested far from the ideal operating conditions it would be extremely difficult to differentiate between losses and lack of performance due to the operating conditions.

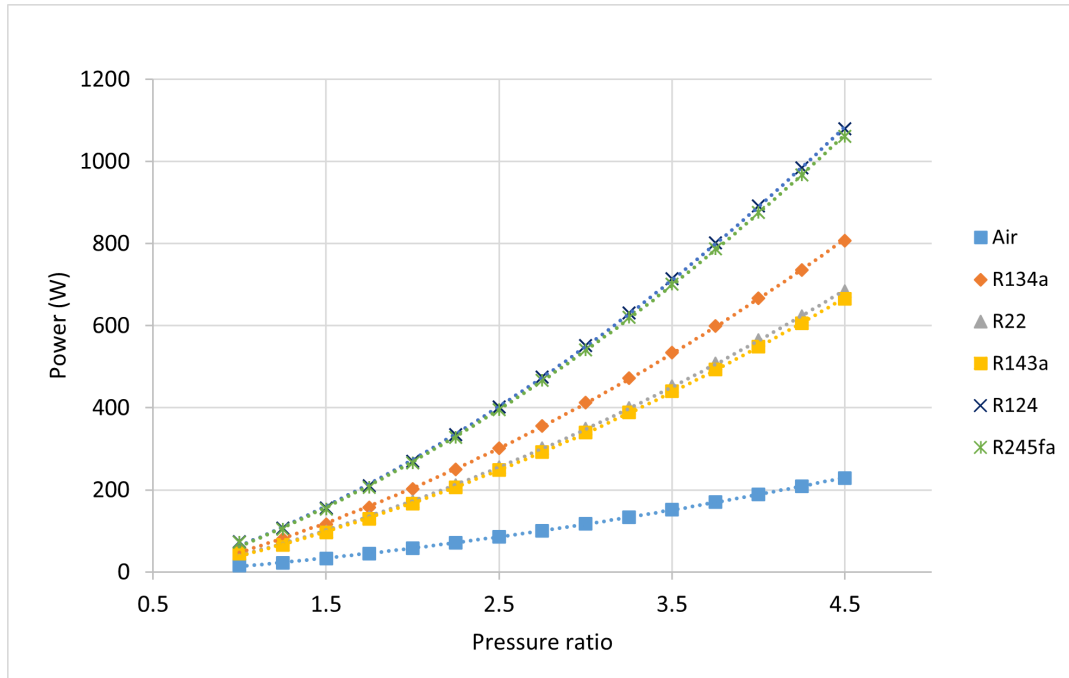


FIGURE 5.11: Predicted performance with different organic working fluids as derived from dimensional analysis.

Dimensional analysis and dynamic similarity were used to predict the performance of the turbine when using organic fluids as working fluid. Figure 5.11 above shows an idealistic extrapolation of what the increase in performance of the turbine could be if the supply pressure is increased for air and a selection of organic fluids. For this graph it is assumed that the temperature and efficiency will remain the same as with the experiment. Most likely one would expect efficiency to increase as the pressure ratio increases.

These values were calculated with the following equation by using the efficiency that was calculated experimentally and the densities of the fluids at the experimental conditions [54]:

$$P = \rho \rho g Q \eta \quad (5.12)$$

See appendix F for the full set of results.

Figure 5.11 shows that at these low temperatures the heavier working fluids like R245fa and R124 performs better. This is also partly due to the temperature differential being negligible in the experiment. However, most refrigerants would be a better alternative to use in place of air especially if the temperature and pressures are increased to more ideal conditions.

It would be a good continuation of this study to determine the efficiency of turbines using these different organic working fluids close to their ideal operating temperatures. The selection would have to be based upon the system in which it will be used as that will determine the temperature available to do work in the system.

Chapter 6

Summary and conclusions

6.1 Summary

Greenhouse gas emissions from fossil fuels contribute significantly to climate change and global warming. Utilising Organic Rankine Cycles (ORCs) to harvest waste and low grade heat assist in improving the efficiency of fossil fuel derived power and heat systems. Hence the utilisation of ORCs directly assist in reducing greenhouse gas emissions. AM is an attractive production method, especially for the small and intricate components required to produce such systems, with the turbine being one of them.

The advantage of AM is the ability to simplify the design of such turbines by reducing the parts count and reducing production time. However, care must be taken with the design to ensure that reduction of parts count and production time is achieved. The turbine components must be designed to keep supporting structures, during manufacturing, to a minimum and make them easy to remove during post processing. Care should also be taken to reduce the parts count such as reducing the number of fasteners.

In this study a 90° radial inflow turbine was designed and manufactured from titanium alloy Ti6Al4V by means of selective laser melting. Three rotor configurations and three inlet guide vane configurations were tested. Initial testing was done with air as working fluid and the performance of the turbine with organic working fluids was estimated with dimensional analysis.

The turbines were able to withstand the velocities, pressures and vibrations that they underwent in the experiment. This show that AM seems to be a viable method to produce these turbines. However destructive testing would be required to determine mechanical limitations and the strength of the turbines.

The turbines performed worse than was predicted by the ideal design. This is attributed to the off-design test conditions. A contributing factor was the small size of the turbines. In small turbomachines efficiencies are generally lower than the efficiencies of large turbomachines. The performance of the turbines changed as expected when changing the design or the experimental conditions. The results were also consistent when changing the conditions providing reliable data. This is positive in terms of the experimental setup and for future studies. With a few modifications to the experimental and turbine design the benefits of AM could be addressed in further studies.

6.1.1 Turbine design and methodology

The turbines were created using known theoretical methods for 90° Ti-6Al-4V radial inflow turbines. Whereby the initial design was done using velocity triangles and theoretical energy using the Mollier diagram. From these base values it was possible to determine the geometry and blade profiles to be used in the experiment. The actual turbines were designed and manufactured to be as close to these values as possible while still being practical to manufacture. Three variations were designed of both the rotor blades and the inlet guide vanes. This was done to test multiple inlet and exit angles, as well as the influence of clearance between rotor outer diameter and inlet guide vane inner diameter.

6.1.2 Experiment design and methodology

Important parameters like the inlet/exit, temperature and pressure and the flow rate of the working fluid were used as driving parameters for the experimental setup. It was further also important to measure the power that was produced by the turbines under various conditions. The turbines were tested with air as the working fluid at multiple flow rates to characterize the performance of the turbines. The data was collected via an electronic data logger.

6.1.3 Experimental results

From the results of the testing as discussed in chapter 5 a total of 225 tests were run. Broken down it was 9 different configurations, each tested 5 times at 5 different flow rates. The experiment provided repeatable and consistent results per setup, thus giving confidence in the data that was collected. Minor deviations are suspected to be mostly due to slight changes in temperatures of the working fluid and slight differences in the mass flow rate.

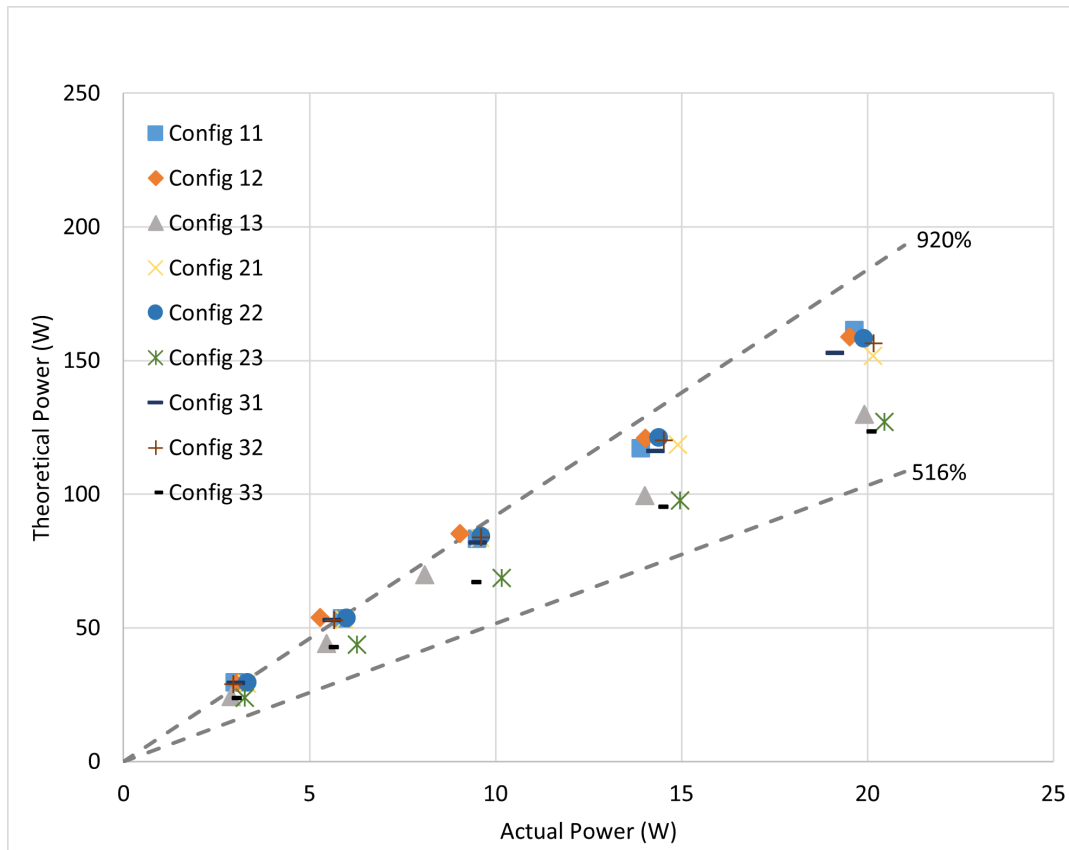


FIGURE 6.1: Theoretical versus actual power for all turbine and guide vane configurations.

There was a significant error between the theoretical expected results and the actual experimental values as can be seen in figure 6.1. This was mostly due to the off-design conditions at which the turbines were tested due to limitations in the experiment setup as previously discussed.

The efficiency of small turbomachines are significant lower than larger turbines. This is caused by the influence of losses that are proportionally larger on small turbomachines. Boundary layer losses are proportionally larger on smaller turbomachines since the boundary layer formation is a function of fluid properties, flow rates and pressure. Leakage losses between rotor and stator are also proportionally higher with a decrease in turbomachine impeller diameter. Mechanical losses are also proportionally higher for smaller turbomachines. For the above reasons it is to be expected that the efficiency of the small experimental turbine will be lower than for the theoretical turbine.

However, figure 6.1 show that even though there was a significant deviation from the theoretical performance that the data was consistent. Furthermore, as the flow rate increased, the results trended closer to the theoretical values. Therefore, indicating that higher pressure ratios would likely lead to the turbines being more efficient.

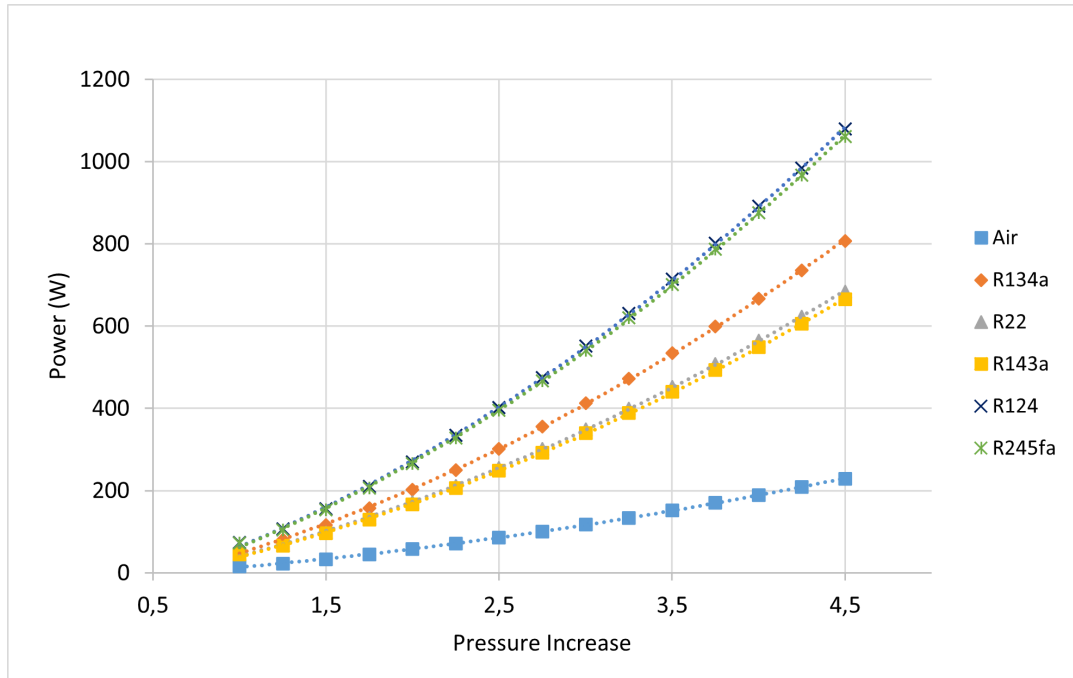


FIGURE 6.2: Dimensional Analysis of Different Working Fluids

Using the values obtained from the experiment and applying the principles of dynamic similarity, which assumes that the turbine efficiency remains the same, the results were extrapolated for the expected performance at higher pressure ratios. This was done for air as the working fluid as well as a selection of refrigerants. From figure 6.2 it can be seen that the expected power of the turbines would increase significantly for air as the pressure ratios moves closer to the ideal operating values for a 90° radial inflow turbine. Furthermore, the prospect of using refrigerants should result in a significant increase in performance of the turbine. This would be mostly due to the increased density of the working fluid and being able to carry more energy at lower temperatures.

6.2 Lessons Learned

6.2.1 Design for Additive Manufacturing

The majority of the findings is of a practical nature mostly focusing on the production of the turbines using additive manufacturing. With the methods used to produce the turbines and guide vanes a few lessons were learned. These lessons will ensure that production of the turbines are more efficient in terms of time and cost. The production of the turbine rotors and guide vanes were quick, but the post processing took a significant amount of time.

This post processing time includes removal of the supporting structures from the rotors and guide vanes. The rotor blades had a section of difficult supporting structures to remove as well as holes that needed to be tapped after being produced with AM. The guide vane inserts also had to be produced in two parts so that the supporting structures could be easily removed, adding to the parts count of the turbines.

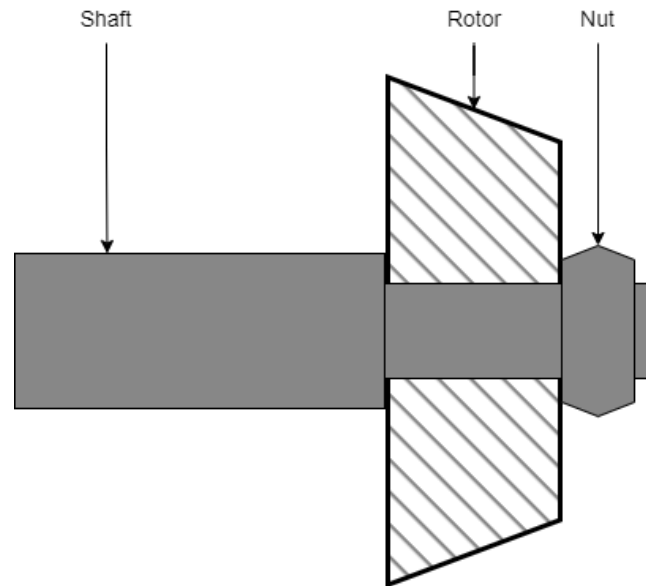


FIGURE 6.3: Alternative shaft design

Regarding the difficult supporting structures on the rotor blades. This could have been easily avoided with a simple design change as seen in figure 6.3. By allowing the shaft to pass through the centre of the rotor blades and fixing the rotor at the top with a locking nut. Supporting structures would not have been necessary at the bottom of the turbine rotors. It would also remove the need to drill and tap holes at the bottom of the turbine rotors.

This modification would have significantly reduced the post processing time as well as improve the assembly of the rotor blades to the shaft and the casing. Due to that fact that the turbine rotors were fastened at the bottom with screws it hindered changing to different configurations as the whole shaft needed to be removed in order to change a turbine rotor.

Due to the turbine rotor blades being very thin they got damaged during the first attempt of production. This happened due to the material scraper on the DSLM machine scraping against the turbine rotor blades. This was rectified by producing a cylinder around the turbines in order to stabilise the scraper and ensure that the blades were not damaged.

Another promising concept would be to manufacture the shaft and rotor as one part using additive manufacturing. This would however require some carefully planned designs of the rotor and rotor blades to minimize the need for supports during printing. This could provide an attractive manufacturing method due to greatly reducing the number of parts needing to be produced.

6.2.2 Experiment Design and Manufacturing

Due to the experiment being an open loop system a very large compressor had to be used in order to get a usable amount of pressure with sufficient flow and capacity to run the experiment. Even when using the large compressor, the pressure ratios were limited and could not run the turbine indefinitely. Although compressed air supply

lasted long enough to reach stable conditions, it did limit the maximum pressure ratios at which could be tested.

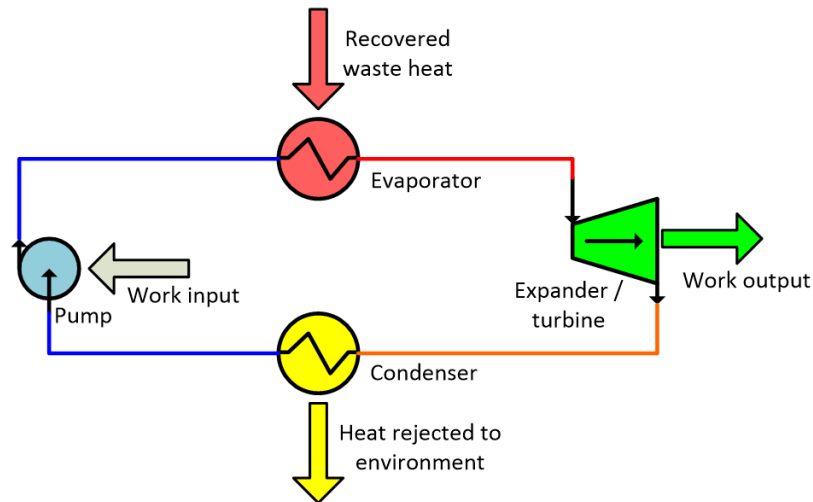


FIGURE 6.4: Closed loop Organic Rankine Cycle system

One major improvement to the experiment would be to have a closed loop Organic Rankine Cycle system as shown in figure 6.4. This will enable testing the turbine at much higher-pressure ratios and remove the atmospheric pressure having a negative effect on the turbine. This is due to an open loop system having to firstly overcome atmospheric pressure before any work can be done. Whereas a closed system does not have this limitation. It would also simplify the process of testing with other organic working fluids, which would also be more beneficial than working with air.

Initially the casing was designed with only the front end on the turbine discharge side, being able to open. This provided some issues, because the turbine rotor had to be fixed to the shaft and the slid into the casing and supporting bearings. With the slide fit the turbine started to float (moving forwards and backwards) in the casing, causing some vibrations at low speeds. Another issue with a sliding fit was that at higher speeds the clearances between the bearings and the shaft as well as the internal clearance of the bearing would cause vibrations.

The solution to the problem was to manufacture a split casing. This meant that the shaft could be press-fitted into the bearings, taking into account the internal clearance of the bearings as well as eliminating float and the clearance between the shaft and the bearings. This significantly reduced the vibration and allowed for the turbine to operate at higher rotational speeds.

Measuring torque also proved to be quite difficult due to the high speed of the turbine and the low torque produced. In this research an absorption brake and lever arm configuration were used. This configuration was unstable at the speeds and torques measured. Consideration should be given to making use of alternative dynamometer technologies such as hydraulic and electrical dynamometers with the inclusion of a speed reduction gearbox between the turbine and dynamometer.

6.3 Conclusion and Future Work

6.3.1 Conclusion

To produce small turbines for Organic Rankine Cycles is an expensive process. AM provides an avenue that can be used to reduce the cost as well as simplify the manufacturing and assembly process. The turbines used in this experiment held up well to the operational conditions of the experimental testing phase.

However, the design point power generation was not achieved during the experiments due to experimental, as well as some manufacturing limitations. The turbines operated adequately in the conditions that were provided. Some conclusions can be drawn from the results due to the results captured being repeatable.

Due to the specific speeds of the turbines being very low in the experiments, ranging between 0.003 and 0.051, with an average of 0.22 and standard deviation of 0.05. It is clear that the turbines did not operate in ideal conditions, especially for radial in-flow turbines. The ideal operating range for these turbines is generally considered to be between 0.25 and 2.3. It can also be seen that as the mass flow rate increased, which also meant that the pressure ratio increased, that the specific speed increased significantly. Thus, the turbines should be able to achieve much better efficiencies as the pressure ratios are increased. Ideal flow rates and pressure ratios would be much simpler to achieve in a closed loop system.

The performance of the rotors and guide vanes behaved as expected theoretically. As the inlet and exit angles came closer to the ideal as the performance of the turbine setup improved. At the lower flow rates and pressure ratios values used in the experiment, it is almost impossible to distinguish between the performance of the different configurations. However, when looking at higher flow rate the 25° and 50° rotor exit angle performed slightly better. They also seem to improve more in performance as the pressure ratio was increased.

Similarly, the inlet angle to the rotors also performed similarly at the lower pressure ratios. The 65° angle of guide vane 1 and 2 seems to have performed worse than that of guide vane 3 with a 55° angle. As expected, guide vane 2 performed worse than guide vane 1, due to it having a larger clearance between guide vane and rotor. The goal of quantifying this loss was however not achieved due to the off-design operating conditions.

Another clear conclusion drawn from the experiment is that air, although a usable working fluid to characterize the turbines is not ideal. Even with no heat added to the working fluid for the same pressure ratios organic working fluids such as R134a, R22 and R245fa seems to have much more potential. This would be even more so when the working fluid is at a higher temperature than was tested where the heat and density of these working fluids would be able to transfer significantly more energy to the turbine.

Further work needs to be done on these turbines to determine the full benefit of using AM to produce small radial flow turbines. A closed loop system for the turbines should produce conditions that can provide much higher-pressure ratios. This would increase the power produced by the turbines significantly, thus providing the conditions in

which the turbines can more accurately be compared with one another, by providing greater separation between more and less ideal designs.

6.3.2 Future Work

This study provided a base for future research on the AM of small turbines. This study can be expanded upon in numerous ways. It would be recommended to take the designs from these experiments and improve on the experimental conditions.

Firstly, the operational flow rates should be increased significantly to reach more optimal conditions. To do this ideally a closed loop experimental system should be used, as this would allow for much higher-pressure ratios to be achieved. Furthermore, the turbine and shaft would have to be balanced in order to reduce vibrations at the higher rotational velocities.

Secondly further studies can be done on optimizing the profiles of the turbines and improving on the basic design of these turbines. This can be done by testing more inlet and outlet angles and polishing the surface of the rotors to determine the effect of surface roughness on the turbines. One would expect the rougher surface from the AM process to produce more rotor blade frictional losses than is the case for a polished rotor blade vane. Another design optimization could be to add a diffuser at the turbine exit to improve turbine efficiency.

Thirdly organic fluids can be used in similar experiments, which should prove beneficial in reducing the rotational velocity and increasing the torque produced. Ideally these fluids should also be heated to various temperatures to determine the ideal working fluid at various applicable temperatures. However, the increase in density of these fluids alone should already prove an increase in performance of these turbines.

All of these suggestions should provide operational conditions closer to the ideal and provide much more efficient turbines. That information can then be used to more accurately characterize and optimize the turbines for use in similar systems.

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Appendix A

Turbine design calculations in EES

Inputs

```

.....
alpha_2 = 65 [deg] Nozzle inlet angle
r_2 = 0,02 [m] Tip radius of impeller blade
r_3s = 0,0199 [m] Shoud radius of impeller
r_3h = 0,007 [m] Hub radius of impeller
P_1 = 650000 [Pa] Inlet pressure
P_2 = 600000 [Pa] Turbine inlet pressure
P_3 = 87000 [Pa] Turbin exit pressure
T_1 = 510 [K] Inlet temperature
T_2 = 500 [K] Turbine inlet temperature
Z = 10 Number of impeller blades
m_dot = 0,075 [kg/s] Mass flow rate of working fluid
D_1 = 0,00635 [m] Inlet pipe diameter
A_2 = 0,00022 [m^2] Tubine int area
phi = 0,25 Opimal velocity ratio

```

Calculations

Dimension calculations

```

.....
A_1 = pi/4 * D_1^2 Inlet pipe diameter
beta_2 = (90 [deg] - alpha_2) * 2 Blade inlet angle
beta_3 = arccos(c_3m/omega_3) Blade exit angle
r_3avg = (r_3s + r_3h)/2 Average radius of impeller exit

```

Velocity calculations

```

.....
omega_2 = m_dot/(A_2 * rho_2) Relative inlet velocity
U_2 = omega_2 * tan(alpha_2) Blade tip velocity
c_2 = sqrt(U_2^2 + omega_2^2) Inlet velocity
c_1 = m_dot/(rho_1 * A_1) Volicity of working fluid in pipe
c_3m = phi * U_2 Working fluid velocity at turbine exit
U_3 = omega * r_3avg Average exit velocity of working fluid
omega_3 = sqrt(U_3^2 + c_3m^2) Relative exit velocity of working fluid
Omega = U_2/r_2 Rotaional speed of turbine
N = Omega * 60/(2 * pi) Rotaional speed of turbine
c_0 = sqrt(0,5 * (h_01 - h_03s)) Sprouting velocity

```

Density calculations

$$\rho_1 = \rho(\text{Air}_{\text{ha}}; T = T_1; P = P_1) \quad \text{Density of working fluid in pipe}$$

$$\rho_2 = \rho(\text{Air}_{\text{ha}}; T = T_2; P = P_2) \quad \text{Density of working fluid at turbine inlet}$$

$$\rho_3 = \rho(\text{Air}_{\text{ha}}; T = T_3; P = P_3) \quad \text{Density of working fluid at turbine outlet}$$

Temperature and pressure calculations

$$T_3 = T(\text{Air}_{\text{ha}}; P = P_3; h = h_3) \quad \text{Temperature of working fluid at turbine exit}$$

$$P_{03} = P(\text{Air}_{\text{ha}}; h = h_{03}; s = s_{03})$$

Enthalpy and entropy calculations

$$h_2 = h(\text{Air}_{\text{ha}}; T = T_2; P = P_2) \quad \text{Turbine inlet enthalpy}$$

$$h_{02} = h_2 + 0.5 \cdot c_2^2 \quad \text{Turbine inlet stagnation enthalpy}$$

$$h_{01} = h_{02} \quad \text{Inlet stagnation enthalpy}$$

$$h_1 = h_{01} - 0.5 \cdot c_1^2 \quad \text{Inlet enthalpy}$$

$$h_{02,\text{rel}} = h_2 + 0.5 \cdot \omega_2^2 \quad \text{Turbine relative inlet enthalpy}$$

$$h_{03} = h_{02,\text{rel}} - 0.5 \cdot U_2^2 \quad \text{Exit stagnation enthalpy of working fluid}$$

$$h_3 = h_{03} - 0.5 \cdot c_{3m}^2 \quad \text{Exit enthalpy of working fluid}$$

$$s_1 = s(\text{Air}_{\text{ha}}; T = T_1; P = P_1) \quad \text{Inlet entropy}$$

$$s_{2s} = s_1 \quad \text{Isentropic entropy at turbine inlet}$$

$$s_{3ss} = s_1 \quad \text{Isentropic entropy at turbine exit}$$

$$s_2 = s(\text{Air}_{\text{ha}}; T = T_2; P = P_2) \quad \text{Entropy at turbine inlet}$$

$$s_{3s} = s_2 \quad \text{Isentropic entropy at turbine exit}$$

$$s_3 = s(\text{Air}_{\text{ha}}; T = T_3; P = P_3) \quad \text{Entropy at turbine exit}$$

$$s_{03} = s_3 \quad \text{Stagnation entropy at turbine exit}$$

$$h_{2s} = h(\text{Air}_{\text{ha}}; s = s_{2s}; P = P_2) \quad \text{Isentropic enthalpy at turbine inlet}$$

$$h_{3ss} = h(\text{Air}_{\text{ha}}; s = s_{3ss}; P = P_3) \quad \text{Isentropic enthalpy at turbine exit}$$

$$h_{3s} = h(\text{Air}_{\text{ha}}; s = s_{3s}; P = P_3) \quad \text{Isentropic enthalpy at turbine exit}$$

$$h_{03ss} = h(\text{Air}_{\text{ha}}; s = s_{3ss}; P = P_{03}) \quad \text{Stagnation enthalpy at turbine exit}$$

Power and efficiency calculations

$$\eta_{ts} = \frac{h_{01} - h_{03}}{h_{01} - h_{3ss}} \quad \text{Total to static efficiency}$$

Appendix B

Uncertainty calculations

Variable	Units	Min Range	Max Range	Instrument Description	Source of Uncertainty	± Value	Probability Distribution	Divisor	Standard Uncertainty	(u) ²
Inlet Temperature	°C	0	200	KNE PT100 RTD	Picolog accuracy	1	Normal	2	0.50	0.25
					PT100 RTD	1.3	Normal	2	0.65	0.42
					Resistor circuit	0.9	Normal	2	0.45	0.20
					Picolog 12 bits resolution	0.006103	Rectangular	1.732	0.00	0.00
					Combined standard uncertainty (68% confidence level)		Assumed normal	n/a	0.94	Expanded uncertainty % of FS
					Expanded uncertainty (95% confidence level)		Assumed normal (k=2)	n/a	1.87	°C
Inlet Pressure	kPa	0	1000	Gems 3100 Series Pressure transducer Connected to PicoLog 1216 DAQ	Picolog accuracy	5	Normal	2	2.50	6.25
					GEMS pressure error	10	Normal	2	5.00	25.00
					Resistor circuit	0.9	Normal	2	0.45	0.20
					Picolog 12 bits resolution	0.006103	Rectangular	1.732	0.00	0.00
					GEMS accuracy	2.5	Normal	2	1.25	1.56
					Combined standard uncertainty (68% confidence level)		Assumed normal	n/a	5.75	Expanded uncertainty % of FS
Discharge Temperature	°C	0	200	KNE PT100 RTD	Picolog accuracy	1	Normal	2	0.50	0.25
					PT100 RTD	1.3	Normal	2	0.65	0.42
					Resistor circuit	0.9	Normal	2	0.45	0.20
					Picolog 12 bits resolution	0.006103	Rectangular	1.732	0.00	0.00
					Combined standard uncertainty (68% confidence level)		Assumed normal	n/a	0.94	Expanded uncertainty % of FS
					Expanded uncertainty (95% confidence level)		Assumed normal (k=2)	n/a	1.87	°C
Discharge Pressure	kPa	0	1000	Gems 3100 Series Pressure transducer Connected to PicoLog 1216 DAQ	Picolog accuracy	5	Normal	2	2.50	6.25
					GEMS pressure error	10	Normal	2	5.00	25.00
					Resistor circuit	0.9	Normal	2	0.45	0.20
					Picolog 12 bits resolution	0.006103	Rectangular	1.732	0.00	0.00
					GEMS accuracy	2.5	Normal	2	1.25	1.56
					Combined standard uncertainty (68% confidence level)		Assumed normal	n/a	5.75	Expanded uncertainty % of FS
Force Applied on Brake by Turbine	kg	0.3	3	FC22 Compression Load Cell Connected to PicoLog 1216 DAQ	Picolog accuracy	0.015	Normal	2	0.01	0.00
					FC22 load cell	0.0354	Normal	2	0.02	0.00
					Picolog 12 bits resolution	0.006103	Rectangular	1.732	0.00	0.00
					Combined standard uncertainty (68% confidence level)		Assumed normal	n/a	0.020	Expanded uncertainty % of FS
					Expanded uncertainty (95% confidence level)		Assumed normal (k=2)	n/a	0.039	kg
Speed of Turbine	RPM	0	99,999	Laser photo tachometer	Tachometer	49.9995	Normal	2	25.00	624.99
					Combined standard uncertainty (68% confidence level)		Assumed normal	n/a	25.00	Expanded uncertainty % of FS
					Expanded uncertainty (95% confidence level)		Assumed normal (k=2)	n/a	50.00	RPM
					Techfluid flowmeter	0.288	Normal	2	0.14	0.02
Volume of Air passing through the turbine	Nm ³ /h	1.8	18	Techfluid series C313-06301	Combined standard uncertainty (68% confidence level)		Assumed normal	n/a	0.14	Expanded uncertainty % of FS
					Expanded uncertainty (95% confidence level)		Assumed normal (k=2)	n/a	0.29	Nm ³ /h

Appendix C

Summarized recorded data

[illegible]

[illegible]

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Appendix D

Turbine characterization calculations

Theoretical Calculations of Experimental Results

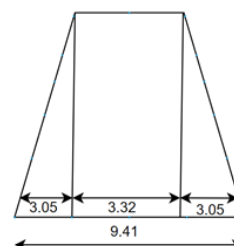
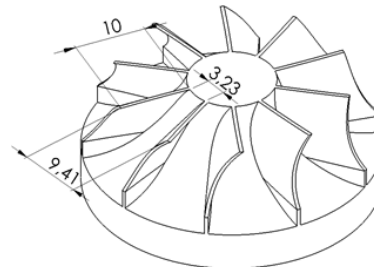
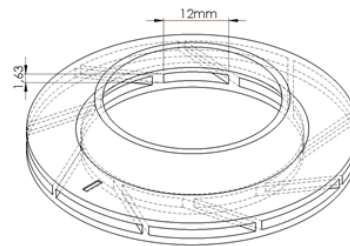
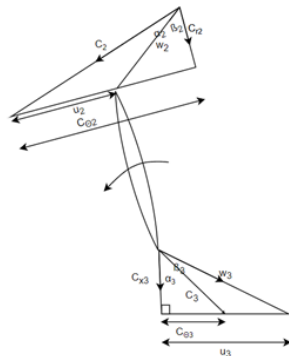
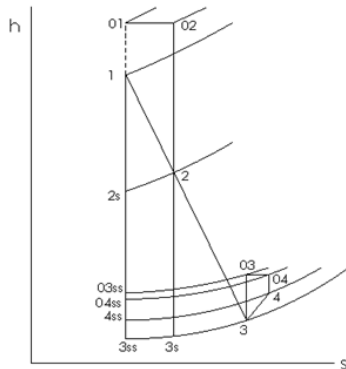
```

 $\alpha_2 := 65 \text{ deg}$ 
 $\beta_3 := 0 \text{ deg}$ 
 $t_2 := 19 \text{ }^\circ\text{C}$ 
 $t_3 := 19 \text{ }^\circ\text{C}$ 
 $p_2 := 5966,12 \text{ Pa}$ 
 $p_3 := 2074,12 \text{ Pa}$ 
 $p_{atm} := 86550 \text{ Pa}$ 
 $m_{dotflow} := 40 \frac{\text{kg}}{\text{hr}}$ 
 $n := 1763 \frac{\text{rev}}{\text{min}}$ 
 $m_{stat} := 353,428 \text{ g}$ 
 $m_{operational} := 373,553 \text{ g}$ 
 $m_{preload} := 400 \text{ g}$ 
 $m_{fric} := 100 \text{ g}$ 
 $m_{net} := 20125 \text{ g}$ 
 $\gamma := 1,4$ 
 $d_2 := 13,9 \text{ mm}$ 
 $d_3 := 33,55 \text{ mm}$ 

 $r_{brake} := 0,08 \text{ m}$ 

 $M_{air} := 0,0289654 \frac{\text{kg}}{\text{mol}}$ 
 $T_2 := t_2$ 
 $T_3 := t_3$ 
 $p_{2abs} := p_2 + p_{atm}$ 
 $p_{3abs} := p_3 + p_{atm}$ 
 $m_{dot} := m_{dotflow}$ 
 $m_{dot} = 0,0111 \frac{\text{kg}}{\text{s}}$ 
 $T_2 = 292,15 \text{ K}$ 
 $T_3 = 292,15 \text{ K}$ 
 $p_{2abs} = 92516,12 \text{ Pa}$ 
 $p_{3abs} = 88624,12 \text{ Pa}$ 
 $R_o := 8,3145 \cdot \frac{\text{J}}{(\text{mol K})}$ 
 $R := 287 \frac{\text{J}}{\text{kg K}}$ 
 $r_2 := 20 \text{ mm}$ 
 $r_3 := 8 \text{ mm}$ 

```



The first step we assume nominal conditions and determine what the turbine velocity and power should be with the measured temperature, pressure and flow rate values.

$$A_2 := 12 \text{ mm} \cdot 1,63 \text{ mm} \cdot 9$$

$$A_2 = 0,0002 \text{ m}^2$$

$$A_3 := 10 \cdot \left((3,32 \text{ mm} \cdot 10 \text{ mm}) + \left(2 \cdot \left(\frac{1}{3} \cdot 10 \text{ mm} \cdot 3,05 \text{ mm} \right) \right) \right)$$

$$A_3 = 0,0005 \text{ m}^2$$

$$\rho_{air2} := \frac{(p_{2abs} \cdot M_{air})}{(T_2 \cdot R_o)}$$

$$\rho_{air2} = 1,1032 \frac{\text{kg}}{\text{m}^3}$$

$$Q_2 := \frac{\dot{m}}{\rho_{air2}}$$

$$Q_2 = 0,0101 \frac{\text{m}^3}{\text{s}}$$

$$c_2 := \frac{Q_2}{A_2}$$

$$c_2 = 57,2125 \frac{\text{m}}{\text{s}}$$

$$c_{r2} := c_2 \cdot \cos(\alpha_2)$$

$$c_{r2} = 24,1791 \frac{\text{m}}{\text{s}}$$

$$\rho_{air3} := \frac{(p_{3abs} \cdot M_{air})}{(T_3 \cdot R_o)}$$

$$\rho_{air3} = 1,0568 \frac{\text{kg}}{\text{m}^3}$$

$$Q_3 := \frac{\dot{m}}{\rho_{air3}}$$

$$Q_3 = 0,0105 \frac{\text{m}^3}{\text{s}}$$

$$c_{x3} := \frac{Q_3}{A_3}$$

$$c_{x3} = 19,6401 \frac{\text{m}}{\text{s}}$$

Below are the actual measured velocities that was measured experimentally and then calculated using the radius

$$u_{2act} := r_2 \cdot n$$

$$u_{2act} = 3,6924 \frac{\text{m}}{\text{s}}$$

$$u_{3act} := r_3 \cdot n$$

$$u_{3act} = 1,477 \frac{\text{m}}{\text{s}}$$

The experimental blade velocity can then be calculated and compared to the experimental value, the entry and exit angles are known. This is assuming that we are working with nominal conditions

$$u_{2theor} := c_2 \cdot \sin(\alpha_2)$$

$$u_{2theor} = 51,8522 \frac{\text{m}}{\text{s}}$$

$$n_{theoretical} := \frac{u_{2theor}}{r_2}$$

$$n_{theoretical} = 2592,6085 \text{ Hz}$$

$$n_{theoretical} = 24757,5871 \text{ rpm}$$

Therefore u_{3theor} will rotate at the same rotational velocity

$$u_{3theor} := n_{theoretical} \cdot r_3$$

$$u_{3theor} = 20,7409 \frac{\text{m}}{\text{s}}$$



$$c_3 := \sqrt{(c_{x3})^2 + (u_{3_theor})^2}$$

$$c_3 = 28,5642652 \frac{\text{m}}{\text{s}}$$

$$a_2 := \sqrt{\gamma \cdot R \cdot T_2}$$

$$a_2 = 342,6162 \frac{\text{m}}{\text{s}}$$

$$Mach_2 := \frac{c_2}{a_2}$$

$$Mach_2 = 0,167$$

$$Mach_3 := \frac{c_3}{a_3}$$

$$Mach_3 = 0,0834$$

$$T_{o2} := \left(1 + \left(\left(\frac{\gamma - 1}{2} \right) \cdot (Mach_2)^2 \right) \right) \cdot T_2$$

$$T_{o2} = 293,7793 \text{ K}$$

$$T_{o3} := \left(1 + \left(\left(\frac{\gamma - 1}{2} \right) \cdot (Mach_3)^2 \right) \right) \cdot T_3$$

$$T_{o3} = 292,5561 \text{ K}$$

$$p_{o2} := p_{abs} \cdot \left(1 + \left(\left(\frac{\gamma - 1}{2} \right) \cdot (Mach_2)^2 \right) \right)^{\left(\frac{\gamma}{\gamma - 1} \right)}$$

$$p_{o2} = 94334,5957 \text{ Pa}$$

$$p_{o3} := p_{abs} \cdot \left(1 + \left(\left(\frac{\gamma - 1}{2} \right) \cdot (Mach_3)^2 \right) \right)^{\left(\frac{\gamma}{\gamma - 1} \right)}$$

$$p_{o3} = 89056,0713 \text{ Pa}$$

$h_{o1}: h_{o1} = h_{o2}, h_{o2}$ can be calculated using the available information using Engineering equation solver

h_{o3} : Can be calculated using the available information using Engineering equation solver

$h_{o3ss}: s_{3ss} = s_{o3ss} = s_1$, @ pressure $o3$ Calculated using engineering equation solver

$$h_{o2} := 294,1 \frac{\text{kJ}}{\text{kg}} \quad s_{o2} := 6,883 \frac{\text{kJ}}{\text{kg K}} \quad h_{o1} := h_{o2} \quad s_{o1} := 6,878 \frac{\text{kJ}}{\text{kg K}}$$

$$h_{o3} := 292,6 \frac{\text{kJ}}{\text{kg}} \quad s_{o3} := 6,878 \frac{\text{kJ}}{\text{kg K}}$$

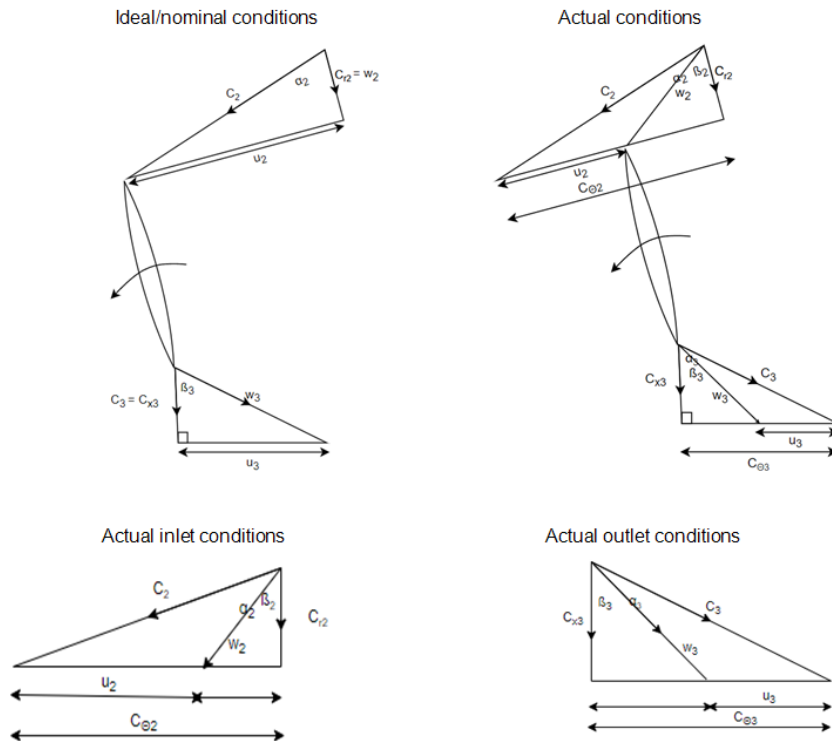
$$h_{o3ss} := 292,3 \frac{\text{kJ}}{\text{kg}}$$

$$\eta_{ts} := \frac{(h_{o1} - h_{o3})}{(h_{o1} - h_{o3ss})}$$

$$\eta_{ts} = 83,3333 \%$$

It can clearly be seen that the theoretical velocities are significantly larger than the actual velocities, therefore we are not operating under nominal conditions. This is due to running at relatively low differential pressure than would be ideal. There will also be losses in the system due to slip, clearances and surface roughness to name a few.

The results of the theoretical calculation can then be used to determine the actual velocity diagrams that can then be compared to the theoretical diagram.



$$c_{\theta 3} := u_{3_theor} - u_{3_act}$$

$$c_{\theta 3} = 19,2639 \frac{\text{m}}{\text{s}}$$

$$c_{3_act} := \sqrt{(c_{x3})^2 + (u_{3_act})^2}$$

$$c_{3_act} = 19,6956 \frac{\text{m}}{\text{s}}$$

$$mach_{3_act} := \frac{c_{3_act}}{a_3}$$

$$T_{o3} := \left(1 + \left(\left(\frac{\gamma - 1}{2} \right) \cdot (mach_{3_act})^2 \right) \right) \cdot T_3$$

$$T_{o3} = 292,3431 \text{ K}$$

$$p_{o3} := p_{3abs} \cdot \left(\left(1 + \left(\left(\frac{\gamma - 1}{2} \right) \cdot (mach_{3_act})^2 \right) \right) \left(\frac{\gamma}{\gamma - 1} \right) \right)$$

$$p_{o3} = 88829,2974 \text{ Pa}$$

$$h_{o3_act} := 292,8 \frac{\text{kJ}}{\text{kg}}$$

$$\eta_{ts_act} := \frac{(h_{o1} - h_{o3_act})}{(h_{o1} - h_{o3ss})}$$

$$\eta_{ts_act} = 0,7222$$

$$f_{net} := m_{net} g_e$$

$$f_{net} = 197,3588 \text{ N}$$

$$\tau_{act} := r_{brake} \cdot f_{net}$$

$$\tau_{act} = 15,7887 \text{ Nm}$$

$$P := \tau_{act} \cdot n$$

$$P = 2914,9257 \text{ W}$$

$$P_{theo} := \tau_{act} \cdot \eta_{theoretical}$$

$$P_{theo} = 40933,934 \text{ W}$$

$$P_{ideal} := \frac{P_{theo}}{\eta_{ts}}$$

$$P_{ideal} = 49120,7208 \text{ W}$$

Using the power out and rotational velocities the rest of the actual inlet and outlet conditions can be calculated.

We can use the power in the workfluid at the entry conditions and the output power measure of the turbine to determine the efficiency of the turbine. The difference between the power in the fluid at the inlet conditions and the power measured on the turbine at the outlet conditions can be considered as the losses. From the theoretical calculations and the theoretical work determined from the turbine the following is expected.

Calculating the efficiency using the actual vs theoretical speed of the turbine:

$$\eta_{speed} := \frac{u_{3_act}}{u_{3_theor}}$$

$$\eta_{speed} = 7,121 \%$$

Due to the low operating pressures this values makes sense. Using the power out we should be able to determine a similar value to confirm if this is in fact the case. Therefore the expected turbine result of this equation should be roughly in the region of 7 – 12% (Assumption made to be checked in the following calculations).

Power efficiencies:

Ideal vs theoretical:

$$P_{it} := \frac{P_{theo}}{P_{ideal}}$$

$$P_{it} = 83,3333 \%$$

Theoretical vs actual:

$$P_{ta} := \frac{P}{P_{theo}}$$

$$P_{ta} = 7,121 \%$$

Ideal vs actual:

$$P_{ia} := \frac{P}{P_{ideal}}$$

$$P_{ia} = 5,9342 \%$$

Using the same conditions as the theoretical calculation we can work back what the inlet

We also know that:

For $h_{01} = h_{02}$:

$$\Delta W = h_{02} - h_{03} = h_2 - h_3 + \frac{1}{2}(c_2^2 - c_3^2) = \frac{1}{2}[(U_2^2 - U_3^2) - (w_2^2 - w_3^2) + (c_2^2 - c_3^2)]$$

Therefore

$$A := u_{2_act}^2 - u_{3_act}^2$$

$$A = 0,0115 \frac{\text{kJ}}{\text{kg}}$$

$$w_{2t} := \sqrt{c_{2t}^2 - u_{2_act}^2}$$

$$w_{2t} = 57,0933 \frac{\text{m}}{\text{s}}$$

$$w_{3t} := \sqrt{c_{3t}^2 + u_{3_act}^2}$$

$$w_{3t} = 28,6024 \frac{\text{m}}{\text{s}}$$

$$B := w_{2t}^2 - w_{3t}^2$$

$$B = 2,4415 \frac{\text{kJ}}{\text{kg}}$$

$$C := c_{2t}^2 - c_{3_act}^2$$

$$C = 2,8854 \frac{\text{kJ}}{\text{kg}}$$

$$\Delta W := \frac{1}{2} \cdot (A - B + C)$$

$$\Delta W = 0,2276 \frac{\text{kJ}}{\text{kg}}$$

$$\Delta \dot{W}_{dot} := \Delta W \cdot \dot{m}_{dotflow}$$

$$\Delta \dot{W}_{dot} = 2,5293 \text{ W}$$

$$At := u_{2_theor}^2 - u_{3_theor}^2$$

$$At = 2,2585 \frac{\text{kJ}}{\text{kg}}$$

$$w_{2t} := \sqrt{c_{2t}^2 - u_{2_theor}^2}$$

$$w_{2t} = 24,1791 \frac{\text{m}}{\text{s}}$$

$$w_{3t} := \sqrt{c_{3t}^2 + u_{3_theor}^2}$$

$$w_{3t} = 35,3002 \frac{\text{m}}{\text{s}}$$

$$Bt := w_{2t}^2 - w_{3t}^2$$

$$Bt = -0,6615 \frac{\text{kJ}}{\text{kg}}$$

$$Ct := c_{2t}^2 - c_{3t}^2$$

$$Ct = 2,4574 \frac{\text{kJ}}{\text{kg}}$$

$$\Delta Wt := \frac{1}{2} \cdot (At - Bt + Ct)$$

$$\Delta Wt = 2,6886 \frac{\text{kJ}}{\text{kg}}$$

The definition of total to static (enthalpy) efficiency is presented in Equation 1-18

$$\eta_{ts} = \frac{\text{Shaft power output of an actual turbine}}{\text{(Shaft power output + Power wasted in exhaust kinetic energy) of an ideal turbine}} \quad (1-18)$$

Actual $\Delta W = 0.2276 \text{ kJ/kg}$

Theoretical $\Delta W = 2.9026$

$$\eta_{ts_act_calculated} := \frac{\Delta W}{\Delta W_t}$$

$$\eta_{ts_act_calculated} = 8,4665 \%$$

Dimensional analysis

$$\frac{p_{o2}}{p_{o3}} = 1,062 \quad \text{Pressure ratio}$$

$$\phi := \frac{Q_3}{n \cdot (r_2 \cdot 2)^3} \quad \text{Flow coefficient}$$

$$\phi = 0,8898$$

$$H_2 := \frac{p_{2abs}}{\rho_{air2} g_e} \quad \text{Pressure head of air}$$

$$H_2 = 8551,4906 \text{ m}$$

$$\psi := \frac{(g_e \cdot H_2)}{n^2 \cdot (r_2 \cdot 2)^2} \quad \text{Head coefficient}$$

$$\psi = 1537,732$$

$$\Omega_s := \frac{\phi^{\frac{1}{2}}}{\frac{\psi^{\frac{3}{4}}}{3}} \quad \text{Specific speed}$$

$$\Omega_s = 0,0038$$

+

Appendix E

Entropy and enthalpy values from EES

$$T_1 = 19,19 \text{ [C]}$$

$$P_1 = 88,83 \text{ [kPa]}$$

$$S_1 = 6,878 \text{ [kJ/kg*K]}$$

$$H_1 = 294,1 \text{ [kJ/kg]}$$

$$H_1 = h \left(\text{Air}_{ha} ; T = T_1 ; P = P_1 \right)$$

$$S_1 = s \left(\text{Air}_{ha} ; T = T_1 ; P = P_1 \right)$$

$$S_1 = s \left(\text{Air}_{ha} ; P = P_1 ; h = H_1 \right)$$

$$H_1 = h \left(\text{Air}_{ha} ; P = P_1 ; s = S_1 \right)$$

Appendix F

Dimensional analysis calculations

Dimensional analysis

$$\frac{p_{o2}}{p_{o3}} = 1,062 \quad \text{Pressure ratio}$$

$$\Phi := \frac{Q_3}{n \cdot (r_2 \cdot 2)^3} \quad \text{Flow coefficient}$$

$$\Phi = 0,8898$$

$$H_2 := \frac{p_{2abs}}{\rho_{air2} g_e} \quad \text{Pressure head of air}$$

$$H_2 = 8551,4906 \text{ m}$$

$$\psi := \frac{(g_e \cdot H_2)}{n^2 \cdot (r_2 \cdot 2)^2} \quad \text{Head coefficient}$$

$$\psi = 1537,732$$

$$\Omega_s := \frac{\Phi^{\frac{1}{2}}}{\frac{3}{\psi^{\frac{1}{4}}}} \quad \text{Specific speed}$$

$$\Omega_s = 0,0038$$

Air																			
P	Temperature v (°C)	Temperature in K	M _{gas} (kg/mol)	R _o (J/moleK)	Head Coef	Specific diameter	STD Dev	g	P	Pressure head	Increase pressure	Increased Pressure head	r	Dia	Dia ²	n	Total efficiency actual	Total efficiency theoretical	Q (m ³ /s)
114896	16.209	289.359	0.029	8.315	20.876	15.250	2.579	9.810	1.383	8466.904	1.000	8466.904	0.020	0.040	0.002	1576.9	0.016	0.068	15.516
150000	16.209	289.359	0.029	8.315	20.876	15.250	2.579	9.810	1.806	8466.904	1.250	10583.629	0.020	0.040	0.002	1763.1	0.016	0.068	22.647
200000	16.209	289.359	0.029	8.315	20.876	15.250	2.579	9.810	2.408	8466.904	1.500	12700.355	0.020	0.040	0.002	1931.4	0.016	0.068	33.078
250000	16.209	289.359	0.029	8.315	20.876	15.250	2.579	9.810	3.010	8466.904	1.750	14817.081	0.020	0.040	0.002	2086.1	0.016	0.068	44.661
300000	16.209	289.359	0.029	8.315	20.876	15.250	2.579	9.810	3.612	8466.904	2.000	16933.807	0.020	0.040	0.002	2230.1	0.016	0.068	57.293
350000	16.209	289.359	0.029	8.315	20.876	15.250	2.579	9.810	4.214	8466.904	2.250	19050.533	0.020	0.040	0.002	2365.4	0.016	0.068	70.897
400000	16.209	289.359	0.029	8.315	20.876	15.250	2.579	9.810	4.816	8466.904	2.500	21167.259	0.020	0.040	0.002	2493.4	0.016	0.068	85.408
450000	16.209	289.359	0.029	8.315	20.876	15.250	2.579	9.810	5.418	8466.904	2.750	23283.985	0.020	0.040	0.002	2615.1	0.016	0.068	100.773
500000	16.209	289.359	0.029	8.315	20.876	15.250	2.579	9.810	6.020	8466.904	3.000	25400.711	0.020	0.040	0.002	2731.4	0.016	0.068	116.949
550000	16.209	289.359	0.029	8.315	20.876	15.250	2.579	9.810	6.622	8466.904	3.250	27517.437	0.020	0.040	0.002	2842.9	0.016	0.068	133.897
600000	16.209	289.359	0.029	8.315	20.876	15.250	2.579	9.810	7.224	8466.904	3.500	29634.163	0.020	0.040	0.002	2950.2	0.016	0.068	151.584
650000	16.209	289.359	0.029	8.315	20.876	15.250	2.579	9.810	7.826	8466.904	3.750	31750.888	0.020	0.040	0.002	3053.7	0.016	0.068	169.979
700000	16.209	289.359	0.029	8.315	20.876	15.250	2.579	9.810	8.428	8466.904	4.000	33867.614	0.020	0.040	0.002	3153.9	0.016	0.068	189.058
750000	16.209	289.359	0.029	8.315	20.876	15.250	2.579	9.810	9.030	8466.904	4.250	35984.340	0.020	0.040	0.002	3251.0	0.016	0.068	208.796
800000	16.209	289.359	0.029	8.315	20.876	15.250	2.579	9.810	9.632	8466.904	4.500	38101.066	0.020	0.040	0.002	3345.2	0.016	0.068	229.173
R134a																			
Pressure in	Temperature in C	Temperature in K	M _{gas} (kg/mol)	R _o (J/moleK)	Head Coef	Specific diameter	STD Dev	g	P	Pressure head	Increase pressure	Increased Pressure head	r	Dia	Dia ²	n	Total efficiency actual	Total efficiency theoretical	Q (m ³ /s)
114896	16.209	289.359	0.102	8.315	20.876	15.250	2.579	9.810	4.873	8466.904	1.000	8466.904	0.020	0.040	0.002	1576.9	0.016	0.068	54.654
150000	16.209	289.359	0.102	8.315	20.876	15.250	2.579	9.810	6.361	8466.904	1.250	10583.629	0.020	0.040	0.002	1763.1	0.016	0.068	79.774
200000	16.209	289.359	0.102	8.315	20.876	15.250	2.579	9.810	8.482	8466.904	1.500	12700.355	0.020	0.040	0.002	1931.4	0.016	0.068	116.518
250000	16.209	289.359	0.102	8.315	20.876	15.250	2.579	9.810	10.602	8466.904	1.750	14817.081	0.020	0.040	0.002	2086.1	0.016	0.068	157.317
300000	16.209	289.359	0.102	8.315	20.876	15.250	2.579	9.810	12.723	8466.904	2.000	16933.807	0.020	0.040	0.002	2230.1	0.016	0.068	201.814
350000	16.209	289.359	0.102	8.315	20.876	15.250	2.579	9.810	14.843	8466.904	2.250	19050.533	0.020	0.040	0.002	2365.4	0.016	0.068	248.732
400000	16.209	289.359	0.102	8.315	20.876	15.250	2.579	9.810	16.963	8466.904	2.500	21167.259	0.020	0.040	0.002	2493.4	0.016	0.068	300.847
450000	16.209	289.359	0.102	8.315	20.876	15.250	2.579	9.810	19.084	8466.904	2.750	23283.985	0.020	0.040	0.002	2615.1	0.016	0.068	354.972
500000	16.209	289.359	0.102	8.315	20.876	15.250	2.579	9.810	21.204	8466.904	3.000	25400.711	0.020	0.040	0.002	2731.4	0.016	0.068	411.952
550000	16.209	289.359	0.102	8.315	20.876	15.250	2.579	9.810	23.325	8466.904	3.250	27517.437	0.020	0.040	0.002	2842.9	0.016	0.068	471.650
600000	16.209	289.359	0.102	8.315	20.876	15.250	2.579	9.810	25.445	8466.904	3.500	29634.163	0.020	0.040	0.002	2950.2	0.016	0.068	533.951
650000	16.209	289.359	0.102	8.315	20.876	15.250	2.579	9.810	27.566	8466.904	3.750	31750.888	0.020	0.040	0.002	3053.7	0.016	0.068	598.749
700000	16.209	289.359	0.102	8.315	20.876	15.250	2.579	9.810	29.686	8466.904	4.000	33867.614	0.020	0.040	0.002	3153.9	0.016	0.068	665.953
750000	16.209	289.359	0.102	8.315	20.876	15.250	2.579	9.810	31.807	8466.904	4.250	35984.340	0.020	0.040	0.002	3251.0	0.016	0.068	735.481
800000	16.209	289.359	0.102	8.315	20.876	15.250	2.579	9.810	33.927	8466.904	4.500	38101.066	0.020	0.040	0.002	3345.2	0.016	0.068	807.257
R22																			
Pressure in	Temperature in C	Temperature in K	M _{gas} (kg/mol)	R _o (J/moleK)	Head Coef	Specific diameter	STD Dev	g	P	Pressure head	Increase pressure	Increased Pressure head	r	Dia	Dia ²	n	Total efficiency actual	Total efficiency theoretical	Q (m ³ /s)
114896	16.209	289.359	0.086	8.315	20.876	15.250	2.579	9.810	4.130	8466.904	1.000	8466.904	0.020	0.040	0.002	1576.9	0.016	0.068	46.319
150000	16.209	289.359	0.086	8.315	20.876	15.250	2.579	9.810	5.331	8466.904	1.250	10583.629	0.020	0.040	0.002	1763.1	0.016	0.068	67.608
200000	16.209	289.359	0.086	8.315	20.876	15.250	2.579	9.810	7.188	8466.904	1.500	12700.355	0.020	0.040	0.002	1931.4	0.016	0.068	98.748
250000	16.209	289.359	0.086	8.315	20.876	15.250	2.579	9.810	8.985	8466.904	1.750	14817.081	0.020	0.040	0.002	2086.1	0.016	0.068	133.325
300000	16.209	289.359	0.086	8.315	20.876	15.250	2.579	9.810	10.782	8466.904	2.000	16933.807	0.020	0.040	0.002	2230.1	0.016	0.068	171.037
350000	16.209	289.359	0.086	8.315	20.876	15.250	2.579	9.810	12.579	8466.904	2.250	19050.533	0.020	0.040	0.002	2365.4	0.016	0.068	211.647
400000	16.209	289.359	0.086	8.315	20.876	15.250	2.579	9.810	14.376	8466.904	2.500	21167.259	0.020	0.040	0.002	2493.4	0.016	0.068	254.967
450000	16.209	289.359	0.086	8.315	20.876	15.250	2.579	9.810	16.174	8466.904	2.750	23283.985	0.020	0.040	0.002	2615.1	0.016	0.068	300.838
500000	16.209	289.359	0.086	8.315	20.876	15.250	2.579	9.810	17.971	8466.904	3.000	25400.711	0.020	0.040	0.002	2731.4	0.016	0.068	349.127
550000	16.209	289.359	0.086	8.315	20.876	15.250	2.579	9.810	19.768	8466.904	3.250	27517.437	0.020	0.040	0.002	2842.9	0.016	0.068	399.722
600000	16.209	289.359	0.086	8.315	20.876	15.250	2.579	9.810	21.565	8466.904	3.500	29634.163	0.020	0.040	0.002	2950.2	0.016	0.068	452.521
650000	16.209	289.359	0.086	8.315	20.876	15.250	2.579	9.810	23.362	8466.904	3.750	31750.888	0.020	0.040	0.002	3053.7	0.016	0.068	507.437
700000	16.209	289.359	0.086	8.315	20.876	15.250	2.579	9.810	25.159	8466.904	4.000	33867.614	0.020	0.040	0.002	3153.9	0.016	0.068	564.393
750000	16.209	289.359	0.086	8.315	20.876	15.250	2.579	9.810	26.956	8466.904	4.250	35984.340	0.020	0.040	0.002	3251.0	0.016	0.068	623.317

R143a																			
Pressure in	Temperature	M _{gas}	R ₀	Head	Specific	STD Dev	p	Pressure	Increase	Increased	r	Dia	Dia ²	n	Total efficiency	Total efficiency	Q (m ³)	P	
in C	In K	(kg/mol)	(J/molk)	Coef	diameter	g		head	pressure	Pressure head					actual	theoretical			
114896	16.209	289.359	0.084	8.315	20.876	15.250	2.579	9.810	4.013	8466.904	1.500	10583.629	0.020	0.040	0.002	1576.9	0.016	0.068	45.017
150000	16.209	289.359	0.084	8.315	20.876	15.250	2.579	9.810	5.240	8466.904	1.250	10583.629	0.020	0.040	0.002	1763.1	0.016	0.068	65.708
200000	16.209	289.359	0.084	8.315	20.876	15.250	2.579	9.810	6.986	8466.904	1.500	12700.355	0.020	0.040	0.002	1931.4	0.016	0.068	95.973
250000	16.209	289.359	0.084	8.315	20.876	15.250	2.579	9.810	8.733	8466.904	1.750	14817.081	0.020	0.040	0.002	2086.1	0.016	0.068	129.578
300000	16.209	289.359	0.084	8.315	20.876	15.250	2.579	9.810	10.479	8466.904	2.000	16933.807	0.020	0.040	0.002	2230.1	0.016	0.068	166.230
350000	16.209	289.359	0.084	8.315	20.876	15.250	2.579	9.810	12.226	8466.904	2.250	19050.533	0.020	0.040	0.002	2365.4	0.016	0.068	205.699
400000	16.209	289.359	0.084	8.315	20.876	15.250	2.579	9.810	13.972	8466.904	2.500	21167.259	0.020	0.040	0.002	2493.4	0.016	0.068	247.801
450000	16.209	289.359	0.084	8.315	20.876	15.250	2.579	9.810	15.719	8466.904	2.750	23283.985	0.020	0.040	0.002	2615.1	0.016	0.068	292.383
500000	16.209	289.359	0.084	8.315	20.876	15.250	2.579	9.810	17.466	8466.904	3.000	25400.711	0.020	0.040	0.002	2731.4	0.016	0.068	339.316
550000	16.209	289.359	0.084	8.315	20.876	15.250	2.579	9.810	19.212	8466.904	3.250	27517.437	0.020	0.040	0.002	2842.9	0.016	0.068	388.469
600000	16.209	289.359	0.084	8.315	20.876	15.250	2.579	9.810	20.959	8466.904	3.500	29634.163	0.020	0.040	0.002	2950.2	0.016	0.068	439.804
650000	16.209	289.359	0.084	8.315	20.876	15.250	2.579	9.810	22.705	8466.904	3.750	31750.888	0.020	0.040	0.002	3053.7	0.016	0.068	493.177
700000	16.209	289.359	0.084	8.315	20.876	15.250	2.579	9.810	24.452	8466.904	4.000	33867.614	0.020	0.040	0.002	3153.9	0.016	0.068	548.532
750000	16.209	289.359	0.084	8.315	20.876	15.250	2.579	9.810	26.198	8466.904	4.250	35984.340	0.020	0.040	0.002	3251.0	0.016	0.068	605.800
800000	16.209	289.359	0.084	8.315	20.876	15.250	2.579	9.810	27.945	8466.904	4.500	38101.066	0.020	0.040	0.002	3345.2	0.016	0.068	664.921
R124																			
Pressure in	Temperature	M _{gas}	R ₀	Head	Specific	STD Dev	p	Pressure	Increase	Increased	r	Dia	Dia ²	n	Total efficiency	Total efficiency	Q (m ³)	P	
in C	In K	(kg/mol)	(J/molk)	Coef	diameter	g		head	pressure	Pressure head					actual	theoretical			
114896	16.209	289.359	0.136	8.315	20.876	15.250	2.579	9.810	6.518	8466.904	1.000	8466.904	0.020	0.040	0.002	1576.9	0.016	0.068	73.105
150000	16.209	289.359	0.136	8.315	20.876	15.250	2.579	9.810	8.509	8466.904	1.250	10583.629	0.020	0.040	0.002	1763.1	0.016	0.068	106.706
200000	16.209	289.359	0.136	8.315	20.876	15.250	2.579	9.810	11.345	8466.904	1.500	12700.355	0.020	0.040	0.002	1931.4	0.016	0.068	155.855
250000	16.209	289.359	0.136	8.315	20.876	15.250	2.579	9.810	14.182	8466.904	1.750	14817.081	0.020	0.040	0.002	2086.1	0.016	0.068	210.428
300000	16.209	289.359	0.136	8.315	20.876	15.250	2.579	9.810	17.018	8466.904	2.000	16933.807	0.020	0.040	0.002	2230.1	0.016	0.068	269.948
350000	16.209	289.359	0.136	8.315	20.876	15.250	2.579	9.810	19.854	8466.904	2.250	19050.533	0.020	0.040	0.002	2365.4	0.016	0.068	334.044
400000	16.209	289.359	0.136	8.315	20.876	15.250	2.579	9.810	22.690	8466.904	2.500	21167.259	0.020	0.040	0.002	2493.4	0.016	0.068	402.415
450000	16.209	289.359	0.136	8.315	20.876	15.250	2.579	9.810	25.527	8466.904	2.750	23283.985	0.020	0.040	0.002	2615.1	0.016	0.068	474.813
500000	16.209	289.359	0.136	8.315	20.876	15.250	2.579	9.810	28.363	8466.904	3.000	25400.711	0.020	0.040	0.002	2731.4	0.016	0.068	551.029
550000	16.209	289.359	0.136	8.315	20.876	15.250	2.579	9.810	31.199	8466.904	3.250	27517.437	0.020	0.040	0.002	2842.9	0.016	0.068	630.883
600000	16.209	289.359	0.136	8.315	20.876	15.250	2.579	9.810	34.036	8466.904	3.500	29634.163	0.020	0.040	0.002	2950.2	0.016	0.068	714.216
650000	16.209	289.359	0.136	8.315	20.876	15.250	2.579	9.810	36.872	8466.904	3.750	31750.888	0.020	0.040	0.002	3053.7	0.016	0.068	800.890
700000	16.209	289.359	0.136	8.315	20.876	15.250	2.579	9.810	39.708	8466.904	4.000	33867.614	0.020	0.040	0.002	3153.9	0.016	0.068	890.783
750000	16.209	289.359	0.136	8.315	20.876	15.250	2.579	9.810	42.545	8466.904	4.250	35984.340	0.020	0.040	0.002	3251.0	0.016	0.068	983.784
800000	16.209	289.359	0.136	8.315	20.876	15.250	2.579	9.810	45.381	8466.904	4.500	38101.066	0.020	0.040	0.002	3345.2	0.016	0.068	1079.793
R245fa																			
Pressure in	Temperature	M _{gas}	R ₀	Head	Specific	STD Dev	p	Pressure	Increase	Increased	r	Dia	Dia ²	n	Total efficiency	Total efficiency	Q (m ³)	P	
in C	In K	(kg/mol)	(J/molk)	Coef	diameter	g		head	pressure	Pressure head					actual	theoretical			
114896	16.209	289.359	0.134	8.315	20.876	15.250	2.579	9.810	6.402	8466.904	1.000	8466.904	0.020	0.040	0.002	1576.9	0.016	0.068	71.806
150000	16.209	289.359	0.134	8.315	20.876	15.250	2.579	9.810	8.338	8466.904	1.250	10583.629	0.020	0.040	0.002	1763.1	0.016	0.068	104.810
200000	16.209	289.359	0.134	8.315	20.876	15.250	2.579	9.810	11.144	8466.904	1.500	12700.355	0.020	0.040	0.002	1931.4	0.016	0.068	153.084
250000	16.209	289.359	0.134	8.315	20.876	15.250	2.579	9.810	13.929	8466.904	1.750	14817.081	0.020	0.040	0.002	2086.1	0.016	0.068	206.687
300000	16.209	289.359	0.134	8.315	20.876	15.250	2.579	9.810	16.715	8466.904	2.000	16933.807	0.020	0.040	0.002	2230.1	0.016	0.068	265.130
350000	16.209	289.359	0.134	8.315	20.876	15.250	2.579	9.810	19.501	8466.904	2.250	19050.533	0.020	0.040	0.002	2365.4	0.016	0.068	328.106
400000	16.209	289.359	0.134	8.315	20.876	15.250	2.579	9.810	22.287	8466.904	2.500	21167.259	0.020	0.040	0.002	2493.4	0.016	0.068	395.262
450000	16.209	289.359	0.134	8.315	20.876	15.250	2.579	9.810	25.073	8466.904	2.750	23283.985	0.020	0.040	0.002	2615.1	0.016	0.068	466.373
500000	16.209	289.359	0.134	8.315	20.876	15.250	2.579	9.810	27.859	8466.904	3.000	25400.711	0.020	0.040	0.002	2731.4	0.016	0.068	541.234
550000	16.209	289.359	0.134	8.315	20.876	15.250	2.579	9.810	30.645	8466.904	3.250	27517.437	0.020	0.040	0.002	2842.9	0.016	0.068	619.668
600000	16.209	289.359	0.134	8.315	20.876	15.250	2.579	9.810	33.431	8466.904	3.500	29634.163	0.020	0.040	0.002	2950.2	0.016	0.068	701.520
650000	16.209	289.359	0.134	8.315	20.876	15.250	2.579	9.810	36.217	8466.904	3.750	31750.888	0.020	0.040	0.002	3053.7	0.016	0.068	786.654



700000	16,209	289,359	0,134	8,315	20,876	15,250	2,579	9,810	39,002	8466,904	4,000	33867,614	0,020	0,040	0,002	3153,9	0,016	0,068	0,004	874,949
750000	16,209	289,359	0,134	8,315	20,876	15,250	2,579	9,810	41,788	8466,904	4,250	35984,340	0,020	0,040	0,002	3251,0	0,016	0,068	0,004	966,296
800000	16,209	289,359	0,134	8,315	20,876	15,250	2,579	9,810	44,574	8466,904	4,500	38101,066	0,020	0,040	0,002	3345,2	0,016	0,068	0,004	1060,598