BIDIRECTIONAL
IMPULSE TURBINES FOR
THERMOACOUSTIC DEVICES

MICHAEL TIMMER

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BIDIRECTIONAL IMPULSE TURBINES FOR THERMOACOUSTIC DEVICES

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Summary

A thermoacoustic device is an attractive machine to convert low-grade heat into useful power in a cost-effective manner. Due to the lack of moving parts in the hot regions of the device, the system is inherently reliable and requires little maintenance. It operates using noble gases and the simplicity of the design ensures that it is still economically attractive to utilize low-grade heat. The produced acoustic power is mostly used for refrigeration, but it is also possible to convert the acoustic power into electricity.

This thesis starts with an extensive literature review on the four main methods to convert thermoacoustic power into electricity. Devices with piezoelectric components are found to only produce a very low power output at a moderate efficiency, while magnetohydrodynamic solutions show a theoretical promise, but practical implementation is hard due to the power transfer across the gas-liquid interface. The most widely used acoustic to electric transducer is the linear alternator, which can reach a conversion efficiency of 75 % and is shown to work up to the kW range. However, the linear alternator is very expensive relative to the rest of the thermoacoustic device, and it shows increasing problems when scaling to larger power outputs. An attractive alternative is the use of a bidirectional turbine, which has been shown in oscillating water columns to convert wave power into electricity up to the MW range at an efficiency of $\sim 40\%$. While the maximum efficiency is lower than that of a linear alternator, the bidirectional turbine is easily scalable and much cheaper, such that a low-grade heat source can still be used in a cost-effective manner.

Even though the bidirectional turbines show a good promise for thermoacoustic devices, there is a lack of a fundamental basis and examples of practical applications in literature. Therefore, in this thesis the performance of bidirectional impulse turbines is studied under varying operating conditions. This is done by using a loudspeaker as the acoustic source, while the power conversion is done by 3D printed turbine prototypes that are coupled to a generator. The experimental procedure is extensively calibrated and validated, such that small changes in operating conditions and the turbine

design can be reliably measured.

To provide a fundamental basis for measurements with bidirectional turbines in thermoacoustic devices, the performance of a first turbine prototype is characterized under varying acoustic amplitude and frequency for three generator loads. In contrast to conventional turbomachinery, it is found that the flow coefficient and following velocity diagram do not uniquely determine the turbine efficiency. This is caused by the phase difference between pressure and velocity, which introduces an extra variable that can be included by using the acoustic power difference over the turbine as a performance indicator. In doing so, the turbine efficiency is shown to be independent of the acoustic frequency. This is an important advantage over linear alternators, since no frequency matching with a thermoacoustic device is necessary for efficient performance. For a given acoustic power difference, the turbine efficiency is shown to vary as a function of the generator load, but the maximum efficiency is around 25 % for all cases. Furthermore, by dimensional analysis a combination of the flow coefficient and acoustic power difference is found which does uniquely determine the turbine efficiency. The identified thermoacoustic input coefficient can be used for scaling and determining the most efficient operating point in varying thermoacoustic conditions. To implement the bidirectional turbine in a thermoacoustic device, the real and imaginary part of the turbine impedance are given as a function of varying operating conditions, and all results are presented in a supplementary data publication.

With the relevant performance indicators identified, the same experimental set-up is used to optimize the performance of the bidirectional impulse turbine. For this purpose, many different turbine prototypes are 3D printed and their performance is measured under varying operating conditions. It is found that including a shroud ring around the rotor has a positive influence on the turbine efficiency, especially for a large tip clearance. This can be explained by the reduced leakage flow over the tips of the rotor blades. For decreasing tip clearance, both the turbines with shrouded and unshrouded rotors become significantly more efficient. For a very small tip clearance there is a break-even point, after which the unshrouded rotor is more efficient. Especially when scaling the turbine design to larger sizes than used in this work, it is relatively easier to produce a small tip clearance, such that an unshrouded rotor is more efficient. For four different turbines, the axial spacing between the rotor and guide vanes is also varied. A significant decrease in turbine efficiency is found for an increasing spacing, where also a difference in maximum turbine efficiency is present as a function of the acoustic frequency and the generator load. The influence of spacing is in contrast to literature from oscillating water columns in the same range. This difference is identified as an effect of the displacement amplitude of the wave, which is relatively small in acoustic flow. A clear trend in maximum turbine efficiency is found by examining the displacement in relation to the axial spacing for

varying operating conditions. Finally, a design study is performed, focusing on the geometry of the guide vane and rotor blades. No large differences in turbine efficiency are found in the investigated range. However, it is shown that using a slightly larger rotor angle than guide vane angle has a positive effect on the turbine performance. When combining all efforts to optimize the turbine, an efficiency of approximately 40 % can be reached, which is in the same range as for similar turbine designs in oscillating water columns.

To check whether the results from the lab experiments can be reproduced in practice, one of the turbine prototypes is implemented in a thermoacoustic refrigerator. For this purpose, the measured turbine impedance is used in a thermoacoustic model of the refrigerator to identify a good location for implementing the bidirectional turbine. For the chosen position, after correcting for acoustic losses in two flange connections, a maximum turbine efficiency of 36 % is reached. This performance is found to be in good agreement with the lab experiments, therewith validating the experimental procedure. To further test the bidirectional turbine under varying operating conditions, the mean pressure and gas type in the thermoacoustic refrigerator are varied, which was not possible in the lab experiments. For all measurements the same maximum turbine efficiency is found, which is in contrast to previous literature where a significant increase in efficiency as a function of the mean pressure was found. It is concluded that the results from the current work are more likely correct, since the acoustic power has been directly measured instead of being estimated. For the large range of operating conditions, it is found that scaling with the thermoacoustic input coefficient still holds. Furthermore, the amount of acoustic power that is absorbed by the turbine is shown to be a function of the real part of the turbine impedance. Finally, a case study is presented in which the electricity produced by the turbine is exactly enough to power the fluid pumps of the thermoacoustic refrigerator. The remaining acoustic power can then be used for cooling, therewith providing a completely off-grid thermoacoustic refrigerator working purely on heat. Besides this use for the bidirectional turbine, several other options and areas of application are discussed in the final part of this thesis.

Samenvatting

Een thermoakoestisch apparaat is een aantrekkelijke machine om laagwaardige warmte op een kosteneffectieve manier om te zetten in nuttig vermogen. Vanwege het ontbreken van bewegende delen in de warme gebieden van het apparaat, is het systeem inherent betrouwbaar en vereist het weinig onderhoud. Het werkt met edelgassen en de eenvoud van het ontwerp zorgt ervoor dat het nog steeds economisch aantrekkelijk is om laagwaardige warmte te gebruiken. Het geproduceerde akoestische vermogen wordt meestal gebruikt voor koeling, maar het is ook mogelijk om het akoestische vermogen om te zetten in elektriciteit.

Dit proefschrift begint met een uitgebreid literatuuroverzicht over de vier belangrijkste methoden om thermoakoestische energie om te zetten in elektriciteit. Apparaten met piëzo-elektrische componenten produceren een laag uitgangsvermogen met een middelmatige efficiëntie, terwijl magnetohydrodynamische oplossingen een theoretische belofte tonen, maar praktische implementatie is moeilijk vanwege de vermogensoverdracht over het gas-vloeistof contactvlak. De meest gebruikte akoestisch naar elektrisch omzetter is de lineaire alternator, die een conversie-efficiëntie van 75 % kan bereiken en waarvan is aangetoond dat deze werkt tot het kW-bereik. De lineaire alternator is echter erg duur in vergelijking met de rest van het thermoakoestische apparaat en vertoont toenemende problemen bij het schalen naar grotere vermogens. Een aantrekkelijk alternatief is het gebruik van een bidirectionele turbine, waarvan is aangetoond in oscillerende waterkolommen dat golfvermogen omgezet kan worden in elektriciteit tot het MW-bereik met een efficiëntie van $\sim 40\%$. Hoewel het maximale rendement lager is dan dat van een lineaire alternator, is de bidirectionele turbine gemakkelijk schaalbaar en veel goedkoper, zodat een warmtebron van lage kwaliteit nog steeds op een kosteneffectieve manier kan worden gebruikt.

Hoewel de bidirectionele turbines veelbelovend zijn voor thermoakoestische toepassingen, ontbreekt er een fundamentele basis en praktische toepassing in de literatuur. Daarom wordt in dit proefschrift de prestaties van bidirectionele impuls turbines bestudeerd onder verschillende werkom-

standigheden. Dit wordt gedaan door een luidspreker als akoestische bron te gebruiken, terwijl de stroomconversie gebeurt door 3D-geprinte turbine-prototypen gekoppeld aan een generator. De experimentele procedure is uitgebreid gekalibreerd en gevalideerd, zodat kleine veranderingen in werkomstandigheden en het turbineontwerp met vertrouwen kunnen worden gemeten.

Om een fundamentele basis te bieden voor metingen met bidirectionele turbines in thermoakoestische apparaten, zijn de prestaties van een eerste turbine prototype gekarakteriseerd door een variërende akoestische amplitude en frequentie voor drie generatorbelastingen. In tegenstelling tot conventionele turbomachines is gevonden dat de stroomcoëfficiënt en het bijbehorende snelheidsdiagram niet op unieke wijze de turbine-efficiëntie bepalen. Dit wordt veroorzaakt door het faseverschil tussen druk en snelheid, wat een extra variabele introduceert welke meegenomen kan worden door het akoestische vermogensverschil over de turbine te gebruiken als een prestatie indicator. Wanneer dit gebruikt wordt blijkt de turbine-efficiëntie onafhankelijk te zijn van de akoestische frequentie. Dit is een belangrijk voordeel ten opzichte van lineaire alternatoren, omdat geen frequentie afstelling met een thermoakoestisch apparaat nodig is voor efficiënte prestaties. Voor een gegeven akoestisch vermogensverschil blijkt de turbine-efficiëntie te variëren als functie van de generatorbelasting, maar de maximale efficiëntie is voor alle gevallen ongeveer 25 %. Verder is door dimensieanalyse een combinatie van de stroomcoëfficiënt en het akoestische vermogensverschil gevonden die wel op unieke wijze de turbine-efficiëntie bepaalt. De geïdentificeerde thermoakoestische ingangcoëfficiënt kan worden gebruikt voor het schalen en bepalen van het meest efficiënte werkpunt in variërende thermoakoestische omstandigheden. Om de bidirectionele turbine in een thermoakoestisch apparaat te implementeren, is het reële en imaginaire deel van de turbine-impedantie gegeven als functie van variërende werkomstandigheden, en zijn alle resultaten gepresenteerd in een aanvullende datapublicatie.

Met de geïdentificeerde prestatie indicatoren wordt dezelfde experimentele opstelling gebruikt om de prestaties van de bidirectionele impuls turbine te optimaliseren. Voor dit doel zijn veel verschillende turbine-prototypen 3D-geprint en hun prestaties zijn gemeten onder verschillende werkomstandigheden. Het blijkt dat het opnemen van een mantelring rond de rotor een positieve invloed heeft op de turbine-efficiëntie, vooral voor een grote tipspeling. Dit kan worden verklaard door de verminderde lekstroom over de uiteinden van de rotorbladen. Voor het verkleinen van de tipspeling worden zowel de turbines met gehulde en ongehulde rotoren aanzienlijk efficiënter. Voor een zeer kleine tipspeling is er een break-even punt, waarna de ongehulde rotor efficiënter is. Vooral bij het schalen van het turbineontwerp naar grotere afmetingen dan bij dit werk, is het relatief eenvoudiger om een kleine tipspeling te produceren, zodat een ongehulde rotor efficiënter

is. Voor vier verschillende turbines is de axiale afstand tussen de rotor en de stator ook gevarieerd. Een significante afname in turbine-efficiëntie is gevonden voor een toenemende afstand, waar ook een verschil in maximale turbine-efficiëntie aanwezig is als een functie van de akoestische frequentie en de generatorbelasting. De invloed van axiale afstand is in tegenstelling tot resultaten uit de literatuur van oscillerende waterkolommen in hetzelfde bereik. Dit verschil is geïdentificeerd als een effect van de verplaatsingsamplitude van de golf, die relatief klein is in akoestische stroming. Een duidelijke trend in maximale turbine-efficiëntie is gevonden door de verplaatsing van de golf te onderzoeken in relatie tot de axiale afstand voor variërende werkomstandigheden. Ten slotte is een ontwerpstudie uitgevoerd, gericht op de geometrie van de stator- en rotorbladen. Er zijn geen grote verschillen in turbine-efficiëntie gevonden in het onderzochte bereik. Er is echter aangetoond dat het gebruik van een iets grotere rotorhoek dan de hoek van de statorbladen een positief effect heeft op de prestaties van de turbine. Bij het combineren van alle inspanningen om de turbine te optimaliseren, kan een efficiëntie van ongeveer 40 % worden bereikt, wat in hetzelfde bereik ligt als voor vergelijkbare turbineontwerpen in oscillerende waterkolommen.

Om te controleren of de resultaten van de lab-experimenten in de praktijk kunnen worden gereproduceerd, is één van de turbineprototypen geïmplementeerd in een thermoakoestische koeler. Voor dit doel is een thermoakoestisch model van de koeler gebruikt om een goede locatie te vinden voor het implementeren van de bidirectionele turbine. Voor de gekozen positie is, na correctie voor akoestische verliezen in twee flensverbindingen, een maximale turbine-efficiëntie van 36 % bereikt. Deze prestatie komt goed overeen met de lab-experimenten, waarmee de experimentele procedure is gevalideerd. Om de bidirectionele turbine onder verschillende werkomstandigheden verder te testen, zijn de gemiddelde druk en het gas type in de thermoakoestische koeler gevarieerd. Voor alle metingen is dezelfde maximale turbine-efficiëntie gevonden, wat in tegenstelling is tot eerdere literatuur waar een grote toename van de efficiëntie als functie van de gemiddelde druk werd gevonden. De resultaten van het huidige werk zijn waarschijnlijk correct, omdat het akoestische vermogen direct is gemeten in plaats van afgeschat. Voor het grote bereik van verschillende werkomstandigheden is gebleken dat de schaling met de thermoakoestische ingangscoefficiënt nog steeds geldt. Verder is aangetoond dat de hoeveelheid akoestisch vermogen die wordt geabsorbeerd door de turbine een functie is van het reële deel van de impedantie. Ten slotte wordt een studie gepresenteerd waarin de door de turbine geproduceerde elektriciteit precies genoeg is om de pompen van de koeler te draaien. Het resterende akoestische vermogen kan vervolgens worden gebruikt voor koeling, waardoor een volledig off-grid thermoakoestische koeler is voorgesteld die puur op warmte werkt. Naast deze mogelijkheid voor de bidirectionele turbine worden in het laatste deel van dit proefschrift verschillende andere opties en toepassingsgebieden besproken.

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1

Introduction

Initiated by the industrial revolution, the global energy demand is growing at an ever increasing rate as the population size and living standards rise. However, in the grand scheme of things, the amount of energy in question is actually relatively small. In only a few hours, our average sized sun delivers more energy to the earth than consumed by the human race in over a year.¹ The difficulty lies in capturing, storing, and using this energy, which has not been done in a sustainable way so far. The solar energy that has been captured and stored by organisms over millions of years, is currently being depleted in only a few hundred years. Not only are these fossil fuels used at an extreme rate, the energy storage in chemical form causes the emission of greenhouse gases upon usage. This way of energy consumption is clearly not sustainable, and can cause a rapid climate change that is harmful to humans, and more importantly, to many organisms on this planet.

1.1 Utilizing low-grade heat

The need for an energy transition into a sustainable system is becoming clear to more and more people, but much work still remains in completing this task. Due to the wide variety in which energy is used, advancements have to be made in many different fields of technology. For instance, the International Energy Agency identifies cooling as a field that is critical, yet often overlooked in solving the energy issues.² On average, cooling accounts for 20 % of the global energy use in buildings, while locally this percentage can be much higher. Furthermore, with the economic and demographic rise especially in warmer climates, the energy demand for air conditioning stands to triple by 2050 if left unchecked.² Since nearly all of the cooling is pro-

Figure 1.1: Solar tube collectors that use thermal oil to capture the heat absorbed from solar radiation.

vided by electrically driven vapor compression cycles, the electrical grid can become oversaturated, especially during peak hours in the summer.

Following the electricity need for cooling, the modern society is developing in such a way that there is a general increase in electricity demand across the board. Satisfying this need in an efficient and increasingly sustainable way can partly be done by utilizing low-grade heat, such that widely available waste heat or solar power can be used as an energy source. Especially in the industrial sector, there is a significant amount of heat wasted. In the European Union, approximately 17 % of the industrial heat consumption is discarded as waste heat, of which most is in the 100 °C to 200 °C range.³ Although this heat is mostly produced by fossil fuels, and thus not renewable, utilizing the otherwise wasted heat is still beneficial to reducing emissions and increasing the efficiency of energy consumption. Alternatively, completely renewable heat sources can be used, such as solar power that can be harvested with a solar tube collector. In Fig. 1.1, such a solar collector is depicted which can use solar power to heat a working fluid up to 200 °C.

The low-grade heat sources can be used to produce electricity through several thermodynamic cycles,⁴ of which a well known example is the Organic Rankine Cycle (ORC).⁵ An ORC produces electricity by expanding a working fluid through a turbine, similar to water-steam systems, but the main difference is the lower temperature at which the working fluid changes phase. The latter ensures that low-grade heat can still be used to produce power. However, similar to vapor-compression cooling systems, the working fluids are usually human-made hydrofluorocarbons (HFCs), which are less destructive to the ozone layer than previous refrigerants, but still have a significant impact on climate change. Therefore, the Montreal protocol to prevent ozone layer depletion has been amended in 2016 by the Kigali

agreement, which strives to reduce HFCs by more than 80 % over the next thirty years.⁶ Furthermore, even if the utilization of low-grade heat does not happen with HFCs, the process still relies on a phase-change, and must therefore operate around the evaporation temperature. However, many low-grade heat sources will fluctuate in temperature over time, therewith making it more complex to run these kind of systems on low-grade heat sources.

As an alternative, thermoacoustic devices use a thermodynamic process that is not reliant on a phase change and it can be performed with non-harmful working fluids such as noble gases. The thermoacoustic effect is used to convert (low-grade) heat into useful work, which can be done for a temperature difference with the environment as small as ~ 30 K.⁷ To provide an alternative to electric vapor-compression cooling systems, the acoustic power in thermoacoustic devices is mostly used to provide refrigeration, but it can also be converted into electric power. More details about these thermoacoustic devices are given in Sec. 1.3, but since the field of thermoacoustics is still relatively small and unknown, first an introduction to the working principle of thermoacoustic devices is given in the next section.

1.2 Thermoacoustic effect

Thermoacoustics encompasses the fields of thermodynamics and acoustics. While acoustic waves are usually regarded as pressure and velocity oscillations, there is also a fluctuating temperature component, which is generally very small. For example, at the level of humans speech, the temperature oscillation of the sound wave is in the order of $10 \mu\text{K}$.⁸ However, this temperature fluctuation can be used to increase to amplitude of the acoustic wave by adding and subtracting heat during the right time of the oscillation. As Lord Rayleigh first described it in 1887:

“If heat be given to the air at the moment of greatest condensation, or taken from it at the moment of greatest rarefaction, the vibration is encouraged.”

— Lord Rayleigh⁹

This conversion of heat into acoustic power, and vice versa, is referred to as the thermoacoustic effect, and can be used to achieve very high acoustic amplitudes. For example, while a shouting human can produce approximately 1 mW of acoustic power,¹⁰ thermoacoustic devices generally work at power levels 5 to 7 orders of magnitude higher.

When the thermoacoustic effect is used to produce acoustic power, the component in question is often referred to as a prime mover, while for the inverse it is called a heat pump or refrigerator. In an attempt to explain both of these conversions, a gas parcel is followed as it oscillates near a solid wall. Inspired by the animations from Swift,¹⁰ Fig. 1.2 presents a schematic of the temperature and position of the gas parcel for both cases. The conditions

Figure 1.2: Temperature of an oscillating gas parcel near a wall for two distinctive cases. For the prime mover (a), the temperature gradient of the wall is larger than that experienced by the gas parcel, while for the refrigerator (b), this is the other way around. The arrow denotes the corresponding heat transport, δQ , from the wall to the gas parcel and vice versa.

are such that as the gas parcel moves towards the right, it is compressed and its temperature rises, while towards the left it expands and its temperature drops. For the prime mover, the gradient of the wall temperature is larger than that experienced by the traveling gas parcel. Therefore, following the description from Lord Rayleigh, heat is added when the gas parcel is in its compressed state, while heat is taken from it when it has expanded. This increases the amplitude of the acoustic wave while reducing the temperature gradient of the wall, therewith converting heat into acoustic power. Similarly, the inverse happens at the refrigerator, where the temperature gradient of the wall is smaller than that of the gas parcel. This results in a reduction of the acoustic amplitude, while the temperature gradient of the wall is increased. In this way, the thermoacoustic effect converts the available acoustic power into useful heat transport. Although such an acoustic heat pump can be used to upgrade heat, it is primarily used for the refrigeration that is provided at the cold side of the wall. Alternatively, the acoustic power can be used by additional components that convert the available energy into electricity.

1.3 Thermoacoustic devices

Utilizing the thermoacoustic effect has resulted in the first thermoacoustic devices around the 1950's from Bell Telephone Laboratories.^{11,12} These engines converted a temperature gradient into standing waves using 'singing pipes',¹³ and subsequently produced electricity from the acoustic power using an electromagnetic converter. Although promising due to the simple

and reliable concepts, the overall efficiency was still unsatisfactory. Ceperley stated that the latter was mainly due to the fact that their engines were based on standing wave phasing, where an imperfect heat transfer has to be present to facilitate the necessary phasing between the pressure and particle velocity.¹³ As an alternative, he proposed to use devices working on traveling waves,¹⁴ in which the gas undergoes a cycle similar to the inherently efficient Stirling cycle.¹⁵ In the past few decades after the work of Ceperley, both the standing wave and traveling wave thermoacoustic devices have been further developed. A unifying perspective of both thermoacoustic branches with their underlying mathematics and working principles is given in the book of Swift.¹⁰

Thermoacoustic devices can be used wherever there is a sufficient heat source available. The onset temperature difference for the device to start producing power depends on the design and operating conditions (e.g. mean pressure and working fluid), but it can be relatively low compared to other technologies. For example, a four-stage traveling wave engine has been shown to start producing acoustic power for a temperature difference as low as ~ 30 K.⁷ This opens the market for thermoacoustic devices wherever there is such a relatively small temperature difference available. As long as done cost-effectively, this can either boost the efficiency of current systems having a stream of unused waste heat or utilize a (possibly sustainable) heat source for stand-alone thermoacoustic devices. Possible application areas for thermoacoustics include waste-heat recovery,^{16–19} solar powered devices,^{19–21} and small low-cost applications for e.g. rural areas.^{22–27} An overview of more applications and thermoacoustic devices can be found in the works of Garrett²⁸ and Jin.²⁹

Besides the low temperature difference required to operate, thermoacoustics has gotten increasing attention due to several other advantages it has over competing technologies. One of the most important characteristics is the need for no or, in case of producing electricity, few mechanically moving parts. Furthermore, any moving part is not situated near the high temperature region of the thermoacoustic device, therewith reducing the material requirements when compared to conventional technologies such as automotive engines or Rankine cycle based power plants. The lack of moving parts and low material requirements make thermoacoustic devices inherently simple, robust, and economical to produce and results in reliable devices that require little to no maintenance. As Ceperley stated, this lack of moving parts, simplicity and reliability also makes thermoacoustics attractive for isolated equipment.¹³ Examples of this are outer space applications^{30–32} and remote sensing techniques.^{33,34} Further advantages of using thermoacoustic devices are that they work on noble and inert gases (e.g. helium and argon) and that they do not rely on a phase change during the thermodynamic cycle.³⁵ Therefore, no harmful, ozone depleting refrigerants are needed and operation over a wide range of temperatures is possible since the devices do

Figure 1.3: Thermoacoustic refrigerator in a 4-stage traveling wave configuration rated for 25 kW of cooling power. The two vessels on the left produce the acoustic power (prime movers), and the two vessels on the right produce cooling. The pump for the hot circuit is present between the vessels on the left, while two additional pumps are needed for the ambient and cold circuits.

not have to operate around a phase transition temperature.

With the general properties of the thermoacoustic devices given, it can be said that thermoacoustic refrigerators are a promising alternative to replace conventional vapor compression systems in a sustainable way. While this class of cooling devices is still maturing, several systems have become commercially available, such as the one presented in Fig. 1.3. These thermoacoustic coolers are mostly driven by thermal power, but some electricity is typically still necessary to drive the pumps of the fluid circuits. Since the electricity need is relatively small, and acoustic power can also be converted into electric power instead of cooling, it should be possible to drive the fluid pumps from acoustic power while a sufficient amount remains for cooling purposes. In this way, a completely off-grid thermoacoustic refrigerator can be made which is driven purely by a (sustainable) heat source. Since the thermoacoustic coolers are readily available, this is an interesting application to first introduce the conversion of acoustic power into electricity. Besides such a hybrid version, low-grade heat can also be utilized by a thermoacoustic engine that purely focuses on producing electricity. For either of these options, there is a need to efficiently and reliably convert thermoacoustic power into electricity.

1.4 Thesis outline

The main focus of this work will be on the conversion of acoustic power into electricity, such that it can be used in any thermoacoustic device. This is started by reviewing the four main ways to convert thermoacoustic power into electricity in Chapter 2. From this review, the bidirectional impulse turbine is found to be an attractive option to produce electricity from acoustic

power, especially when scaling up to industrial sizes. Due to the lack of literature on bidirectional impulse turbines in thermoacoustic conditions, the first focus is on proving the concept and carefully investigating the turbine performance under varying experimental conditions.

In Chapter 3, the experimental set-up and the measurement procedure are presented and validated. In Chapter 4, the results for a first turbine prototype are presented. The measurements under varying operating conditions are used to characterize the turbine by identifying the relevant performance indicators and determining the rules of scaling.

In Chapter 5, the design of the bidirectional impulse turbine is varied in an attempt to optimize its performance. Besides a blade design study, this includes results for a comparison between shrouded and unshrouded rotors, as well as the influence of tip clearance and axial spacing between the guide vane and rotor.

In Chapter 6, the optimized turbine design is implemented in a thermoacoustic refrigerator with the help of a thermoacoustic model. The combination of refrigerator and electricity production is operated for varying gas types, mean pressures, and turbine loads. Since the application of a bidirectional turbine to make an off-grid thermoacoustic refrigerator is identified as a promising first application, the feasibility and performance of such a combined system is presented in a case study.

In Chapter 7, the conclusions of this work are given along with several recommendations. This includes an overview of the conclusions from individual chapters, as well as more general conclusions and recommendations, such as the possible application areas of the presented work.

2

Conversion of thermoacoustic power into electricity

This chapter provides an overview of the current technologies for converting acoustic power into electricity. This review work is done since there is a need for a clear overview of the different possibilities with accompanying advantages, disadvantages and recommended areas of application. This should attract designers to the area of thermoacoustics and ensure that they can more easily develop an engine suited for their needs. Besides the overview of current literature, knowledge on gaps and conflicts in current literature should be pointed out, which will make it easier to subsequently provide guidelines about where future thermoacoustic research should be headed. In part these requirements have been met by previous review papers, such as the ones by Avent³⁵ and Pillai.³⁶ However, these articles generally provide a broader view on thermoacoustic energy harvesting, resulting in only a small amount of attention for the acoustic power to electricity conversion. Therefore, some of the conversion methods are not treated and/or not enough in depth knowledge of these is given.

In this chapter, an acoustic to electric specific review is presented, where all the topics pointed out in the previous paragraph are included. This will be done in individual sections for the four main conversion methods that are shown in Fig. 2.1. Electromagnetic devices are treated in Sec. 2.1, piezoelectric devices in Sec. 2.2, magnetohydrodynamic devices in Sec. 2.3, and bidirectional turbines in Sec. 2.4. After the individual methods, conclusions

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Figure 2.1: Overview of different methods for the acoustic to electric conversion.

about these specific sections as well as general conclusions and recommendations are given in Sec. 2.5.

2.1 Electromagnetic devices

This section will provide a review of electromagnetic devices for converting thermoacoustic power into electricity. This will contain the devices that directly use electromagnetic induction to convert acoustic power into electrical power. Besides magnets and coils as the main components for the electromagnetic induction, iron is nearly always present as well for lower costs and a higher transduction efficiency. The main principle of the devices is that the acoustic power initiates mechanical movement of one component relative to the other two, therewith inducing an electric current in the coil. There are different configurations possible to ensure this relative movement. An overview of the options used in thermoacoustics will be given in Sec. 2.1.1.

After introducing the different configurations, details about the use of them in thermoacoustic engines are given in Sec. 2.1.2. This will include advantages, disadvantages and details of different aspects of these electromagnetic devices in the field of thermoacoustics. For the latter, the most important topics are the efficiency of the acoustic to electric conversion, the maximum electric power output, and the coupling between the acoustic field and the electromagnetic component.

2.1.1 Configurations

Conventional loudspeakers use electromagnetic induction to produce mechanical movement from electrical power. However, the loudspeakers can also be used in reverse to produce electricity, just as an electric motor can be used in reverse as a generator. Most loudspeakers use a moving coil in combination with static permanent magnets and iron, as shown schematically in Fig. 2.2. An incident sound wave will force the cone to move axially, therewith moving the connected coil with respect to the permanent magnets and

Figure 2.2: Schematic of a moving coil loudspeaker.

inducing an electric current.

Besides using a loudspeaker in reverse, one can also use the group of devices dedicated for converting acoustic power to electricity, namely linear alternators. In principle loudspeakers and linear alternators can be seen as the same electromagnetic devices. However, a distinction is appropriate since the main purpose of loudspeakers is to produce sound with a flat response over a large frequency range. In contrast, linear alternators are designed to convert acoustic power to electricity at a single resonance frequency, which can result in quite different characteristics for both devices. This distinction can be seen in Fig. 2.1, where the linear alternator is further divided into moving magnet and moving coil devices, whilst practically all loudspeaker configurations use a moving coil to minimize the amount of moving mass.

2.1.1a Loudspeakers

A typical schematic representation of a moving coil loudspeaker in a tube is shown in Fig. 2.2. Such commercial loudspeakers are used in thermoacoustics mainly because they are relatively cheap and readily available. This makes them well suited for simple, initial experiments and low cost applications such as envisaged in the SCORE (Stove for Cooking, Refrigeration and Electricity supply) project.^{22,26,27} However, loudspeakers usually have a poor power-transduction efficiency,¹⁰ since their design is more focused on linearity than efficiency³⁷ and because they are designed to have a flat response over a large frequency range. Although loudspeakers should not generally be preferred over linear alternators, they can still be used if low costs are extremely important. This could be the case in situations where there is an abundant amount of (usually low-grade) heat. Note that loudspeakers are mainly used in low power thermoacoustics, due to their weak and fragile paper cones, their limited stroke lengths, and poor impedance match at a high mean pressure of the gas.¹⁰ Loudspeakers are not generally suitable at high power and acoustic amplitudes where there is a high pressure difference across the cone, but they may still be well usable when this pressure

Figure 2.3: Schematic of a moving magnet linear alternator.

difference is in the kPa range.²⁶ Furthermore, robustness against more extreme operating conditions can be acquired by replacing the conventional cone with tougher yet still lightweight materials, such as aluminum or carbon fiber.¹⁷ Tijani *et al.* have shown this concept by modifying a loudspeaker with a 0.1 mm thick aluminum cone.³⁸

To select a suitable loudspeaker one can look at the procedure set out by Kang *et al.*,³⁹ which is based on the method by Yu *et al.*,²⁴ where both mainly focus on the acoustic coupling (see Sec. 2.1.2b for more details) between the loudspeaker and the rest of the thermoacoustic engine. Besides these works, there are several other representative applications of loudspeakers in thermoacoustic engines.^{17,22,26} Typical operating pressures, efficiencies and power outputs for loudspeakers in thermoacoustic engines are given in Sec. 2.1.2a.

2.1.1b Linear alternators

The term linear alternator is used for all devices that are dedicated for converting acoustic power into electricity using electromagnetic induction. In thermoacoustics there are two main configurations for linear alternators: moving coil and moving magnet. The moving coil designs use the same principle as the loudspeaker shown in Fig. 2.2, but don't have the weak and fragile paper cone that commercial loudspeakers have. For the moving magnet linear alternators the iron is either connected to the moving magnet, as is shown schematically in Fig. 2.3, or the iron remains fixed. In either case, the structure with the coil is fixed and the permanent magnet is forced to move axially (or linearly) by the incident sound wave. Although both moving coil and moving magnet alternators have been widely studied as a power producing component in thermoacoustics, the moving magnet linear alternators are more extensively used. It is worth noting that the linear alternators could also have a configuration where only the iron is moving, which is sometimes done in an effort to remove the costly permanent magnets.³⁷ However, there is only a single example in the thermoacoustic literature of such an alterna-

Figure 2.4: Calculated relative seal gap losses of a linear alternator as a function of the seal gap width for several frequencies.

tor.⁴⁰ Due to this lack of practical applications and available literature, the moving iron alternators are omitted in the rest of this work. A wide range of moving coil, magnet and iron linear alternator configurations for general purposes is presented in the book of Boldea and Nasar.⁴¹

For any linear alternator design, flexure bearings ensure a very robust and reliable design.^{30,42} These metal plates provide a low stiffness in axial direction but a high resistance to radial and rotational movement. Therefore, the linear alternator can oscillate in the outer tube with seal gaps (see Fig. 2.3) as small as $10\ \mu\text{m}$ without wearing against the outer tube.³⁰ These small gaps are essential to reduce blow-by and viscous friction inside this seal.⁴³ Fig. 2.4 illustrates the magnitude of the seal gap losses with respect to the total alternator power as a function of the seal gap width. This represents a combination of basic calculations on the blow-by losses and shear losses of a typical, commercial linear alternator (John Corey, Qdrive, private communication, 2002). The results show that the seal gap losses increase severely for larger seal gaps, where the relative power loss can easily reach 5 % or more. To have an efficient linear alternator, one should therefore pay close attention to minimizing the seal gap dimensions. Furthermore, it can be seen from Fig. 2.4 that a higher acoustic frequency will cause relatively less power dissipation. This is caused by the linear increase of piston power as a function of frequency, whilst the seal gap losses only slightly increase as a function of frequency due to increasing shear losses.

As an alternative to linear alternators with a seal gap, one could use flexure seal designs based on metal or composite bellows.^{44,45} These designs do not require an axial alignment that must be accurate to within a few microns and eliminate the existence of the seal gap with accompanying losses. So far, these linear alternators have only been designed as acoustic drivers, where the seal gap is eliminated by bellows that connect the piston to the thermo-

coustic refrigeration part in a flexible manner.^{44,45} A similar design can be envisioned for the acoustic to electric conversion, where the bellows should connect the outer tube with the piston of the alternator, however these designs are yet to be constructed and tested.

Thermoacoustic engine designs with linear alternators often use a double-acting configuration where the alternators are placed in pairs.^{30,46–48} If it is ensured that the alternators are phased correctly, they can counteract each other's vibrations. This provides a balanced design that is much less affected by spurious vibrations than a single acting configuration can have. This idea can of course be extended to more linear alternators, as long as they have a plane of symmetry such that they can balance each other.

Compared to commercial loudspeakers, linear alternators generally have a smaller range of mechanical resonance, a larger mass, and a higher power-transduction efficiency.¹⁰ They can also produce much more electrical power since they have larger stroke lengths and can operate at higher mean pressures and larger acoustic amplitudes. However, these dedicated components require more precision manufacturing, e.g. to minimize the losses through the seal gap shown in Fig. 2.3, and are made in much smaller volumes, thereby also making them significantly more expensive. Commercial linear alternators generally cost a few thousand dollars, while loudspeakers of similar dimensions cost around a hundred dollars. A linear alternator for large powers with a high efficiency can therefore easily be the most expensive part of a thermoacoustic engine. Furthermore, the mass of the linear alternator is the main contributor to the overall mass of a thermoacoustic engine.³⁰ Nevertheless, compared with loudspeakers the increased robustness, higher efficiency and larger power output of the linear alternator still make it a generally better option. An overview of linear alternator power outputs and efficiencies that have been achieved in thermoacoustics are given in Sec. 2.1.2a, where this can also be compared with that of commercial loudspeakers.

2.1.2 Characteristics

With the different types of electromagnetic devices described in the previous section, it is interesting to look at several performance characteristics such as efficiency and power output (see Sec. 2.1.2a) to be able to compare these devices. Furthermore, some other aspects such as the coupling of the electromagnetic device with the acoustic field (see Sec. 2.1.2b) and analytical and numerical methods to study these devices (see Sec. 2.1.2c) are also treated.

2.1.2a Efficiency and electric power

An overview of the performance of several electromagnetic devices, often situated in an entire thermoacoustic engine, is given in Table 2.1. The operating frequency for these typical thermoacoustic applications is in the range

of 50 Hz to 150 Hz. The most used devices are moving magnet linear alternators, which are often homemade. Although the exact dimensions of the homemade designs are often not given, their listed mechanical and electrical properties such as amount of moving mass and the transduction coefficient can still be used to characterize the device. Furthermore, two commercial alternators are also used and listed in Table 2.1, namely one by Lihan Thermoacoustic Technologies⁴⁹ and one by Qdrive (Chart Industries).⁴⁸ It is noted that the Qdrive 2s297 is originally designed as a compressor for cryocoolers⁴⁸, but still reaches an acoustic to electric conversion efficiency of 73 %. Dedicated linear alternators, if produced with high precision and used properly, can reach efficiencies in the range of 80 %-90 %. These alternators will generally be quite expensive, which is why often an attempt is made to produce homemade alternators for prototype engines. As can be seen from Table 2.1, these alternators still have an efficiency in the range of 65 %-75 %.

Besides dedicated linear alternators, there has been quite some effort in using loudspeakers for the acoustic to electric conversion. As can be seen in Table 2.1, the efficiency of the loudspeakers is in the range of 35 %-60 %. It is interesting to see that the extremities of this range are reached with the same commercial loudspeaker (B&C 6PS38), but in different experimental set-ups. Furthermore, the works using a loudspeaker focus on low-cost applications with a relatively small power output in the range of ~ 10 W-200 W, where the reported 200 W is actually for two loudspeakers.³⁹ For the linear alternators the highest power outputs in Table 2.1 are 4690 W and 2300 W, for six and two moving magnet linear alternators, respectively. This yields 1150 W for the Qdrive linear alternator and 100 W for a commercial loudspeaker, which is about an order of magnitude difference. It is still quite interesting that a standard commercial loudspeaker can be used at 18 bar mean pressure with an amplitude of 0.2 bar to produce 100 W for at least a short period without rupturing.³⁹ However, much more power output and reliability can not be expected from loudspeakers. Therefore, one should either use a lot of loudspeakers under relaxed conditions, or preferably, use linear alternators if a power output larger than a few hundred Watt is desired.

As shown in Table 2.1, so far linear alternators in thermoacoustic engines have reached electrical power outputs in the kW range. To increase this power to the MW range, the dimensions of the thermoacoustic engine have to be increased, resulting in a lower operating frequency. This reduces the electromagnetic induction, causing the need for larger and stronger magnets which significantly increase the cost and mass of the alternator. Furthermore, a lower frequency means the alternator should have a larger stroke length within the small tolerance of the seal gap. These necessary adjustments result in a more than linear increase of alternator complexity and cost in terms of upscaling output power, which eventually constrains the practical and economical feasibility.

Table 2.1: Overview compiled from literature of different types of electromagnetic devices with operating characteristics. With η_{a2e} the acoustic to electric conversion efficiency, η_{h2e} the efficiency from heat input to electricity, P_{elec} the amount of electric power generated by the entire engine, ΔT the temperature difference of the heat supply of the engine, P_{mean} the mean operating pressure, drive ratio the percentage of the pressure amplitude divided by the mean pressure, and device specs the specifics of the electromagnetic conversion device if available. All values are from experimental work, unless denoted with an asterisk (*), and are the maximum values reported in the papers. A notion of n/a means the data could not be found in the paper and n/r means the field is not relevant since it does not apply for the given work.

Type	Year ^{Ref}	η_{a2e} [%]	η_{h2e} [%]	P_{elec} [W]	ΔT [K]	P_{mean} [bar]	Drive ratio [%]	Method	Device specs
Moving coil alt.	2004 ³⁰	75	18	39	620	55	9.8	num/exp	n/a
Moving magnet alt.	2011 ⁴⁶	68	15	481	625	35	4.8	exp	homemade
Moving magnet alt.	2012 ²¹	65	15	481	625	35	5.0	exp	homemade
Moving magnet alt.	2014 ⁵⁰	74	20	1043	635	40	6.5	num/exp	homemade
Moving magnet alt.	2014 ⁵¹	n/a	17	1570	620	50	n/a	num/exp	homemade
Moving magnet alt.	2015 ⁵²	68	18	4690	625	60	7.5	num/exp	homemade
Moving magnet alt.	2013 ⁵³	n/a	12	345	620	30	4.0	num/exp	homemade
Moving magnet alt.	2016 ⁴⁷	51 ⁴⁹	16	750	620	32	5.4	num/exp	commercial ⁴⁹
Moving magnet alt.	2017 ⁴⁸	73	n/a	2300	390	40	9.3	num/exp	Qdrive 2s297
Loudspeaker	2011 ²⁴	60	n/r	3	n/r	1	1	exp	B&C 6PS38
Loudspeaker	2012 ^{17,23}	47	1	12	n/a	1	4.8	num/exp	B&C 6PS38
Loudspeaker	2013 ²²	35	n/a	23	675	1.5	n/a	num/exp	B&C 6PS38
Altered loudspeaker	2012 ⁵⁴	57	n/r	n/a	n/r	1	n/a	num/exp	Halbach array
Loudspeaker	2017 ²⁷	60	1.9	18	500	1	8	num/exp	B&C 8BG51
Loudspeaker	2015 ³⁹	45*	3	204	500	18	1.1*	num/exp	B&C 8NW51

2.1.2b Coupling and impedance

In thermoacoustic engines a working gas is used to transfer work in the form of acoustic power to an electricity generating component. Partly due to the large density difference between the gas and solid parts, this power transmission is not at all trivial. For a good coupling, the resonance frequency of the mechanical and electrical parts of the electromagnetic transducer should equal the working frequency of the engine.^{53,55} Furthermore, the transducer can be seen as an acoustic load. Therefore, acoustic impedance matching is also necessary for an efficient power transmission. Note that acoustic impedance is the ratio of the pressure amplitude over the flow rate, or in other words, the amount of driving force needed for a given volumetric displacement of the gas.

Linear alternators generally need a large force for a small displacement, and can therefore be referred to as 'non-compliant' transducers and have to be placed in a high impedance region of the thermoacoustic engine. This high impedance generally leads to a high pressure drop,¹⁷ therewith creating the necessity for a small seal gap (see Fig. 2.3) and precision engineering. Therefore, the high impedance and placement can ensure a high efficiency for alternators but also makes them relatively expensive. Loudspeakers are low impedance (small force large displacement) transducers and can be referred to as 'ultra-compliant'. They are generally placed in a low impedance region and therefore experience a smaller pressure drop. The fragile paper cone and limited stroke of loudspeakers limits them to produce a high power output. In an effort to improve this, one could take advantage of available loudspeaker technology and produce relatively cheap, robust and ultra-compliant alternators.¹⁷

Independent of the electromagnetic devices, a good acoustic coupling can be achieved by making sure the imaginary part of the acoustic load impedance is near zero whilst the real part is large. Wang *et al.* use this to describe a procedure where they first analyze the impedance of the alternator and rest of the engine separately, after which they utilize this to optimize the acoustic coupling (by varying the alternator load resistance).⁵⁶ Other works have achieved similar results by varying the electric capacitance of the linear alternator⁵² and optimizing the position of it in the thermoacoustic engine.³⁹ It should be noted that the imaginary part of the acoustic load can also be used to tune the resonance circuit. Therefore, retaining a small imaginary part of the load can actually be beneficial for the acoustic coupling and therewith the engine performance.

As shortly mentioned before, for an effective power transmission the resonance frequency of the electromagnetic device should equal that of the rest of the thermoacoustic engine. This can prove to be difficult, because placing such a device changes the acoustic field in the rest of the engine. Especially for linear alternators this can be a problem, since they only have a small

frequency band in which they have resonance and therewith a high power transduction efficiency. The linear alternator dominates the coupling problem, resulting in the need to tune the acoustic circuit to the same resonance frequency. For loudspeakers, it has been shown that at typical operating frequencies of thermoacoustic engines the acoustic to electric efficiency can be relatively constant²⁴ (although lower than for linear alternators, as shown in Sec. 2.1.2a). Furthermore, the displacement amplitude only has a small influence on the efficiency, which is a good property for loudspeakers that depend on large stroke lengths due to their low impedance.²⁴ Nevertheless, for any device acoustic impedance coupling is still critical for a good power transduction efficiency. A few attempts to solve the coupling problem have been given in the previous paragraph. More options will be given in Sec. 2.1.2c, where analytical and numerical methods to study electromagnetic devices and entire thermoacoustic engines are shown.

2.1.2c Other

When developing or selecting an electromagnetic device, it is recommended to start this process with some analytical calculations to get an idea of the operating characteristics and performance. For this purpose, mathematical equations are often derived by using an electric circuit analogy for the alternators^{47,53,57} and loudspeakers.²⁴ Yu *et al.* validate experimentally that the acoustic to electric conversion efficiency can be accurately predicted in this manner.²⁴ In subsequent work they also use these calculations to select a commercial loudspeaker from fourteen possible options, whilst mainly focusing on the acoustic impedance for the highest power output and efficiency.³⁹ Gonen and Grossman extend the analytical calculations by including the velocity distribution, viscous friction, and blow-by losses in the seal gap to accurately predict the acoustic to electric conversion efficiency.^{43,58} Besides purely analytical calculations for the electromagnetic device, one can also use numerical simulations. Saha *et al.* show the use of this by optimizing their double Halbach array linear alternator design by using 2D finite element method simulations.⁵⁴ For a more elaborate mathematical background of calculations dedicated for linear electric generators one can look at the book of Boldea and Nasar.⁴¹

To predict the performance of an entire thermoacoustic engine, one can add acoustic relations to the analytical calculations. The basis of the linear thermoacoustic theory was constructed by Rott⁵⁹ and further developed by Swift.¹⁰ The set of equations following from this linear theory are often solved numerically by the Design Environment for Low-amplitude Thermoacoustic Energy Conversion (DeltaEC).⁶⁰ There are quite a few examples that show the successful use of DeltaEC for thermoacoustic engines with electromagnetic devices by giving the governing equations and experimentally validating the numerical results.^{17,47,52} The examples show that for linear op-

erating conditions (drive ratio $< 10\%$), DeltaEC can be a relatively accurate tool for predicting and optimizing the thermoacoustic engine performance before actually constructing and testing the engine.

Backhaus *et al.* used a one-dimensional numerical model based on a lumped-element electric circuit analogous to the acoustic circuit.^{30,61} Besides a good correspondence of the model with experiments, they also identify the different loss mechanisms of their linear alternator by carefully setting up different experiments.³⁰ This provides a nice overview of the magnitude of the alternator losses, which they use to optimize their device with a specific focus on reducing the total mass and volume for electricity generation aboard a spacecraft. The mass of the electromagnetic device is not only important in these exotic applications but, especially for the amount of moving mass, also in general. Attempting to increase the power output results in a higher moving mass, where eventually a practical limit is reached due to the difficulty to maintain a stable, large stroke amplitude in the seal gap under the extreme periodic forces.⁶²

Since linear alternators can already achieve quite large power conversion efficiencies, it is important to focus on reducing the costs. This is in accordance with the rest of the thermoacoustic engine, which can be produced in a relatively cheap manner and therewith compete with alternative technologies. As of now, linear alternators are the most expensive component in thermoacoustic engines, which is why designers often choose for cheaper commercial loudspeakers at the cost of a lower efficiency and possible power output. To reduce the price of alternators, it is important to provide clarity about the costs that are made for a given work, especially if the linear alternators are homemade. In this manner, designers can learn which aspects are the most expensive and try to reduce these in future work. A good example, albeit for a thermoacoustic engine with a loudspeaker instead of linear alternator, can be found in the work of Chen *et al.*²² They clearly show the costs of all components and the different stages in which they have reduced the costs of the thermoacoustic engine. Providing such information helps others to not only see how to reach a certain efficiency and power output, but also what the corresponding costs are per amount of power output.

2.2 Piezoelectric devices

This section will provide a review of piezoelectric devices for converting thermoacoustic power into electricity. Piezoelectric materials produce electricity during mechanical deformation, which in the field of thermoacoustics is caused by the incident acoustic wave. Sec. 2.2.1 provides some piezoelectric materials that have been successfully used in thermoacoustic engines to produce electricity and gives information on the possible configurations for these devices. Subsequently, Sec. 2.2.2 will focus on the characteristics of the

piezoelectric components, such as power output, efficiency and coupling with a thermoacoustic engine. Furthermore, analytical and numerical methods to study thermoacoustic engines with piezoelectric materials are presented. Note that the focus of this section about piezoelectric devices is on the application in thermoacoustic engines. For the basic principles and mathematical background for piezoelectric devices one can look at general literature such as APC's book on piezoelectrics.⁶³

2.2.1 Configurations

Piezoelectric materials, such as certain crystals and ceramics, build up a voltage difference across their opposite faces during mechanical deformation. If these sides are connected in an electrical circuit a current is induced, thereby producing electrical power. The reverse is also true, piezoelectric material will mechanically deform if it is connected in a circuit and a voltage difference is applied across it. The latter can be used in the field of thermoacoustics to produce acoustic waves, which can for example drive refrigerators. However, in this work we focus on the former, where the piezoelectric material is placed inside a thermoacoustic engine to convert the available acoustic power into electricity. An important side note is that one has to be careful with using piezoelectric material, because a large voltage potential can build up in an open electrical circuit, which can be very harmful if discharged upon human touch.

The most widely used piezoelectric material for thermoacoustic purposes is the ceramic lead zirconate titanate, also referred to as PZT.^{64–66} This is generally of a high quality and provides a good coupling,⁶⁴ but its brittle nature also limits the strain it can experience without being damaged.⁶⁷ Other materials used successfully in thermoacoustics include polyvinylidene fluoride, or PVDF, and piezoelectric fiber composite, or PFC.⁶⁴ These materials can be used for their increased flexibility and resistance to cracking compared with PZT.^{67,68} Furthermore, one investigation has focused on the use of lead magnesium niobate-lead titanate (PMN-PT) crystals.⁶⁹ A more elaborate review on piezoelectric materials can be found in the work of Anton and Sodano.⁶⁷ This includes the aforementioned materials amongst others, several tuning schemes, and different spatial configurations in which the material can be constructed and used.

The piezoelectric material is generally of a small mass with little inertia. It is therefore suitable for operating efficiently at a relatively high acoustic frequency and small wavelength.^{35,70} This results in small resonators and therewith compact thermoacoustic engine designs. However, due to the small inertia the power output can also be limited, as will be shown in Sec. 2.2.2a. Lin *et al.* have shown a weight can be added to increase the inertia and also tune the resonance frequency of the piezoelectric material.⁷¹ Nouh *et al.* also added a weight, but connected this with a spring to create a mass-

spring system they refer to as a dynamic magnifier.^{72–75} This component is placed between the acoustic resonator and the piezoelectric material in an attempt to enhance the strain experienced by the piezo-element for the same acoustic power. Note that they were inspired to use dynamic magnifiers by piezoelectric work from other fields of application.^{76–78}

So far, all presented thermoacoustic work involving piezoelectric material has been for standing wave engine designs. Very little work is done on traveling wave engines with piezoelectric harvesters. Furthermore, none of the found work is experimental, there is only analytical and numerical research.^{79,80} The traveling wave work uses the classical Backhaus and Swift engine design,^{61,81} with the piezoelectric element placed at the end of the resonator. For the standing wave engines, the following configurations are identified: a straightforward tube section,^{82,83} a Rijke tube based design,⁶⁶ a push-pull concept,⁶⁹ the most commonly used Helmholtz-like resonators,^{65,70,71,73} and a looped-tube configuration with 'wagon wheel' style piezoelectric alternators.^{84–86} Note that the latter looped-tube designs by Keolian *et al.* have traveling wave phasing in the regenerator but standing wave phasing at the piezoelectric element.⁸⁶ Furthermore, these designs are quite different in comparison with the other designs because they have a large amount of piezoelectric elements and can produce significantly more power (see Sec. 2.2.2a).

2.2.2 Characteristics

This section will provide details on the performance characteristics of piezoelectric power harvesters in thermoacoustic engines. Achieved efficiencies and power outputs will be presented in Sec. 2.2.2a, followed by information on the impedance and coupling of the piezoelectric materials in Sec. 2.2.2b. Furthermore, Sec. 2.2.2b also outlines several analytical and numerical methods that have been used to investigate piezoelectric systems in thermoacoustics. Experimental validation of these methods is provided where it is available.

2.2.2a Efficiency and electric power

An overview of achieved power outputs and efficiencies for piezoelectric harvesters in the field of thermoacoustics is given in Table 2.2. Note that all values are for standing wave devices, since no output characteristics were found for the work on traveling wave piezoelectric engines.^{79,80} The most important point about the available research is that nearly all of it is for very low electric power output, namely in the order of a few milliwatt. This is an inherent result of using only one or a few small piezoelectric diaphragms. When used in this configuration, the thermoacoustic engines are not useful in producing significant amounts of energy from (low-grade) heat streams, and

are therefore in a different field of application than the other energy harvesting methods reviewed in this chapter. These amounts of electric power could however still be used in small and self powered, maintenance-free devices where physical access and electrical wiring is difficult, such as thermoacoustically powered remote sensors.

The work by Jensen and Raspet approaches the field of significant power output with their simple thermoacoustic waste heat engine that produces 0.4 W of electric power. However, this is based on numerical work only, and should therefore still be proven experimentally and subsequently scaled to an electrical output about two orders of magnitude higher to be of significant use. From all the results presented in Table 2.2, the only work found in literature that is already in the order of significant power output is therefore that of Keolian *et al.* They developed a thermoacoustic piezoelectric engine (and refrigerator) for a project where the heat produced by a heavy-duty diesel truck is used as an energy input. Details about their design can be found in their final report⁸⁶ and patents.^{84,85} Numerical calculations have predicted their generator can produce 4 kW of electrical power.⁸⁶ So far, an early prototype has produced 37 W and they claim that they can modify this prototype to produce 600 W.^{86,87} This is still less than predicted for their final design, however, it is significantly more electrical power than any other work has produced using piezoelectric harvesters in thermoacoustic engines.

Keolian *et al.* predict using an analytical calculation that their piezoelectric harvester can convert acoustic power into electricity with an efficiency of 90 % or higher.^{85,86} However, unfortunately no values are found in their experimental work to confirm this calculation with such a high efficiency. As can be seen from Table 2.2, the other works investigated show a maximum efficiency of around 30 % numerically and 20 % experimentally. These efficiencies are still quite low, since research from other fields show that it should be possible to use piezoelectric energy harvesters in the range of 70 % to 90 %.⁸⁸

An improvement in the performance of piezoelectric devices has been made by Nough *et al.*^{72–75} by introducing dynamic magnifiers, as explained in Sec. 2.2.1. Table 2.2 lists the output power and efficiency for two numerical studies^{72,75} and one experimental work,⁷⁴ where either a dynamic magnifier is used or not. The piezoelectric material and the rest of the thermoacoustic set-up is kept the same for this comparison. For both numerical works the efficiency approximately doubles by using a dynamic magnifier. Furthermore, the electric power is also increased by a factor of two and six. Besides this increase in performance, the critical temperature gradient for self-sustained oscillations is also significantly decreased by using a dynamic magnifier.

Table 2.2: Overview compiled from literature of piezoelectric harvesters in the field of thermoacoustics. With P_{elec} the amount of electric power generated, η_{a2e} the acoustic to electric conversion efficiency, the abbreviation of the used materials as outlined in Sec. 2.2.1, and specifics of the piezoelectric material if available. All values are the maximum values reported in the papers. A notion of n/a means the data could not be found in the work and values denoted with an asterisk (*) are normalized and dimensionless and should therefore only be compared with each other.

Authors	Year ^{Ref}	Method	P_{elec} [mW]	η_{a2e} [%]	Material	Piezoelectric specs
Smoker <i>et al.</i>	2012 ⁷⁰	num/exp	0.1	10	PZT	Piezo Systems Inc. T107-A4E-573
Zhao	2013 ⁶⁶	num/exp	2	22	PZT	Piezo Systems Inc. T216-A4NO-573X
Wekin <i>et al.</i>	2008 ⁶⁴	exp	0.2	n/a	PZT,PVDF,PFC	only dimensions
Nouh <i>et al.</i>	2014 ⁷⁴	num/exp	0.001	0.2	PZT	normal, Digi-Key buzzer PZT-4
Nouh <i>et al.</i>	2014 ⁷⁴	num/exp	0.02	2	PZT	dynamically magnified, Digi-Key buzzer PZT-4
Jensen and Raspert	2010 ⁶⁹	num	400	n/a	PMN-PT	characteristics given
Matveev <i>et al.</i>	2007 ⁸³	num	n/a	10	n/a	n/a
Nouh <i>et al.</i>	2012 ⁷²	num	0.6 [-]*	15	n/a	normal, characteristics given
Nouh <i>et al.</i>	2012 ⁷²	num	1.3 [-]*	34	n/a	dynamically magnified, characteristics given
Nouh <i>et al.</i>	2014 ⁷⁵	num	0.5 [-]*	14	n/a	normal, characteristics given
Nouh <i>et al.</i>	2014 ⁷⁵	num	3 [-]*	22	n/a	dynamically magnified, characteristics given
Keolian <i>et al.</i>	2011 ⁸⁴⁻⁸⁶	num	4×10^6	90	PZT	Morgan Electro Ceramics PZT807
Keolian <i>et al.</i>	2011 ^{86,87}	exp	3.7×10^4	n/a	PZT	Morgan Electro Ceramics PZT807

For the experimental study, Nouh *et al.*⁷⁴ are aware that they use a piezoelectric material that is not as efficient as they have used in their previous experimental work by Smoker *et al.*⁷⁰ They still choose to work with this material since it is more practical for the purpose of choosing and optimizing several dynamic magnifiers. The efficiency and output power of their work is therefore quite low, as can be seen from Table 2.2. Nevertheless, using their best dynamic magnifier they increase their power output by a factor 20 and their efficiency by a factor 10 compared with the case where no dynamic magnifier is used. Finally, it is worth mentioning that, although not as straightforwardly visible as in the work of Nouh *et al.*, Keolian *et al.*^{85,86} also use diaphragms and mechanical springs that could be classified as dynamic magnifiers.

2.2.2b Calculations and coupling

To predict the performance of thermoacoustic engines with piezoelectric components, several analytical and numerical tools are available. Zhao has derived a nonlinear mathematical theory that shows a good agreement with experimental results on the acoustic velocity near the piezoelectric element.⁶⁶ However, his model does not include the performance characteristics of the piezoelectric element. Such governing equations are derived in several other works, where the coupling between the acoustic and piezoelectric part is done by impedance matching.^{69,75,83} These calculations give insight into the optimum efficiency and power output for a given engine design by varying the parameters of the piezoelectric transducer. Nouh *et al.* search for this optimum by applying weights to several performance objectives and solving their equations numerically.⁷⁵ As described in the previous section, the performance characteristics from the aforementioned calculations can be found in Table 2.2.

Aldraihem and Baz derive similar equations including the piezoelectric component, but subsequently use the root locus method from control theory to predict the onset of self-sustained oscillations for their traveling wave engine.⁷⁹ Nouh *et al.* later use this same theory and compare it with an electrical analogy approach that combines the descriptions of the acoustic resonator and the stack with the characteristics of the piezoelectric diaphragm.^{65,80} The analogous electrical circuit is numerically solved with the Simulation Program with Integrated Circuit Emphasis (SPICE) software.⁸⁹ They choose to use SPICE since it can be used for steady-state and transient analyses, in contrast with the widely used DeltaEC⁶⁰ that is limited to steady-state results. In their latest work, Nouh *et al.* compare the frequency of oscillation for the root locus method, DeltaEC, SPICE and experimental results and show quite a good agreement.⁶⁵ Furthermore, they show the tools can also predict the onset temperature gradient reasonably well. Note that the experimental power output and efficiency of the piezoelectric harvester for this specific

thermoacoustic engine are presented in more detail in the work by Smoker *et al.*⁷⁰ Besides the acoustic part of the engine, this research shows that the predicted piezoelectric power output and efficiency of DeltaEC are also in very good agreement with that of their experiments.

Inspired by the work of Smoker *et al.*,⁷⁰ a CFD model of their complete standing wave thermoacoustic engine is build by Scalo *et al.*⁹⁰ and further developed by Lin *et al.*⁷¹ They solve the fully compressible Navier-Stokes equations in the entire engine, where the piezoelectric harvester is modeled as a multi-oscillator time-domain impedance boundary condition. This high-fidelity simulation shows a consistent match with the experimental work of Smoker *et al.*, and provides an accurate tool for predicting the thermal-to-acoustic and acoustic-to-electric conversion efficiency.⁷¹

The aforementioned methods, from simple analytical calculations to complex numerical simulations, can all be used to predict aspects of the thermoacoustic engine performance. An essential property of the piezoelectric energy harvesters is the conversion efficiency of acoustic power to electricity. To achieve a high efficiency, it is important to ensure a good coupling between the piezoelectric material and the acoustic resonator. This can be achieved by matching the acoustic impedance of the resonator with the mechanical impedance of the piezoelectric harvester.⁷⁹ Note that, as described in more detail in Sec. 2.1.2b, acoustic impedance is the ratio of the pressure amplitude over the flow rate of the working gas. The mechanical impedance is similarly the force applied to the piezoelectric material over the induced velocity of the material. In practice, one can thus ensure a good coupling by altering the acoustic impedance through a change in operating conditions or dimensions of the thermoacoustic engine and/or by tuning the piezoelectric harvester. The latter can be done by e.g. varying the piezoelectric material and its configuration,⁶⁴ adding a weight to the harvester to tune its resonance frequency,⁷¹ or using a dynamic magnifier.^{72,74,75}

2.3 Magnetohydrodynamic devices

This section will provide a review of magnetohydrodynamic (MHD) devices for converting thermoacoustic power into electricity. The MHD transducers use an electrically conducting fluid that is forced to oscillate by the incident acoustic wave of the thermoacoustic engine. By applying a magnetic field over the oscillating working fluid, electricity can be produced in either an inductive or conductive manner. Since the working fluid is used to convert the acoustic power directly into electricity, there are no mechanically moving parts, just as in the rest of the thermoacoustic engine. Therefore, thermoacoustic engines with MHD generators are well suited for applications where no maintenance is possible, such as in outer space.^{91,92}

The working principle and details of both the inductive and conductive

Figure 2.5: Schematic representation of an inductive magnetohydrodynamic transducer driven by an acoustic wave.

MHD transducers will be treated in Sec. 2.3.1, along with different working fluids that have been used in thermoacoustics. Subsequently, Sec. 2.3.2 will elaborate on the performance of the MHD generators and provide several methods for analyzing and optimizing this performance. This review is focused on the field of thermoacoustics, just as the review work of Hamann and Gerbeth from 1993.⁹³ For a more general introduction and mathematical background of magnetohydrodynamic devices one can look at general literature, such as the books by Rosa⁹⁴ and Moreau.⁹⁵

2.3.1 Configurations

The MHD generators can be divided in two main configurations: inductive and conductive. Fig. 2.5 schematically depicts an inductive MHD transducer. The permanent magnet supplies a magnetic field through the ferromagnetic yoke, with the magnetic field lines in the vertical direction of the schematic. The working fluid is driven by the sound wave to oscillate horizontally, and therewith perpendicular to the applied magnetic field. This will induce an alternating current in the fluid, which is accompanied by a pulsating magnetic field. The oscillatory motion of the fluid therewith induces an alternating current in the surrounding coil, which is placed in the direction perpendicular to both the permanent magnetic field and the oscillating fluid. In this manner, the inductive MHD generator creates an alternating electric current with adjustable strength and voltage.^{96,97}

The conductive MHD generators work similar to the inductive devices depicted in Fig. 2.5. The main difference is that there is no coil, but a pair of electrodes that directly collects the alternating current from the working fluid. The electrode pair is placed perpendicular to both the magnetic field lines and the oscillating working fluid, which is in and out of the plane of Fig. 2.5. The conductive MHD generators produce a strong electric current at a low voltage,^{96,97} which is generally not preferred over the adjustable strength and voltage of the inductive MHD transducers.⁹⁸ Furthermore, at high mean pressures of the working fluid, sealing the electrode connections from leaks might be troublesome.⁹⁷ These disadvantages have caused the focus in thermoacoustics to shift from the conductive devices of

the early days^{93,99,100} towards inductive MHD generators in most of the recent work.^{92,96,97}

Independent of the configuration, the electrically conducting working fluid mostly used is liquid sodium.^{91,98,99} This is a suitable working fluid due to its relatively low density and high electrical conductivity, which makes it sensitive to the influence of a magnetic field.⁹⁷ Similarly, a saturated sodium bicarbonate aqueous solution is experimentally shown to be suitable as well.¹⁰¹ Although it shows a good performance, one has to be careful with sodium since it is a highly reactive and possibly hazardous material. Swift *et al.* have reported serious damage to their thermoacoustic engine due to a violent chemical reaction.⁹⁹ Besides sodium, any other electrically conducting fluid can be used, such as other liquid metals, salts and even plasma. For plasma, one can use a gas just as in the rest of a conventional thermoacoustic engine, and locally induce a plasma by periodically discharging a high voltage by electrodes immersed in the gas.⁹²

Swift *et al.* have developed one of the first thermoacoustic engine prototypes with an MHD generator, as presented in a series of studies^{99,102,103} and a patent application.¹⁰⁰ Their engine, including the prime mover generating the acoustic power, is completely filled with liquid sodium at a pressure of 200 bar. Although the engine has shown quite a good performance (see Sec. 2.3.2), the high pressure in the entire engine can cause problems in sealing the electrodes⁹⁷ and generating the acoustic oscillations.¹⁰¹ As an alternative, one could use a gas for the acoustic part of the engine and an electrically conducting fluid for the MHD generator.^{91,97,104} Separating the two fluids can for example be done by gravity using a U-shaped tube.¹⁰¹ A downside of such designs is that there is a gas-fluid interface, as shown in Fig. 2.5. Transmitting all of the acoustic power across the interface can be hard due to the large impedance difference¹⁰⁴ and the possibility of a Rayleigh-Taylor instability,¹⁰⁵ which can occur due to the gas with a smaller density periodically forcing the heavier fluid. For oscillating flows of sufficient amplitude, this instability of the gas-fluid interface has been observed in a magnetohydrodynamic fluid pump.¹⁰⁶ The operating frequency of the two-fluid thermoacoustic engines is mostly around 100 Hz or lower,^{97,104} while that of the fully liquid sodium engines is in the order of 1000 Hz.^{99,102}

Since the design of the MHD generators is symmetric, it can be driven from either side. To increase the power output and reduce the necessary drive ratio in the acoustic part, several engine designs have been proposed that use this symmetry to drive the MHD generator from both sides in a so called push-pull mode.^{91,97,100,104} As will be treated in more detail in the following section, these designs can improve the performance and relax the operating conditions of the thermoacoustic engine, but also result in a more complex coupling of the acoustic gas with the MHD working fluid.

2.3.2 Characteristics

An important operating characteristic for MHD devices is the magnetic Reynolds number, where the kinematic viscosity of the ordinary Reynolds number is replaced with the magnetic diffusivity.⁹⁹ The magnetic Reynolds number is therefore a measure of the advection (or induction) of a magnetic field relative to the magnetic diffusion. In a series of work by Alemany *et al.*, they do calculations for low magnetic Reynolds numbers, which allows them to neglect the induced field compared to the applied one for the MHD part.⁹⁷ For the thermoacoustic part, they use a linearized version of the Navier-Stokes equations that is based on the work of Swift *et al.*¹⁰⁷ The resulting set of equations have provided them with a tool to analyze the performance of thermoacoustic engines with both conductive⁹⁸ and inductive^{96,97} MHD generators. For the conductive MHD generator they predict a maximum acoustic to electric conversion efficiency of 80 %, ⁹⁸ although they later state that this was not for realistic load conditions.⁹⁷ Subsequent calculations for an inductive MHD generator show a theoretical efficiency of 65 % and an electric power output of 500 W.⁹⁷ Further work on the inductive generator focuses on the relation between the acquired efficiency and the magnetic Reynolds number.⁹⁶ Through the working principle of an MHD generator, they explain the influence of the magnetic Reynolds number and show efficiencies in the range of 60 % to 72 %.

The previous paragraph has given promising performance calculations for MHD generators in thermoacoustics, but unfortunately there is not much experimental work to confirm these calculations. One paper presents an experimental prototype of a conductive MHD generator that only produces a few millivolts.¹⁰¹ No load is applied to measure the power output due to this small voltage, but a theoretical prediction based on the book by Rosa⁹⁴ does show a good correspondence with the voltage output. The only experimental prototype with significant power in thermoacoustics is found in a series of work by Swift *et al.*^{99,100,102,103} As described in Sec. 2.3.1, they developed a thermoacoustic engine with a conductive MHD generator that is completely filled with liquid sodium at 200 bar. The highest acoustic to electric conversion efficiency they achieve in their experiments is 45 % and they produce 300 W of electric power.⁹⁹ They compare their results with calculations based on the linearized Navier-Stokes and Maxwell equations, and show a good correspondence for the conversion efficiency, with the experimental values being even slightly higher than their calculations predict. This good correspondence strengthens their claim that, with modest changes of their design that are based on analytical calculations, they expect to reach an efficiency of around 70 %.⁹⁹ Inspired by the work of Swift *et al.*, an analytical optimization study was performed for an MHD generator with similar physical conditions.¹⁰⁸ Characterized by four dimensionless parameters, the isotropic electrical efficiency and the overall second law efficiency, based on

the entropy generation rate, were optimized for maximum performance. Besides the use for this single case, the entropy generation rate can be a useful tool for modeling and optimizing MHD generators in general.¹⁰⁸

For an optimum performance of an MHD generator in a thermoacoustic engine, it is important to make sure there is a good coupling with the acoustic circuit. In the case of a combined gas-fluid engine, one has to ensure a close match of the natural frequency of the acoustic prime mover with the liquid MHD column for a good power transmission.¹⁰¹ However, this can be hard due to the large density, and therewith impedance difference between the gas and liquid. De Blok *et al.* use the impedance difference to calculate that for a typical engine and operating conditions 90 % of the acoustic power is expected to be reflected back at the gas-fluid interface.¹⁰⁴ To transfer significant acoustic power to the high impedance region of the MHD generator, their thermoacoustic drive ratio has to be 15 %, which will result in serious acoustic losses. Therefore, they propose to drive the MHD generator from two sides in a push-pull configuration to reduce the drive ratio (i.e. required pressure amplitude) by a factor 2. Alemany *et al.* also use a push-pull configuration and use the iterative Newton's method to find the optimal conditions of their engine by minimizing the calculated impedance difference.⁹⁷ The latter can result in a good acoustic power transmission, but one has to ensure that there remains a 180 degree phase difference between the two thermoacoustic parts driving the MHD generator. This can be a problem since the two prime movers tend to oscillate in phase.¹⁰⁹ De Blok *et al.* explain how to acoustically ensure that the phase difference remains intact and propose a two-stage thermoacoustic engine design that numerically shows a good performance.¹⁰⁴

2.4 Bidirectional turbines

A bidirectional turbine is a special type of turbine that rotates in the same direction, independent of the direction of the axial flow. It is therefore suited to convert acoustic waves into rotational work, and subsequently into electricity when connected to a generator. Due to the relatively new introduction of bidirectional turbines in the field of thermoacoustics, the available literature is scarce. Nevertheless, bidirectional turbines have potential because they are relatively cheap, can be coupled with highly efficient off-the-shelf generators and are promising for scaling to industrial sizes. Therefore, bidirectional turbines are still included in this review, but this section will be somewhat different than the others. A substantial amount of information will be given from the field of Oscillating Water Columns (OWC), where bidirectional turbines are used to convert marine wave energy into electricity. The focus will be on information from OWC literature that can be used to successfully implement bidirectional turbines in the field of thermoacoustics. In Sec. 2.4.2,

Figure 2.6: Schematic representation of an axial impulse turbine and the rotor blade of a Wells turbine, shown on the left and right hand side of the dashed line, respectively.

several bidirectional turbine designs with their most important features will be introduced, along with their advantages and disadvantages. Sec. 2.4.2 will start with performance results from the field of thermoacoustics, after which some important characteristics from the field of OWCs are given.

2.4.1 Configurations

Bidirectional turbines, which can also be referred to as self-rectifying turbines, can be split into impulse and reaction based configurations. Fig. 2.6 depicts an axial impulse turbine design, where the two rows with static guide vanes convert the acoustic pressure energy to kinetic energy and direct this flow to the rotor blades. Due to the symmetric design, the rotor will experience a torque in the same rotational direction, independent of the incoming axial flow. The mechanical work of the rotor can then be converted to electricity by connecting it with a generator that can be placed inside the nose cone. This nose cone is generally used to guide the incoming flow towards the blades, where the latter do not span the whole radius but run from the central part called the hub towards the tip (see R_{hub} and R_{tip} in Fig. 2.6). Note that the guide vanes are usually kept in place with a ring around the blades that is connected to the outer tube, but this is not depicted in Fig. 2.6 to show the shape of the blades.

A similar, but less widely used configuration is the radial impulse turbine. The working principle of this turbine is the same as the axial version, but here the rotor is placed between the two guide vanes in radial direction, where one guide vane directs the flow outward towards increasing radius and the other inward towards the central axis. The kinetic energy of the bidirectional flow is again converted to rotational work by a similarly shaped impulse rotor. An advantage of the radial impulse turbine over the axial

version is the reduced amount of axial thrust, which relaxes the fatigue on the bearings of the generator.¹¹⁰ Besides this difference, both impulse turbines are characterized by a wide operating range with efficient performance and good starting characteristics, where the latter is ensured by the use of guide vanes. However, the large incidence angle of the flow at the downstream guide vane also causes an increase in aerodynamic loss, which can not be avoided due to the symmetry needed for operating in bidirectional flows.¹¹¹

In contrast with the impulse turbines that use the kinetic energy of the flow, reaction based turbines use the pressure oscillations of the acoustic work to produce lift and therewith torque on a rotor blade. The most widely used reaction based turbine is the Wells turbine, of which the rotor blades have the shape of symmetrical airfoils, as can be seen at the right hand side of Fig. 2.6. The shape of the airfoil will cause a component of lift in the direction from the thin trailing edge to the blunt leading edge, therewith producing a torque in the rotational direction. Several airfoil shapes have been shown to work for bidirectional flows, including the most widely used NACA0015 through NACA0025 shapes.^{112,113} The Wells turbine can be used in combination with guide vanes, but these are often omitted since they are not necessary for the turbine to work (in contrast with the impulse turbines). Furthermore, the Wells turbine generally has more simple and fewer blades than the impulse turbines. These points make the Wells turbine the most simple and economic bidirectional turbine to convert acoustic power into electricity.¹¹⁴ However, there are several downsides during operation of the Wells turbine, such as the poor starting characteristics and the narrow range of efficient operation,^{114,115} as will be discussed in more detail in Sec. 2.4.2.

Next to the distinction in turbine configuration, several design aspects are also important for bidirectional turbines in thermoacoustic engines. A wide range of nose cone designs is available for conventional turbines, of which hemispherical, parabolic and elliptical shapes have been used successfully in thermoacoustics.¹¹² Another important aspect is the radial spacing between the edge of the rotor blades and the outer tube, called the tip clearance (see Fig. 2.6). Minimizing this distance is very important to ensure that the acoustic power is transferred to the turbine blades and does not leak along the outer edge. This importance is illustrated in bidirectional flows for both impulse¹¹⁶ and Wells¹¹⁷ turbines. In case guide vanes are used, another important spacing is the axial distance between the rotor and guide vanes, which should generally be small to reduce the leakage in radial direction and therewith achieve an acceptable efficiency. Furthermore, the ratio of the number of guide vane blades over rotor blades is important. For an efficient energy transfer and no spurious vibrations, there should be a minimal amount of symmetry planes between the guide vane and rotor. This means that if the rotor has an even number of blades (e.g. for a good balance), then it is optimal to choose a prime (or coprime) number of blades for the guide vanes. Finally, note that besides the important aspects mentioned in

this paragraph, there are several other design features that can be important for the turbine performance, such as blade solidity and the hub-to-tip ratio. An overview of the optimum design values for a wide range of aspects is given in the work of Das *et al.* for Wells and impulse turbines in OWCs.¹¹⁰

Just as in conventional steady flow turbines, the Wells and impulse turbines can be extended with additional rotors and guide vanes to introduce multiple stages.¹¹³ This can be done in an effort to enhance the coupling between the turbine and working gas and to increase the power output. For the guide vanes, an option used for OWCs is (self-)pitching guide vanes that pivot around a point to increase the efficiency of the turbine.^{111,115} However, due to the relatively high acoustic frequency and the low-maintenance requirements for the rest of the thermoacoustic engine, it is encouraged to stick with fixed guide vanes for simplicity and reliability. With these fixed designs and the low material requirements due to the ambient temperature, the turbines can be easily manufactured with the cheap process of 3D rapid prototyping. Besides initial hardware costs that are currently dropping rapidly, the material costs of 3D printing a turbine is in the order of only ten dollars per dm^3 of material. When combined with the mature technology of conventional rotary generators, where the costs are approximately a hundred dollars per kW of electric power output, the acoustic to electric conversion using bidirectional turbines can be relatively cheap and results in a reliable design.

2.4.2 Characteristics

The use of bidirectional turbines in thermoacoustic engines is fairly new compared with the other conversion technologies, as shown by the fact that the first published research is from 2014.⁶² In this work, de Blok *et al.* experimentally investigate the use of a 3D printed bidirectional impulse turbine with varying acoustic power up to 30 W. Besides the amplitude, they also varied the acoustic frequency in the range of 20 Hz to 50 Hz. Interestingly, they found no significant influence of the operating frequency on the turbine efficiency, suggesting that it is unnecessary to match the frequency of the acoustic circuit with the turbine for efficient power transduction in the engine. For air at ambient pressure, they measured the acoustic to electric efficiency to range between 25 % and 30 % for varying acoustic power. These values are in line with the performance acquired for impulse turbines in OWCs that operate at ambient pressure.^{110,118} However, thermoacoustic engines generally work at higher mean pressures, and thus larger mean fluid densities, since these are proportional for ideal gases. As measured by de Blok *et al.*,⁶² a larger mean density will significantly increase the efficiency of the impulse turbine, as shown in Fig. 2.7. With a typical thermoacoustic engine operating at 40 bar mean pressure (with density for air $\sim 48 \text{ kg m}^{-3}$), they show that a rotor efficiency of 85 % can be acquired. When coupled with commercial generators that are highly efficient, converting acoustic power to

Figure 2.7: Bidirectional impulse turbine efficiency for various fluid densities as experimentally measured by de Blok *et al.*⁶²

electricity at an efficiency of 80 % thus seems feasible. Furthermore, de Blok *et al.* have shown these results for larger bidirectional turbines as well and see no limitations in scaling their systems to power levels in the MW range.⁶²

At present, two other works are available that investigate the use of bidirectional turbines for thermoacoustic engines. Kaneuchi and Nishimura have presented an extended abstract about an impulse turbine similar to that by de Blok *et al.*, and find a rotor efficiency of around 20 %, which results in 12 % acoustic to electric conversion due to their inefficient generator.¹¹⁹ Boesneck and Salem have investigated both Wells and axial impulse turbines with a wide variety of design changes, including different guide vanes and nose cones.¹¹² Although they envision using their designs in thermoacoustic engines, unfortunately they have only performed steady flow measurements for their current research, which makes their investigation similar to OWC literature. Nevertheless, the extensive amount of designs have still resulted in some interesting bidirectional turbine characteristics, of which the most relevant are given next. For the Wells turbines they have done measurements with the NACA 0015, 0018 and 0021 rotor blades, and found no significant difference between the three. The optimum number of rotor blades is identified as five, and they find mixed results when using guide vanes for their Wells turbine, with a slight increase in performance possible when compared to the case without guide vanes. For the axial impulse turbine, they find that for their guide vanes a 12° inclination angle works better than a 30° one. Varying the blade angle for the rotor blades between 50° and 70° has not resulted in any significant performance change. The best nose cone is of an elliptical shape, and the rapid prototyping technique that produces the smoothest surface has resulted in the best performance. Finally, when comparing the Wells turbine with the axial impulse turbine, they conclude that the Wells turbine has a small range with superior performance, but the

impulse turbine has much better starting characteristics and shows a more stable performance over a wide range of operating conditions.¹¹²

In the field of OWCs, the poorer starting characteristics of the Wells turbine have also been widely shown.^{110,114,115,120} Besides the starting of the turbine at small flow rates, the angle of stall also bounds the operating range of the Wells turbine for high flow rates.¹²⁰ Furthermore, they generally run at a high rotational speed, which is accompanied by unwanted noise and a high axial thrust.^{111,115} The impulse turbines have good starting characteristics and no problems of stall at higher flow rates, and therefore have the advantage of a wider operating range where a change in Reynolds number does not have much influence on the turbine efficiency.¹¹⁰ However, this does come at the cost of a smaller peak efficiency than the Wells turbines, with the radial impulse turbine generally having a smaller efficiency than the axial version.¹¹⁰ Finally, as suggested by de Blok *et al.* for thermoacoustics,⁶² the possibility of scaling bidirectional turbines to the MW range seems plausible. There are several examples of OWCs that have delivered more than 100 kW of power,¹¹⁰ with a maximum reported power output of 500 kW.¹²¹

To predict the performance of the turbines, analytical calculations expressed in torque, input power, and flow coefficients can be useful for initial estimations.^{113,122} Furthermore, Thakker and Dnanasekaran have shown a good correspondence between a CFD model and experimental results in their study to show the large influence of the tip clearance on the turbine efficiency.¹¹⁶ However, one should be careful with using these methods blindly for thermoacoustics because they assume a quasi-steady flow. The frequency of the wave for OWCs is much smaller than the rotational frequency of the bidirectional turbine, while for thermoacoustic applications they can be of the same order. Therefore, this might mean that the calculations are sufficiently accurate for OWCs but not for thermoacoustic engines. On the other hand, in contrast with OWCs, there is a single wave frequency and amplitude in thermoacoustics. This could be used to simplify the equations for irregular bidirectional flows and one can envision turbine design optimizations that are not possible for OWCs. Therefore, the work from OWCs can be used as a starting point, but more specific research about bidirectional turbines under thermoacoustic conditions is needed to utilize its full potential.

2.5 Conclusions and recommendations

This chapter has given a review of the different methods to convert thermoacoustic power into electricity. For each technology, design aspects, operating characteristics, and methods to calculate and optimize the performance have been treated individually. This section will provide general conclusions and recommendations, which are based on a personal view and the following most important findings of the individual sections:

- Electromagnetic transducers are the most mature conversion technology in the field, and have experimentally shown the highest power output ($\sim$ kW) and efficiency ($\sim$ 75 %). Dedicated linear alternators show the best performance and reliability, although they can be relatively expensive. Conventional loudspeakers are useful for cheap, initial prototypes, but their performance is limited and they are too fragile for long time operation.
- Piezoelectric devices provide the possibility of compact thermoacoustic engines due to their ability to operate at relatively high frequencies. The power output is generally very small ($\sim$ mW), with the exception of one series of work, and the maximum efficiency of $\sim$ 20 % in thermoacoustics is not nearly as high as the $\sim$ 70 % acquired in other fields. Using dynamic magnifiers coupled with the piezoelectric material has shown to increase the output performance.
- Magnetohydrodynamic devices have no mechanically moving parts, which makes them very reliable and suitable for inaccessible environments. The numerically predicted performance is fairly good, but there is a lack of experimental prototypes to confirm this in practice. One serious obstacle to build devices with an efficient operation is the power transduction across the gas-liquid interface.
- Bidirectional turbines were recently introduced as potentially cheap and reliable transducers with a good performance. The efficiency in thermoacoustics can reach $\sim$ 80 % and work from the field of oscillating water columns shows that power outputs in the MW range are feasible. Initial work shows an efficient performance that is nearly independent of the acoustic frequency, but more research is necessary to successfully implement bidirectional turbines in thermoacoustic engines.

For any acoustic to electric conversion technology, it has been made clear that assuring a good power coupling between the acoustic field and the transducer is key for a good performance. An impedance mismatch will cause acoustic reflections, which results in redirected acoustic power that is not converted into electricity. Several analytical and numerical methods have been shown to be useful in ensuring a good coupling and optimizing the performance of a thermoacoustic engine, with DeltaEC being the most prominent option. The basis for the acoustic to electric transducer implementation should be based on such calculations, while experimental prototypes can be used for further optimization.

The individual conclusions show that piezoelectric transducers are mainly useful for designing compact thermoacoustic engines, where only a small power output is needed. One should focus on stacking multiple piezoelectric diaphragms in clever designs to increase the efficiency and power output. Furthermore, to make these transducers a viable option for producing

significant amounts of electricity in thermoacoustic engines, developments in the field of piezoelectric materials should lead to larger power densities and the possibility to create more bulky piezoelectric materials. As of now, a linear alternator is the most viable option to directly implement and efficiently convert acoustic power up to the kW range. For increasing power outputs, scaling the linear alternators becomes increasingly challenging due to the large moving mass and the difficulty to maintain clearance seals over large displacement amplitudes. A promising alternative that shows no limitations in scaling the output power is the bidirectional turbine. There are good indications that its conversion efficiency can compete with the linear alternator, while the costs can be lower when relying on the mature technology of conventional rotary generators. Nevertheless, more work on bidirectional turbines needs to be done to successfully implement them and confirm that they can indeed have such a good performance.

The conclusions from this review form the main motivation for an in-depth study of bidirectional turbines in thermoacoustic conditions, as is presented in this thesis. The focus will be on axial impulse turbines, since these are easy to implement, self-starting, and can be tested in a wide range of operating conditions.

3

Experimental procedure

This chapter will provide details about the experimental procedure that is used to investigate the performance of bidirectional impulse turbines for varying thermoacoustic conditions. The experimental set-up is presented in Sec. 3.1, which is used for all the measurements under lab conditions in Chapter 4 and Chapter 5. A description of the measurements with the thermoacoustic refrigerator, in which the turbine is implemented in Chapter 6, is given in Sec. 6.2.

After the description of the experimental set-up, more specific details about the turbine design are given in Sec. 3.2. The way in which the measured data is used to calculate the performance of the turbine is described in Sec. 3.3, while the calibration of the hardware is treated in Sec. 3.4. Finally, an extensive validation and error estimation of the experimental procedure is given in Sec. 3.5.

3.1 Experimental set-up

The experimental set-up is shown schematically in Fig. 3.1. This set-up, previously used by Aben¹²³ and Oosterhuis⁸, has been adjusted for the current work to incorporate the electrical load to the turbine's generator and the measurement thereof. On the right hand side, a 15 inch loudspeaker (JBL W15GTi) is enclosed in a cylindrical back volume. This section is structurally decoupled from the rest of the set-up by a membrane. In its current state the system can not be pressurized, thus all measurements are done with air

This chapter is adapted from: M. A. G. Timmer and T. H. van der Meer, "Characterization of bidirectional impulse turbines for thermoacoustic engines," *J. Acoust. Soc. Am.* **146**(5), 3524-3535 (2019). doi:10.1121/1.5134450

Figure 3.1: Schematic of the experimental set-up (not to scale). The pressure sensor locations (P1-P4) and tube dimensions are given in mm. A detailed schematic of the turbine section (depicted by the dashed lines) is given in Fig. 3.2.

at ambient pressure. A 2 kW audio amplifier (Behringer EP2000) drives the loudspeaker with an audio signal from a computer sound card. The acoustic wave travels through an exponentially contracting horn to the straight tube section where turbines can be mounted and tested. To acquire the largest acoustic amplitudes with the given set-up, a closed end is used on the left, resulting in pressure amplitudes of up to 7500 Pa. The latter is measured by four piezoresistive differential pressure sensors (Honeywell 26PCAFA6D) mounted flush with the tube wall. As shown by P1-P4 in Fig. 3.1, there are two sensors on either side of the turbine to be able to calculate the local acoustic power. More details on these calculations is given in Sec. 3.3 and the calibration of the sensors is presented in Sec. 3.4.

The 60 mm diameter test section is modular in length, but kept at 1300 mm in this work, such that there is sufficient acoustic power for frequencies in the range 50-90 Hz. To perform measurements for substantially lower or higher acoustic frequencies, the test section has to be lengthened or shortened, respectively. The turbine and generator combination is mounted in a 100 mm tube section (shown by the dashed lines in Fig. 3.1). The mid-plane of the turbine is at 850 mm from the closed end, where the acoustic velocity amplitude is large enough for significant turbine performance for all measurement frequencies. A Hacker A10-13L brushless electric motor is connected to the rotor and used as a generator. The induced three-phase alternating current of the generator is connected to three 20 W precision resistors in delta configuration, as denoted by the electrical load in Fig. 3.1. Different sets of $R = 4.7 \text{ Ohm}$, 10 Ohm , and 20 Ohm resistors are used, to be able to have a variable rotational velocity of the turbine for the same acoustic input. The calibration of the generator efficiency for all electrical loads is described in Sec. 3.4. Note that extra clamps have been placed on the generator axis to prevent axial movement of the rotor and axis. High speed imaging has shown that after placing these clamps no significant axial movement is

Figure 3.2: Schematic of an axial bidirectional impulse turbine mounted in a tube section. The guide vanes are connected to the outer tube by a shroud ring (only depicted for the right guide vane). The generator is mounted on the left guide vane and connected to the rotor. During operation, the generator is enclosed by the nose cone, which also ensures the axial flow is redirected towards the guide vane blades.

left. The latter is especially important for measurements where the spacing between the guide vanes and rotor is minimized (see Sec. 5.4).

During each experiment, the four pressure signals, the power to the loudspeaker, and the voltage over one resistor of the electrical load are measured with an NI-6250 data acquisition system. This is done by sampling at 20 kHz for a duration of twenty seconds per loudspeaker setpoint, of which the first ten seconds are discarded to allow the turbine to reach a constant operating speed. The remaining raw data is digitally phase-locked and processed, after which the results are stored in the TDMS file format for further processing. The TDMS files, further processing scripts, and final results can be found in the data publications for the measurements presented in Chapter 4 and Chapter 5.^{124,125}

3.2 Turbine design

For the conversion of acoustic power into electricity, an axial impulse turbine is used which is based on the work from Suzuki *et al.* for an OWC.¹²² A schematic of this turbine mounted in a tube section is shown in Fig. 3.2. This turbine can convert acoustic power into shaft power by redirecting the bidirectional axial flow towards the rotor blades with two sets of guide vanes. At the rotor blades, the flow induces a torque, causing a rotation in the direction given in Fig. 3.3. The resulting shaft power is converted into electricity by connecting a generator to the rotor.

The turbine has 29 rotor blades with 26 guide vanes on either side of

Figure 3.3: Schematic of the bidirectional impulse turbine (to scale). The dimensions are given in mm unless stated otherwise, and provided at the mean turbine radius of 24.8 mm. The dimensions in black are exact (not rounded) and completely determine the geometry. The values in gray follow from the defined geometry and turbine properties, such as number of blades. Note that in the zoomed section the $\sphericalangle$ symbol refers to lines being tangent at the corresponding points ($\bullet$) at the given angles. These 50° and 70° angles ensure an effective angle of 60° between the centerline of the rotor blade and the axial flow direction.

it. Instead of the 30 blades by Suzuki *et al.*,¹²² it is chosen to use a prime number of rotor blades to avoid any planes of symmetry. The guide vanes are connected to the outer tube by a shroud ring (only depicted for the right guide vane). The rotor is shown without a shroud ring, but turbines later in this work also feature such a ring to investigate the performance thereof (see Sec. 5.2). The tip clearance is the radial spacing between the tip of the rotor and the outer tube, as shown in Fig. 3.2. The turbine used for characterization in Chapter 4 has a tip clearance of 1 mm. In Sec. 5.3, the tip clearance is varied to quantify its influence on the turbine performance. The hub-to-tip ratio is 0.7 for the smallest tip radius of 28.0 mm, and slightly reduces for increasing tip radius.

To guide the axial flow towards the blades, nose cones are connected to both guide vanes. The cones have an axial length of 40 mm, consisting of a 30 mm elliptic part running up to the hub radius, followed by a 10 mm straight part. Fig. 3.2 depicts the nose cone in an exploded view to show the generator that is connected to the left guide vane. In practice, the nose cone is connected to the guide vane and covers the generator, leaving only room for the electrical wires to pass towards the outer tube and electrical load. The turbine and nose cones are 3D printed from PA 2200 plastic using Selective Laser Sintering. The accuracy and influence of the 3D printing technique on the turbine performance is presented in Sec. 3.5.

The exact shape of the guide vanes and rotor blades is shown to scale in Fig. 3.3. The design is presented in such a way that the given dimensions and relations exactly determine the geometry and a design study for the length, thickness and angle of the rotor blade can easily be carried out. The results of such a design study will be presented in Sec. 5.5. With a fixed hub radius of 19.6 mm and tube radius of 30 mm, the given geometry from literature¹²² was scaled to the mean radius of 24.8 mm. The axial spacing between the guide vanes and rotor blades, depicted by X_s in Fig. 3.3, is nominally set at 1 mm. In Sec. 5.4 this spacing is varied to investigate the influence of it on the turbine performance.

3.3 Performance calculations

To measure the input power absorbed by the turbine, the acoustic power before and after the turbine has to be known. Acoustic power is defined as:

$$E_2 = \frac{1}{2} |p_1| |U_1| \cos(\phi_{pU}), \quad (3.1)$$

with p_1 the complex pressure amplitude, U_1 the complex flow rate and ϕ_{pU} the phase angle between p_1 and U_1 . Since acoustic power is a difficult property to directly measure, it is approximated with sets of two pressure sensors using the following procedure. The atmospheric pressure and ambient temperature are measured and used to calculate the current gas properties of the air. The complex amplitudes of the pressure signals are determined by performing a discrete Fourier transform at the driving frequency and adding the phase information. The complex pressure amplitudes are subsequently used to determine the complex velocity amplitude and the acoustic power using Eq. 12 and Eq. 23 of Fusco *et al.*,¹²⁶ respectively. These calculations include the effect of viscosity for accurately measuring at the tube wall and are valid for an arbitrary distance. Note that the calculated values are located in the middle of both pressure sensor sets P1-P2 (left of the turbine) and P3-P4 (right of the turbine). This is extended to the complete complex pressure and velocity fields on both sides of the turbine by solving Eq. 4.54 and Eq. 4.70 from Swift with the boundary layer approximation.¹⁰ The acoustic power dissipation in the tube between the turbine and sensor location is then corrected for using Eq. 5.11 of Swift,¹⁰ resulting in the acoustic power absorbed solely by the turbine as follows:

$$\Delta E_2 = E_{2, \text{right}} - E_{2, \text{left}} - E_{2, \text{correction}}. \quad (3.2)$$

To determine the shaft power produced by the turbine, it is coupled to a generator and electric load. By measuring the RMS voltage, V_{rms} , over one of the three resistors, the total electric power is given by: $P_e = 3 \cdot V_{rms}^2 / R$. Since this work is concerned with purely the turbine performance, the influence of

the generator is excluded by dividing the electric power with the generator efficiency. This yields the turbine shaft power as follows:

$$P_m = \frac{P_e}{\eta_{gen}}, \quad (3.3)$$

where the generator efficiency, η_{gen} , is calibrated as described in Sec. 3.4. The rotational speed of the turbine is also determined from the voltage over the resistor by dividing the measured frequency by the 6 pole pairs of the generator. The efficiency of the turbine is determined by regarding the shaft power as output and the acoustic power difference as input, *i.e.*:

$$\eta_t = \frac{P_m}{\Delta E_2}. \quad (3.4)$$

During processing there are automatic checks to exclude any data that is out of the calibrated range or where the generator electric power is not the same as during calibration (probably due to faulty connections). More insight into the exact processing of the experimental data can be found by examining the scripts in the data publications.^{124,125}

3.4 Calibration

The four piezoresistive pressure sensors used in this work are calibrated with respect to a pre-calibrated Kulite XTE-190M pressure sensor. For this purpose, all five sensors are mounted in the same (axial) plane at the left end of the tube section (see Fig. 3.1). A dynamic calibration is carried out by sweeping the acoustic frequency from 40 Hz to 100 Hz and the pressure amplitude from 200 Pa to 7500 Pa. Using the reference sensor, all four sensors are calibrated for their sensitivity and their phase difference. The typical sensitivity is 0.5 mV/Pa with a standard deviation smaller than 0.3 μ V/Pa for the entire calibration range. The phase difference is calibrated for each frequency separately, with a maximum mutual difference of 0.1° over the entire calibration range. More details on the linearity, stability and repeatability of the sensors and its calibration can be found in appendix B of the work by Oosterhuis.⁸

To calculate the shaft power of the turbine, as given by Eq. 3.3, the efficiency of the generator must be calibrated for all operating conditions. This is done by coupling the AC generator to a Maxon A-max 26 DC motor of which the torque constant has been determined to be 13.5 mNm/A. By driving the DC motor and measuring the input current, the torque supplied to the generator is known. The output electric power of the generator is also measured, resulting in the generator efficiency by dividing the electric power with the shaft power (torque times angular velocity). This procedure is carried out in the range of 1000 to 8000 RPM and for resistor sets with $R = 4.7 \text{ Ohm}$, 10 Ohm, and 20 Ohm. For each of these electrical loads, multiple calibration runs have been carried out, taking care that the RPM range is covered by

both ramping up and down and randomly alternating between the two. Note that it is made sure that the rotational direction of the generator is the same as during the experiments and that the voltage is always measured over the same resistor connected to the same coil pair of the generator. As a function of the RPM and the applied load, the measured efficiency of the generator ranges between 40% and 75%. To show the accuracy and reproducibility of the generator calibration, two different generators have been calibrated and connected to the same turbine set-up. The results in Sec. 3.5 show a good agreement in the measured turbine performance for several electrical loads, therewith validating the generator calibration procedure.

3.5 Validation and error estimation

During the first experiments of this work, the measured performance was varying quite significantly, for example after remounting a turbine several weeks later. This resulted in some inconsistent results, especially as a function of the acoustic frequency and the generator load. Since the initial aim of this work is to characterize the turbine performance as a function of these parameters, as well as investigate small changes in the turbine set-up, the experimental procedure needed to be improved. Furthermore, some aspects only have a subtle influence on the turbine performance, such as the effect of the acoustic frequency. Therefore, it needs to be known which measured differences are actually significant and what could be a measurement error. This section will provide the results of the work that has been done to improve and validate the experimental procedure and reduce and quantify the magnitude of several sources of error. This is presented quite extensively in an effort to construct a basis on how bidirectional turbine measurements can be accurately done in thermoacoustics.

Table 3.1 provides an overview of the measured turbine efficiency for the validation and error estimation topics treated in this section. To easily compare different measurements, only the maximum efficiency is given for each experiment. The latter was found to be a sufficient measure for comparison since the conclusions do not change when looking at the full acoustic power sweeps. Furthermore, the presented efficiencies are for a single acoustic frequency and resistor set (unless stated otherwise), but it has been confirmed that the results are similar for other operating conditions.

To investigate the repeatability of the measurements, a turbine has been mounted in the experimental set-up and kept there for over a month. Table 3.1 shows the results of six measurements done during a single day, as well as the efficiency during six days of the month where the largest variations in ambient weather conditions occurred. During the same day, the repeatability of the turbine efficiency is within 0.3%, or within 0.1% if the unrealistic case of extremely heating the generator and load is neglected.

Larger deviations occur during the period of one month, with a maximum difference of around 0.8 % when the atmospheric pressure and temperature are considered constant. In an attempt to have a better repeatability, the ambient pressure and temperature were also logged during the measurements. In the experimental processing, the gas properties of the air have been adjusted accordingly, yielding slightly different values for the calculated acoustic powers (see Eq. 3.2). Including the atmospheric conditions has resulted in a maximum deviation of 0.3 % instead of 0.8 % for the same measurements, showing that the accuracy of the experiments can be significantly improved in this manner.

Another way to improve the measurements is to ensure an accurate phase difference between two adjacent pressure sensors, since even a small phase difference can have a large effect on the calculated acoustic power. The latter becomes clear when varying the pressure sensor locations during a single experiment, which in perfect experiments should not affect the calculated turbine performance. Without incorporating the phase difference measured during calibration, there is a spread of 0.7 % in the efficiency for varying sensor locations (see Table 3.1). This reduces to 0.4 % when including the phase calibration in the acoustic power calculation. Furthermore, there is an average difference of about 0.6 % between the calibrated and non-calibrated case. This shows that for both the absolute value as well as minimizing the spread for varying sensor locations, including the phase difference is important, even though the calibration only shows a maximum difference of 0.1° between the sensors (see Sec. 3.4). Note that for measurement #5, one sensor was only 50 mm from the closed end and 300 mm from its adjacent sensor, which resulted in a significantly deviating calculated efficiency. Acknowledging that a sensor should not be placed there and including the phase calibration, the spread in turbine efficiency is only 0.1 %. This independence of the sensor positions gives confidence in the acoustic power calculation and tube correction presented in Sec. 3.3. As a nuance on this, varying both the total tube length as well as the sensor locations resulted in a larger spread of the efficiency, as shown in Table 3.1. For all results presented in this work, the measurements are done with the intermediate tube length of 1300 mm.

Another possible source of error is the mounting of the turbine in the tube section, since it can be difficult to place the guide vanes straight in the same axial plane of the tube. Especially since their axial length is only 10 mm, the guide vanes tend to twist and rotate when positioning them in the tube. As a consequence, the connected rotor is also not mounted straight, resulting in a changed performance and possibly scraping of the rotor against the outer tube. To prevent this, the guide vanes are mounted by applying force from both sides, and subsequently measuring the axial placement of the guide vane. If there is no significant difference around the whole circumference of both guide vanes, the mount is correct, otherwise it is repeated. By carefully mounting the turbine with this procedure, a maximum difference of 0.1 % in

Table 3.1: Overview of experimental validation and error estimation results. To easily compare different measurements, only the maximum turbine efficiency, η_t , is given for each experiment. The topics have been investigated with varying turbine designs, so one should only compare the values in each individual row, with the exception of the two rows that treat the influence of atmospheric conditions and phase calibration.

	$\eta_{t,1}$ [%]	$\eta_{t,2}$ [%]	$\eta_{t,3}$ [%]	$\eta_{t,4}$ [%]	$\eta_{t,5}$ [%]	$\eta_{t,6}$ [%]
Repeatability on the same day	23.57	23.56	23.53	23.49	23.28 ¹	23.49
Repeatability during one month, with measured atmospheric conditions ²	24.03	23.94	24.10	24.10	24.19	24.19
Repeatability during one month, with constant atmospheric conditions ²	23.64	23.50	23.97	24.16	24.74	23.97
Varying sensor position (with phase calibration) ³	23.08	23.20	23.12	23.18	22.83	
Varying sensor position (without phase calibration) ³	23.80	23.91	23.73	23.85	23.25	
Remounting turbine for 4.7 Ohm (#1 and #2) and 10 Ohm (#3 and #4)	25.14	25.15	25.38	25.32		
Varying generator for 10 Ohm (#1 and #2) and 20 Ohm (#3 and #4)	23.64	23.60	23.36	23.24		
Varying tube length with 900 mm (#1), 1300 mm (#2), and 1700 mm (#3)	25.75	25.51	25.21			
Reproducibility 3D prints ⁴	24.46	24.31	24.51			
Varying 3D print technique and material ⁵	25.45	25.90	26.79			

¹This measurement was done with seriously heated resistors and generator by pre-running at maximum power for a considerable amount of time. After cooling down for about half an hour measurement #6 was carried out.

²Selected from one month of measurements where the largest variation in atmospheric conditions occurred, with #1: 1027 hPa, 18.6 °C, #2: 1027 hPa, 18.0 °C, #3: 1018 hPa, 19.3 °C, #4: 1009 hPa, 18.9 °C, #5: 991 hPa, 19.6 °C, and #6: 1023 hPa, 19.5 °C. For the same measurements, results are given where the atmospheric conditions in the processing are set constant at the mean of 1011 hPa, 18.8 °C.

³Each given efficiency is for a different set of pressure sensor locations, while the same turbine is mounted. The sensor locations vary from 50 mm up to 400 mm from the edge of the turbine. Results are shown for both neglecting and incorporating the phase calibration of the pressure sensors when calculating the turbine efficiency.

⁴Results for three 3D printed rotors of the same design and material, manufactured from PA 2200 plastic using Selective Laser Sintering. The values are corrected up to the same tip clearance. Without correction the efficiencies are 24.38 %, 24.31 %, and 23.82 %.

⁵Results for three 3D printed rotors of the same design but different materials and printing techniques, with #1: PA 2200 plastic using Selective Laser Sintering, #2: SR 200 acrylic plastic using material jetting, and #3: PA 12 nylon plastic using Multi-Jet Fusion. The values are corrected up to the same tip clearance. Without correction the efficiencies are 25.45 %, 26.65 %, and 27.65 %.

turbine efficiency is measured for different mounts (see Table 3.1), which is a sufficiently small remounting error.

When keeping a single turbine mounted, it is possible to investigate the influence of the generator on the turbine efficiency. In theory, the calibration presented in Sec. 3.4 should eliminate any influence of the generator by calculating the shaft power based on the calibrated generator efficiency. To validate this procedure, a second generator of the same type has been calibrated and connected to the rotor while keeping the turbine mounted in the tube. Table 3.1 shows a maximum difference in turbine efficiency of 0.1 % when comparing both generators for two electrical loads. This demonstrates that the generator calibration is properly carried out and that the turbine performance can be regarded as independent of the used generator.

The final topics presented in Table 3.1 are about the accuracy of the 3D printed turbines. This is considered a topic of interest since some dimensions of the turbine design are in the same range as the accuracy of the 3D printing technique (approximately 0.2 mm). Especially the rotor has some small and critical details, *e.g.* at the tip, which is why three rotor prints of the same design have been ordered to investigate its reproducibility. Two of the rotors have a maximum efficiency within 0.1 %. However, the third rotor is found to deviate up to 0.6 %. After close examination and measurement of the prints, it was found that the third rotor has about a 0.05 mm smaller radius. From the results in Sec. 5.3, it is found that a corresponding smaller tip clearance of 0.05 mm for an unshrouded rotor can account for about 0.7 % in turbine efficiency. Table 3.1 presents the efficiencies of the three rotors after they have been corrected according to their relative tip clearance difference. The maximum efficiency spread between the three turbines is now only 0.2 %. First of all, this shows the need to measure the actual size of the prints for accurate tip clearance measurements. Furthermore, if the efficiency is corrected for the tip clearance effect, the performance is reproducible when re-printing turbines using the same material and technique. The latter is sufficient for this work, since all comparisons will be done using prints from this 3D printing procedure. However, it is interesting to check the turbine performance for different materials and production techniques. Therefore, two additional rotors have been produced from different materials and printing methods. Table 3.1 provides production details of these rotors and their performance during measurements. Both of these rotors have a higher maximum efficiency than the rotor used in this work. Even after correcting for the tip clearance, the third rotor yields a 1.3 % higher efficiency. This can be caused by a difference in rotor details due to the printing accuracy (besides a smaller tip clearance), as well as varying material properties. For the latter, especially the surface roughness is important. It is shown that a different surface roughness can have a significant effect on the pressure drop and acoustic losses in appendix D of the work by Oosterhuis.⁸ On its own, this effect would already be large enough to explain the measured differences.

From the topics that are treated in this section, it has become clear that calibrating the phase difference of the pressure sensors and accounting for the atmospheric conditions is important to accurately calculate the acoustic power. If a turbine is confirmed to be correctly mounted and the same material and 3D printing technique is used, the experimental procedure is shown to have good reproducibility and a small error. There is not enough data for each topic to do proper statistics, but from the presented results it is reasonable to conclude that any measured difference larger than 0.5 % is not a measurement error but a trustworthy result. Note that this conclusion is drawn for the turbine design used in Chapter 4, which has a maximum efficiency of around 25 %. By repeating some of the error estimations for turbine designs reaching up to 40 % efficiency in Chapter 5, it was found that these errors scale approximately linear with the maximum efficiency. Therefore, the threshold for the most efficient turbines in this work is around 0.8 % when comparing different results.

4

Characterization

The performance of a bidirectional impulse turbine is investigated experimentally for varying acoustic conditions and turbine loads. The results are used to characterize the turbine performance and compare it to steady flow turbomachinery and turbines in oscillating water columns. A dimensional analysis is done to identify the variables that influence the turbine performance, after which a scaling is sought that uniquely determines the efficiency of the turbine. This chapter is completed by providing the impedance of the bidirectional turbine, such that it can be used to implement it in a thermoacoustic device.

4.1 Introduction

In 2014, it was first shown that it is possible to run a bidirectional impulse turbine in a thermoacoustic device.⁶² However, since the main purpose of this work was to show the proof of concept, many details about the performance characteristics of the turbine remain unknown. Subsequently in other work, several Wells and impulse turbines have been tested in the scope of thermoacoustics, yet without a load attached to the turbine and only for unidirectional flow.¹¹² The underlying assumption for such unidirectional tests is that the oscillating flow in question can be assumed to be quasi-steady, which is indeed shown to be the case for oscillating water columns (OWCs).^{127,128} Moisel and Carolus present a derivation to show that the effect of the inertia of the accelerated or decelerated fluid in the turbine can be neglected in OWCs.¹²⁸ They conclude this for ocean waves with a fre-

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quency of around 0.1 Hz, with the note that this inertial effect becomes more relevant for higher frequencies such as 1 Hz. Since thermoacoustic engines generally work in the range of 20 Hz to 200 Hz, it is questionable whether the flow in the bidirectional turbines can still be regarded as quasi-steady under these circumstances. Since most of the performance characteristics used for turbines in OWCs are also based on steady flow, it is unclear whether the results from this field are still applicable for thermoacoustics. Besides this, it is quite unknown in general how well a bidirectional turbine performs under varying thermoacoustic conditions. These topics have remained open since the literature on bidirectional turbines in thermoacoustics is very scarce. The main aim of the work in this chapter is to alleviate some of this uncertainty by characterizing the performance of a bidirectional impulse turbine under varying thermoacoustic conditions.

The performance of the axial impulse turbine presented in Sec. 3.2 has been measured for four acoustic frequencies, each with three different loads connected to the turbine. For each of these twelve experiments, an acoustic power sweep with approximately fifteen setpoints has been performed using the experimental set-up given in Sec. 3.1. The raw data, processing scripts, and processed results for all experiments presented in this chapter can be found in a supplementary data publication.¹²⁴ The results from these measurements will be used to characterize the bidirectional turbine performance. This starts with analyzing the velocity diagram in Sec. 4.2, followed by an overview of the experimental results with several performance indicators in Sec. 4.3. Subsequently, a dimensional analysis and scaling of the results is done in Sec. 4.4. Finally, the impedance of the turbine, with a focus on implementing it in a thermoacoustic engine, is presented in Sec. 4.5.

4.2 Velocity diagram

A conventional way to visualize the flow angles for steady flow turbines is to draw a velocity diagram; see for example the turbomachinery book of Dixon.¹²⁹ Such a velocity diagram depicts the magnitude and direction of the flow in the turbine stages. Since the rotor is moving relative to the guide vanes, there is a distinction between absolute velocity vectors, c , at angles, α , and relative velocities, w , at angles, β . By assuming the axial velocity is constant and the flow follows the geometry of the blades, the velocity diagram can be drawn. In the simple case of a single guide vane and rotor in steady flow, one can use the velocity diagram to find the most efficient operating point by making sure the relative inlet angle of the rotor, β_2 , is equal to the geometric angle of the rotor blade, since there are the least amount of flow losses in this situation. Because the relative velocity and angle depend on two operating variables, namely the axial velocity, c_x , and the blade speed, U , it is useful to define the ratio of these two as the flow

Figure 4.1: Velocity diagram for the bidirectional impulse turbine at the largest axial velocity during the acoustic period (denoted by the blue dot on the red sinusoidal waveform). The sign convention is such that all angles are taken positive as depicted in this figure. The velocities and blade speed are taken from the experiment with 70 Hz acoustic frequency and 10 Ohm generator load at its most efficient setpoint. A video of the velocity diagram with varying axial velocity at constant RPM can be found in the data publication.¹²⁴

coefficient ϕ :

$$\phi = \frac{c_x}{U}. \quad (4.1)$$

Dixon states that, when ignoring Reynolds number effects, the efficiency of the turbine is a unique function of this flow coefficient, with the most efficient operating point as described before.¹³⁰

Now that the relevance of the velocity diagram for steady flow turbomachinery has been made clear, it is interesting to apply this method for the bidirectional impulse turbine in thermoacoustic conditions. Note that the validity of the velocity diagram will be questionable, since a quasi-steady axial flow assumed. Besides the effect of the acoustic frequency on the quasi-steady assumption, it is at least not completely valid since the axial velocity on either side of the turbine shows a phase difference of up to a few degrees. However, even in the case the flow is not completely quasi-steady, the analysis could still provide useful details about the flow in the bidirectional turbine.

Fig. 4.1 depicts the velocity diagram at the moment when the axial velocity is at its maximum for a given blade speed. A video of the velocity diagram as the axial velocity changes during the first half of the acoustic period is given in the data publication.¹²⁴ Following the steady flow analysis, for efficient operation the relative inlet angle to the rotor, β_2 , should be equal to the 60° angle of the rotor. Furthermore, the absolute outlet angle of the rotor, α_3 , should equal the 60° angle of the downstream guide vane

to have minimal losses here. The first important thing to notice is that, since the axial velocity is changing during the period while the blade speed is constant, there can only be one axial velocity for which these flow angles are as desired. During the rest of the acoustic period, especially for small axial velocities (see the video), the flow angles are far from optimal. Following this observation it stands to reason that, to get the best flow angles on average, one should focus on the flow angles at the RMS velocity. So during the first half of the period, one could say that the angle of the upstream guide vane and downstream guide vane should be changed such that the β_2 and α_3 are optimal at the RMS velocity. However, during the second half of the period the axial flow direction reverses, making the upstream guide vane the downstream guide vane, and vice versa. Changing the guide vane angles as mentioned before would provide drastically unfavorable flow angles during this second half of the period. It is therefore best to stick with the same angle for both guide vanes, as is done in the current work. The consequence of such a design for the flow angles can be seen from Fig. 4.1. Even for the most efficient setpoint of the turbine and at the maximum axial velocity, the β_2 and α_3 are only 23° , while ideally they should be 60° . The analysis in this paragraph shows that the main cause for such unfavorable flow angles is the oscillating nature of the flow, and thus the need for a downstream guide vane with accompanying losses. Therefore, a bidirectional turbine in acoustic flow can never be expected to be as efficient as a conventional turbine in steady flow. Note that in OWCs the flow conditions are sometimes improved by introducing self-pitching guide vanes.¹³¹ However, these are not considered for thermoacoustic devices since the operating frequency is orders of magnitude higher.

So far, this section has focused on the velocity diagram for the bidirectional turbine at a single setpoint. To investigate the effect of the flow angles on the turbine efficiency, the turbine performance at several setpoints has to be compared. Just as done by Dixon for steady flow,¹³⁰ the efficiency is examined for varying flow coefficients, now with the RMS values as the corresponding axial velocity. A video showing the velocity diagram, flow coefficient, and turbine efficiency for 70 Hz and 10 Ohm at varying setpoints is given in the data publication.¹²⁴ From the video it can be seen that flow coefficient where the turbine has the best efficiency is at $\phi = 0.54$, which is in the same range as reported for this bidirectional turbine design in OWCs.¹¹⁵ The flow diagrams show that this most efficient point does not correspond to the best flow angles, as described earlier in this section. The largest turbine efficiency is found where at the RMS velocity the β_2 and α_3 are nearly horizontal, i.e. 0° , which would be the case for $\phi = 0.58$. For the eleven other experiments at different operating conditions, the same holds approximately true, with the flow coefficient at maximum efficiency ranging from 0.49 to 0.63. Since there is such a spread in this optimum flow coefficient, as well as the fact that the most efficient setpoint is not at the best flow angles, it

Figure 4.2: Graph of the measured RPM as a function of the pressure drop amplitude over the turbine. The legend shows the used load resistance and the acoustic frequency for the different measurements.

can be said that the best operating point for the bidirectional turbine in thermoacoustic conditions can not be concluded from a velocity diagram alone. This can in part be true because the flow can not be assumed as quasi-steady. However, and more importantly, in the next sections it will be shown that the maximum turbine efficiency in thermoacoustic conditions is determined by more than just the flow coefficient (and thus velocities) alone.

4.3 Performance indicators

In contrast to the thermoacoustic results presented in the previous section, similar bidirectional turbines do show a unique dependence of the flow coefficient on the turbine performance for both steady and oscillating flow measurements in OWCs.^{114,115,122} This raises the questions of what is different under thermoacoustic conditions and what additional performance indicators should be considered. To answer the former question, the turbine RPM is shown as a function of the pressure drop amplitude over the turbine, ΔP_a , in Fig. 4.2. First of all, there is a difference in RPM for the varying generator loads (depicted by the different colors). This could be expected since for the same power input one generally sets the turbine RPM by changing its load. However, there is also a clear dependence on the acoustic frequency shown by the different markers in Fig. 4.2. The same holds true when comparing the axial velocity amplitude to the RPM, albeit to a lesser extent. Following Suzuki *et al.*,¹²² in OWCs scaling the pressure drop with the density and velocity squared leads to a single, linear performance curve with respect to the

Figure 4.3: Graph of the measured RPM as a function of the acoustic power drop over the turbine. The legend in Fig. 4.2 shows the used load resistance and the acoustic frequency for the different measurements.

flow coefficient. This is not the case for the current measurements, where each combination of electrical load and acoustic frequency leads to a different, non-linear curve. The reason for this is that, in thermoacoustics there is (nearly) always a phase difference between the pressure and velocity, as denoted by ϕ_{pU} in Eq. 3.1. Even in traveling wave engines, where in theory a traveling wave has zero phase difference, there is always a standing wave component resulting in at least a few degrees of phase difference. Note that in the current work, a closed tube is used, leading to much larger standing wave ratios and thus phase differences.

To resolve the ambiguity of individually using the pressure drop and velocity as performance indicators, there is a need to combine the two and correctly incorporate the phase difference between them. As can be seen from Eq. 3.1, the local acoustic power is such a measure. Fig. 4.3 presents the same experimental data as before, but now with the RPM as a function of the acoustic power drop over the turbine. It is interesting to see that, when using the acoustic power drop as a performance indicator, there is no influence of the acoustic frequency on the turbine RPM anymore. Furthermore, the curves for all three resistor sets seem to pass through the origin (0,0), which rightfully indicates that there would be no turbine output when there is zero power input. Note that this is not the case when using the pressure drop or velocity amplitude as a performance indicator. The need for combining these two with the correct phase difference shows why the conventional performance characteristics have only limited success in describing the bidirectional turbine in thermoacoustic conditions. Some indicators, such as the turbine efficiency by Setoguchi *et al.*,¹¹⁵ are still valid when correctly incor-

Figure 4.4: Turbine efficiency as a function of the acoustic power drop over the turbine. The legend in Fig. 4.2 shows the used load resistance and the acoustic frequency for the different measurements.

porating the phase difference. However, to avoid unnecessary errors and provide a clear way to describe the turbine performance, this work continues with using the acoustic power drop as a performance indicator and to calculate the turbine efficiency.

As described in Sec. 3.3, the turbine efficiency is calculated as the delivered shaft power divided by the acoustic power drop. The efficiency for all operating conditions is given in Fig. 4.4 as a function of this acoustic power drop. The first thing to notice is that also for the turbine efficiency there is no clear influence of the acoustic frequency. Some small variations might be present as a function of the frequency, but these are not consistent and the differences lie within the measurement error as given in Sec. 3.5. Therefore, as already suggested by de Blok *et al.*,⁶² the bidirectional impulse turbine performance can be regarded as independent of the acoustic frequency. This is an important result for implementing the turbine in a thermoacoustic engine, since it is not necessary to match the operating frequency of the engine to efficiently run the turbine. This is an advantage over a linear alternator where it can be difficult to match the engine frequency to the resonance frequency of the alternator. Note that the current conclusion is valid for 1 mm spacing between the rotor and guide vanes. In Sec. 5.4 the axial spacing is increased up to a point where there does start to be an influence of the frequency on the turbine efficiency.

Another interesting thing to note from Fig. 4.4 is that the maximum efficiency of the turbine is around 25 %. This is quite a bit lower than the maximum efficiency of 37 % found for this same turbine in OWCs.¹¹⁵ The

main difference between the turbine from the OWC and this work is that the current tip clearance of 1 mm is relatively large. In Sec. 5.3, the influence of this tip clearance is investigated in an effort to increase the turbine efficiency.

The final thing to notice from Fig. 4.4 is that the efficiency varies significantly as a function of ΔE_2 for the different electrical loads, but that the maximum efficiency is the same for all cases. This shows that the turbine can be run at the same maximum efficiency by varying the load for a wide range of operating conditions. To quantify what this load should be in a real engine, the next section focuses on finding a performance indicator that scales the turbine efficiency for different loads into a single metric.

4.4 Dimensional analysis and scaling

To find a function that uniquely determines the turbine efficiency, it is first necessary to identify all variables that influence it. For this purpose, a dimensional analysis for geometrically similar machines is performed which assumes the following relationship among the variables:

$$\eta = f_1(N, D, \rho, \mu, \Delta E_2, c_x), \quad (4.2)$$

with N the rotation rate, D the turbine diameter, ρ the density and μ the dynamic viscosity. Note that the shaft power is also a dependent function of these same variables, but only the efficiency is considered here. By choosing N , D , and ρ as the repeating variables, the efficiency is found to be a function of the following three dimensionless groups:

$$\eta = f_2\left(\frac{\rho N D^2}{\mu}, \frac{c_x}{N D}, \frac{\Delta E_2}{\rho N^3 D^5}\right) = f_2(Re, \phi, \Delta E_2^*). \quad (4.3)$$

By interpolation between the experimental results while keeping ϕ and ΔE_2^* constant, it is found that the influence of the Reynolds number on the turbine efficiency can be neglected. This is in accordance with results found for classical turbomachinery¹³⁰ and OWCs,¹³² where for the latter the critical Reynolds is found to be 40,000 for the same turbine design and a hub-to-tip ratio of 0.7. Note that the typical Reynolds numbers in this work are in the range of 50,000 to 100,000.

What remains is the relationship $\eta = f_2(\phi, \Delta E_2^*)$, with a yet undetermined function f_2 . By examining the experimental data for all three resistor sets at 70 Hz, it is attempted to find this function. To quantify the success of a proposed function, the points at which the maximum turbine efficiency is achieved are compared. For example, in Sec. 4.2 it was shown that the flow coefficient at maximum efficiency varies for different experiments, which for 70 Hz is between $\phi = 0.54$ for 20 Ohm and $\phi = 0.63$ for 4.7 Ohm. This equates to a relative difference of 17 %, from which it was concluded that the flow coefficient does not uniquely determine the turbine efficiency. Similarly, the

(a)

(b)

Figure 4.5: Turbine efficiency as a function of (a) the dimensionless acoustic power difference (b) the thermoacoustic input coefficient. The results are for an acoustic frequency of 70 Hz with the colors for the different resistors sets as show in the legend of Fig. 4.2.

influence of the dimensionless acoustic power drop, ΔE_2^* , is examined. It can be seen from Fig. 4.5a that ΔE_2^* is also not a single expression to determine the turbine efficiency with. Especially for large values, which are at small RPMs, there is a big spread between the curves. The maximum efficiency peaks for the different resistor sets also have a relative difference of 45 %.

From the results presented in the previous paragraph, it is clear that successfully scaling the turbine efficiency has to be done with a function that incorporates both ϕ and ΔE_2^* . A potential function is provided by Setoguchi *et al.*,¹¹⁵ who show a scaling for pressure drop times flow rate (ΔPQ), which they denote by the input coefficient, C_A . This scaling uses both the axial velocity and the blade speed, U , and when applied to ΔE_2 is given as follows:

$$C_A = \frac{\Delta E_2}{\rho D^2 c_x (c_x^2 + U^2)} = \frac{\Delta E_2^*}{\phi^3 + \phi}. \quad (4.4)$$

Note that by using $U \sim ND$, it is possible to rewrite the scaling in terms of ϕ and ΔE_2^* only, as given on the right hand side of the equation. When examining this scaling with respect to the turbine efficiency, it is found that the maximum efficiency peaks still have a relative difference of 17 %. This shows that this is also not a function that uniquely determines the turbine efficiency. However, especially for large and small RPM, this scaling shows more promise than using ϕ or ΔE_2^* alone. Therefore, following this direction, a new scaling is found. It is referred to as the thermoacoustic input coefficient, C_{ta} , and is given as follows

$$C_{ta} = \frac{\Delta E_2}{\rho D^2 c_x^2 (c_x + U)} = \frac{\Delta E_2^*}{\phi^3 + \phi^2}. \quad (4.5)$$

In Fig. 4.5b, this thermoacoustic input coefficient is shown with respect to the turbine efficiency. It can be seen that the results for the different experiments fall onto a single curve. The peaks have a relative difference of less than 4 %, while there is also a good scaling for the operating conditions on either side of this peak. Since including the experiments performed at other acoustic frequencies does not change these conclusions, it can be stated that C_{ta} seems to be a function that uniquely determines the turbine efficiency. In Sec. 6.5, it is examined whether this scaling still holds under more widely varying operating conditions in a real thermoacoustic device.

4.5 Acoustic impedance

The final part of this characterization focuses on the specific acoustic impedance of the bidirectional turbine. This is mainly important for implementing the turbine in a thermoacoustic engine, since the impedance difference caused by the turbine directly influences the acoustic conditions, and therewith the performance of the whole device. The impedance of the turbine is a function of its RPM, where the latter can be set by changing the

(a)

(b)

Figure 4.6: Real part (a) and imaginary part (b) of the normalized specific acoustic impedance of the turbine as a function of its RPM. The legend in Fig. 4.2 shows the used load resistance and the acoustic frequency for the different measurements.

load such that the turbine operates around its most efficient point (according to C_{ta}). It is therefore interesting to know how the RPM will, in turn, change the acoustic impedance, since that information can be used to design an efficient thermoacoustic device as a whole. Note that the latter will be attempted in Sec. 6.3 and Sec. 6.4 by implementing the turbine impedance in a thermoacoustic model.

Locally, the specific acoustic impedance is defined as $z = P_a/u_a$, with P_a the complex pressure amplitude and u_a the complex velocity amplitude, which are determined from the measured pressure signals as described in Sec. 3.3. The specific acoustic impedance is calculated on both edges of the turbine, and subsequently subtracted from each other to determine the impedance of the turbine, z_t . The values are normalized by the characteristic specific acoustic impedance, $z_0 = \rho c$, where c denotes the speed of sound.

In Fig 4.6, the real part (a) and imaginary part (b) of the normalized specific acoustic impedance are given as a function of the turbine RPM. The real part of the impedance represent the in-phase component of the pressure and velocity, and can be regarded as flow resistance. It can be seen that the real part of the impedance increases when more acoustic power is absorbed, either by using a smaller load resistance at a given RPM, or by operating at a larger RPM for the same load. There is also a small influence of the acoustic frequency on the real part of the impedance. For a larger frequency, there is relatively more pressure needed to accelerate the flow, leading to larger values of the impedance. This effect is more clear in the imaginary part of the impedance, which describes the phase difference between the pressure and velocity. The imaginary part of the impedance is positive for all cases, meaning the turbine is an acoustic inertance, which is expected since the turbine causes a narrowing flow path. There is also a small effect of the turbine RPM on the imaginary part of the impedance, but just as for the generator load, this effect is not nearly as large as for the real part of the turbine impedance.

4.6 Conclusions

The performance of a bidirectional impulse turbine is experimentally characterized for varying thermoacoustic conditions and generator loads. From the presented velocity diagram, it is concluded that the flow angles for the bidirectional impulse turbine cannot be optimal due to the sinusoidal axial velocity and the downstream guide vane. Furthermore, in contrast to conventional turbomachinery and OWCs, the flow coefficient does not uniquely determine the turbine efficiency. It is shown that this is caused by the phase difference between the pressure and velocity, resulting in the need for the acoustic power drop as a performance indicator to combine these two. A dimensional analysis shows that the turbine efficiency is a function of the

dimensionless acoustic power drop and the flow coefficient, which are combined into a thermoacoustic input coefficient that does uniquely determine the turbine efficiency.

The specific acoustic impedance of the turbine is presented for varying operating conditions, which can be used to implement the turbine in a thermoacoustic engine. For this purpose, it is found that the acoustic frequency does not influence the efficiency of the turbine itself. This is a big advantage of the bidirectional turbine over a linear alternator, since no frequency matching has to be done to efficiently produce electricity from the acoustic power.

The maximum turbine efficiency is found to be 25 % for all acoustic frequencies and electrical loads. This efficiency is significantly less than the 37 % found for a similar turbine in an OWC. Therefore, future work should focus on increasing the efficiency by optimizing the current design, as will be attempted in Chapter 5.

5

Optimization

The main purpose of this chapter is to optimize the efficiency of the bidirectional impulse turbine, which is pursued by proposing several design alterations under varying thermoacoustic conditions. The influence of adding a shroud ring around the rotor blades is examined along with minimizing the tip clearance of the turbine. The axial spacing between the guide vane and rotor is varied to show the importance of this parameter for thermoacoustic conditions. Finally, a design study for the guide vane and rotor geometry is presented in an effort to further increase the turbine efficiency.

5.1 Introduction

In the previous chapter, it was confirmed that a bidirectional impulse turbine can convert acoustic power into electricity for a range of thermoacoustic conditions. The main focus of this chapter was on proving the concept and characterizing the performance of a first turbine prototype. The maximum efficiency of the turbine was found to be 25 %, which is significantly lower than the 37 % efficiency for the turbine in an OWC on which the first prototype was based.¹¹⁵

This chapter will focus on optimizing the efficiency of the bidirectional impulse turbine by experimentally investigating the influence of design changes for varying thermoacoustic conditions. The optimization is mostly based on work done for OWCs, but it should be kept in mind that the operating conditions in thermoacoustic engines are quite different. For example, the frequency is approximately 3 orders of magnitude higher and the pressure and

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velocity run partly out of phase. It is therefore interesting to investigate design changes since the turbine efficiency might not be affected in the same manner as for OWCs (or steady flow turbomachinery).

For OWCs, there is no significant influence on the turbine efficiency for varying the axial spacing between the guide vanes and the rotor.¹¹⁵ However, for thermoacoustic conditions the influence of the axial spacing could be important, since the displacement amplitude of the gas can be so small that it is in the same order of magnitude as the turbine dimensions. Therefore, the influence of the axial spacing will be investigated in Sec. 5.4. Before examining this, the effect of adding a shroud ring around the rotor blades is investigated in Sec. 5.2. This is done in an effort to increase the turbine efficiency by reducing the tip leakage, where the latter is shown to have an especially large effect for relatively big tip clearances.¹³³ Following this, the tip clearance of both unshrouded and shrouded rotors is varied in Sec. 5.3. With respect to the first prototype, it is expected that the efficiency can be significantly improved in this manner, since the tip clearance was relatively large for that turbine. Once the optimal axial spacing and tip clearance are set, a design study varying the turbine geometry is presented in Sec. 5.5. Since the guide vanes and rotor are chosen as the optimum ones found for a design study in OWCs,¹¹⁵ the main focus of this section is to identify whether this design is also optimal for thermoacoustic conditions. Note that the experiments are performed as described in Chapter 3, and that the raw data, processing scripts, and processed results for all experiments presented in this work can be found in a supplementary data publication.¹²⁵

5.2 Shrouded rotor

For an unshrouded rotor as shown in Fig. 5.1, there are tip losses caused by the pressure difference between the suction and pressure side of the blade, leading to radial flow into the gap between the outer tube and the tip of the blades.¹³³ This loss mechanism can be prevented by featuring a ring at the tip of the rotor blades, similar to the shroud ring shown for the guide vane in Fig. 5.1. For a shrouded rotor, the losses that remain are caused by the leakage flow around the rotor. Both loss mechanisms can be decreased by reducing the tip clearance, as will be investigated in Sec. 5.3. Here, the focus is on comparing an unshrouded rotor with a shrouded rotor for the same tip clearance of 1 mm. According to steady flow measurements by Yoon *et al.*, for this relatively large tip clearance compared to the tube radius of 30 mm, a shrouded rotor should perform significantly better.

The blades of the shrouded rotor are radially shorter by 1 mm, such that adding the 1 mm thick shroud ring provides the same tip clearance as for the unshrouded rotor. In Fig. 5.2, the turbine efficiency is shown for both rotors as a function of the acoustic power drop and for varying turbine loads. It

Figure 5.1: (Duplicate of Fig. 3.2) Schematic of an axial bidirectional impulse turbine mounted in a tube section. The guide vanes are connected to the outer tube by a shroud ring (only depicted for the right guide vane). The generator is mounted on the left guide vane and connected to the rotor. During operation, the generator is enclosed by the nose cone, which also ensures the axial flow is redirected towards the guide vane blades. Besides the unshrouded rotor, also shrouded rotors are used in this work.

5

can be seen that the efficiency of the shrouded rotor is significantly higher than for the unshrouded rotor, reaching up to about 3 % difference at the point of maximum efficiency. Furthermore, the maximum efficiency that can be reached for both cases is independent of the turbine load. Finally, just as shown for the unshrouded rotor in Sec. 4.3, it is found that the acoustic frequency has a negligible influence on the turbine efficiency for the shrouded rotor. It can therefore be said that the shrouded rotor outperforms the unshrouded rotor for all investigated operating conditions.

It is worth noting that the thickness of the shroud ring will also have an influence on the turbine performance, since it determines the blade height and thus the blade area of the rotor blades. Therefore, the influence of both a thinner and thicker shroud ring is investigated in the next section. Furthermore, the current conclusion that a shrouded rotor is superior can only be made for the current tip clearance. In the following section the tip clearance will be varied for both unshrouded and shrouded rotors to check whether this conclusion is true in general.

5.3 Tip clearance

Adding a shroud ring to the rotor has increased the maximum turbine efficiency to approximately 28 %, but this is not yet in the same range as the 37 % that Setoguchi *et al.* have measured in an OWC.¹¹⁵ With the geometry of the blades being the same, the main difference in design is identified as the tip clearance. Setoguchi *et al.* have a 1 mm tip clearance for a 300 mm tube

Figure 5.2: Turbine efficiency as a function of the acoustic power drop for an unshrouded and a shrouded rotor. The legend displays the resistance that is used for the electrical load. The unshrouded rotor is denoted by square ($\square$) markers and the shrouded rotor by diamond ($\diamond$) markers. All results are for an acoustic frequency of 70 Hz.

radius, resulting in a tip clearance ratio of $1/300 = 0.33\%$. This is more than ten times smaller than the tip clearance ratio of approximately 3.8% that has been used so far in this work. In this section the tip clearance is varied in an effort to increase the maximum turbine efficiency, and more generally to identify the influence of the tip clearance under thermoacoustic conditions. This is done for both shrouded and unshrouded turbines, since it has been shown that for steady flow turbomachinery an unshrouded turbine can become more efficient than a shrouded one for a small tip clearance.¹³³ Note that since the tube radius in the experimental set-up is only 30 mm, and the 3D printed turbines have a tolerance of 0.2 mm, the smallest tip clearance ratio that can be achieved is around 1%.

Along with varying the tip clearance of the rotors, the thickness of the guide vane shroud ring is varied, such that the tip of the rotor blades is at the same radius as the tip of the guide vane blades (see R_{tip} and the guide vane shroud ring in Fig. 5.1). For the unshrouded rotor, measurements have also been done by keeping the same nominal guide vane while changing the rotor tip clearance. This can be used to identify the influence of the tip radius of the guide vane blades. The thickness of the rotor shroud ring has been kept at 1 mm, except for two rotors at 2% tip clearance ratio, where also a 0.5 mm and 1.5 mm shroud ring has been used to investigate the influence of the ring thickness. To make a concise comparison between all the different cases, only the maximum turbine efficiency for a 10 Ohm load and a 70 Hz acoustic frequency is considered here. It can be seen from the Fig. 5.2 that

Figure 5.3: Maximum turbine efficiency as a function of the tip clearance ratio, where $R_0 = 30$ mm and the radius of all rotors has been individually measured to determine the tip clearance. All maximum efficiencies are presented for a 10 Ohm load resistance and 70 Hz acoustic frequency. The data for all 192 experiments with varying frequencies and loads can be found in the supplementary data publication.¹²⁵ Note that the two points around 3.8 % tip clearance ratio correspond to the full performance curves given in Fig. 5.2.

this is a representative indicator, but the results for all measurements under varying operating conditions can also be found in the supplementary data publication.¹²⁵

Fig. 5.3 presents the maximum turbine efficiency as a function of the tip clearance ratio for all the turbines described in the previous paragraph. The most important thing to notice is that for the smallest tip clearance the turbine efficiency is more than 36 %, thus reaching the same performance level as measured in OWCs,¹¹⁵ even though the tip clearance ratio is still about three times larger here. For the unshrouded rotor, the turbine efficiency for increasing tip clearance also decreases similarly as computed by Thakker and Dhanasekaran for a bidirectional impulse turbine oscillating flow.¹¹⁶ This approximately linear relation between tip clearance and turbine efficiency, as well as the more parabolic relation for the shrouded rotor, is also found for steady flow measurements, albeit for significantly different turbine designs.¹³³ Furthermore, the shrouded rotor is also more efficient than the unshrouded rotor for all tip clearances in the examined range. However, when extrapolating the trend lines, it seems that the unshrouded rotor might become more efficient than the shrouded rotor for even smaller tip clearances. Such an effect has been experimentally measured by Yoon *et al.*, who denote the crossing point as the break-even clearance. When examining the results

for a 4.7 Ohm turbine load, it is found that the unshrouded rotor is already more efficient than the shrouded rotor for the smallest tip clearance, with a break-even clearance ratio of 1.4 %. At this point, the unshrouded rotor for 10 Ohm still has a 0.6 % lower turbine efficiency than the shrouded one, while for 20 Ohm this goes up to 1.1 %. This difference for a varying turbine load is caused by the acoustic power, and therewith the pressure drop and velocity, at which the maximum efficiency occurs (as can be seen in Fig. 5.2).

An additional result that can be seen in Fig. 5.3 is the dashed red line, which represent measurements for varying tip clearance of the unshrouded rotor while keeping the same set of nominal guide vanes (with $R_{\text{tip}} = 29 \text{ mm}$). For a large tip clearance, it can be seen that this turbine set-up is slightly more efficient than for a turbine where the guide vane blades vary with the tip of the rotor blade, as represented by the solid red line. Similarly, the inverse is true for the smallest tip clearances. This result is as expected, since more blade area at a large radius will generally increase the turbine performance. What is maybe more interesting about these results is the consistency of them, even though the maximum difference in efficiency of 0.7 % is small. This attends to the accuracy at which these measurements can be performed.

The final result that can be seen in Fig. 5.3 is the influence of the thickness of the rotor shroud ring, which is varied at a tip clearance ratio of 2 %. For a 1.5 mm shroud ring, the maximum turbine efficiency is 30.8 %, which increases to 32.5 % for a 1 mm ring and 34.3 % for a 0.5 mm ring. This influence is way more significant than varying the blade height of the guide vanes, and shows that the maximum turbine efficiency for the shrouded rotors can be further improved by minimizing the shroud ring thickness. The latter will also shift the break-even clearance to smaller values, showing that for the current experimental set-up it is best to use shrouded rotors. For systems with much larger dimensions, it is easier to produce a turbine with a smaller tip clearance ratio or relative shroud ring thickness. For such circumstances, the unshrouded rotors could perform better than the shrouded ones. Following the trend line from Fig. 5.3, which is also confirmed by Yoon *et al.* to continue for smaller tip clearance ratios,¹³³ the maximum turbine efficiency could reach up to 40 %.

5.4 Axial spacing

Where the tip clearance usually has a large effect on the performance of turbomachinery, as show in the previous section as well, the axial spacing between the guide vane and the rotor is generally less significant. Setoguchi *et al.* show that in an OWCs with a frequency around 0.1 Hz, there is no influence of the axial spacing on the turbine efficiency.¹¹⁵ Scaled to the dimensions of this work, they showed this for axial spacings from 1 mm to 8 mm. It is interesting to investigate whether this relative independence of

Figure 5.4: Maximum turbine efficiency as a function of the axial spacing for two shrouded rotors and two unshrouded rotors. The shrouded rotors have a tip clearance ratio of 2.1 % ($\triangle$ markers) and 5.3 % ($\square$ markers). The unshrouded rotors have a tip clearance ratio of 2.1 % ($\circ$ markers) and 5.5 % ($\diamond$ markers).

axial spacing is also present in thermoacoustic conditions, since the frequency of oscillation is orders of magnitude higher, resulting in a much smaller displacement amplitude of the working fluid. So far all measurement have been done with an axial spacing of 1 mm. In this section, this is increased up to 6 mm, as well as decreased to 0.5 mm, where the latter is on the limit of what can be achieved in the experimental set-up.

Fig. 5.4 presents the maximum turbine efficiency as a function of seven different axial spacings. Results are presented for unshrouded rotors and shrouded rotors, both with two different tip clearance ratios. The first thing to notice is that there is a significant influence of the axial spacing on the turbine efficiency for all turbines. This effect is the biggest for the shrouded rotor with a large tip clearance ratio (5.3 %). This can be explained by an interaction between the axial spacing and the tip clearance. As briefly described in Sec. 5.2, the main tip clearance losses for a shrouded rotor are associated with a leakage flow around the rotor (as opposed to the unshrouded rotors where the dominant effect is leakage at the tip of the blades). When the spacing between the guide vane and rotor reduces, there is more resistance for the flow in the axial space to move radially outwards and subsequently flow around the shrouded rotor. Since the rotor with the smaller tip clearance already has significantly less leakage flow, the influence of the axial spacing is not nearly as large here. Furthermore, it can be seen that for the smallest axial spacing of 0.5 mm, this shrouded rotor actually has a slightly smaller efficiency, which can also be seen for the unshrouded rotor with a

Figure 5.5: Turbine efficiency as a function of the axial displacement amplitude for the shrouded rotor with a tip clearance ratio of 2.1 %.

small tip clearance. This effect has been measured in other work as well,¹³⁴ and can be explained by the increase in viscous losses due to the interaction between guide vane and rotor, which in this case outweighs the increase in performance by having a smaller spacing.

Another thing that can be seen in Fig. 5.4 is that the efficiency is higher for a smaller acoustic frequency as the axial spacing becomes larger. This might be an effect of the displacement amplitude of the working fluid, since for an equal axial velocity this is inversely proportional to the acoustic frequency. To look at this in more detail, the turbine efficiency as a function of the displacement amplitude is presented in Fig. 5.5 for a shrouded rotor with 1 mm and 6 mm spacing. For the 1 mm axial spacing measurements (in gray), it can be seen that the displacement amplitude at which the maximum turbine efficiency occurs is indeed higher for 50 Hz than for 70 Hz. Furthermore, reducing the resistance to the turbine load for 50 Hz further shifts this displacement amplitude higher, which is expected since the maximum turbine efficiency occurs at a larger acoustic power drop (as shown in Fig. 5.2). The important thing to notice is that, even though the varying operating conditions cause a different displacement amplitude at which the efficiency is at maximum, the actual value of the maximum is constant ($\approx 32\%$). So the displacement amplitude has no influence on the maximum turbine efficiency for 1 mm axial spacing in the investigated range. It can be seen in Fig. 5.5 that this is not true anymore for the measurements done with 6 mm spacing (in black). For the measurement with 20 Ohm and 70 Hz, the turbine efficiency is now significantly lower than the cases where the maximum efficiency is at a larger displacement amplitude. Furthermore, when compared to the 1 mm axial spacing, the maximum turbine efficiency is reached at a

Figure 5.6: Maximum turbine efficiency as a function of the displacement amplitude divided by the axial spacing. Results are shown for all 84 experiments of the shrouded rotor with a tip clearance ratio of 2.1 %. Similar results for the other shrouded rotor and two unshrouded rotors can be acquired from supplementary data publication.¹²⁵

larger displacement amplitude for all cases. Both of these effects show that for this relatively large spacing of 6 mm, there is a clear influence of the displacement amplitude, with a significant decrease in performance for smaller displacement amplitudes.

Now that it is shown that the displacement amplitude can (but does not always) have an effect on the turbine performance, it is interesting to quantify this behavior with respect to the axial spacing. For this purpose, Fig. 5.6 depicts the maximum turbine efficiency for a shrouded rotor as a function of the displacement amplitude divided by the axial spacing. A clear trend of dramatically decreasing turbine efficiency can be seen as the displacement amplitude of the working fluid approaches the length of the axial spacing. This is true for all generator loads and acoustic frequencies, as well as for the other shrouded rotor and two unshrouded rotors. For relatively large displacement amplitudes with respect to the axial spacing, the maximum turbine efficiency is approximately constant. This shows that for efficient operation, one needs to make sure that at the point of maximum turbine efficiency, as determined by the scaling presented in Sec. 4.4, the displacement amplitude is large enough with respect to the axial spacing. This has been shown to be true for all operating conditions at 1 mm spacing, while for 0.5 mm the efficiency even slightly decreased. Therefore, 1 mm axial spacing is concluded as optimal for the given turbine designs, and will be used in the remainder of this work.

Figure 5.7: Schematic of the bidirectional impulse turbine (to scale). The dimensions are given in mm unless stated otherwise, and provided at the mean turbine radius of 24.8 mm. All dimension are given for the reference turbine of the design study. The dimensions in black are exact (not rounded) and completely determine the geometry. The values in gray follow from the defined geometry and the number of blades. Note that in the zoomed section the $\oslash$ symbol refers to lines being tangent at the corresponding points ($\bullet$) at the given angles.

5.5 Design study

In the previous sections, the implementation of the turbine has been optimized under thermoacoustic conditions, such that the efficiency is in the same range as for OWCs. This has been done for a geometric design of the rotor and guide vanes that is shown to be optimal in OWCs using a design study.¹¹⁵ In this section, a similar study is performed to investigate whether the shape of the guide vane and rotor blades is also optimal under thermoacoustic conditions.

The design of the guide vane and rotor blades that has been used so far is shown schematically in Fig. 5.7. The design is parameterized in such a way that all dimensions can be changed individually, while keeping the others constant. The values for the dimensions in Fig. 5.7 are for the reference geometry of the design study, for which the maximum turbine efficiency is measured to be approximately 37 % for 10 Ohm and 70 Hz. All design modifications in this section will be compared to the maximum efficiency of the reference design for the same operating conditions. Note that a tip clearance ratio of 1.3 % and a shroud ring thickness of 0.7 mm is used. The guide vanes have a tip radius of 29 mm, such that the tip of the guide vane blades aligns with the tip of the rotor blades. If desired, a higher value of the turbine efficiency can be achieved by using guide vanes with a tip radius of 29.5 mm, a shroud ring thickness of 0.5 mm, and a smaller tip clearance ratio, as shown

in Fig. 5.3. However, the aforementioned values are chosen since turbines with these dimensions can be more consistently 3D printed, which is an important consideration for the design study.

In Table 5.1, the maximum turbine efficiency for several geometry variations is compared with the reference design. Since all measured efficiencies were found to be very similar, all measurements have been done twice and compared with the efficiency for the reference design that has been measured on the same day. The first two variations that are presented in Table 5.1 focus on the tip of the rotor blade, which are points that are specific for how the turbine is designed and produced in this work. In theory, the tip of the rotor blade should be smooth and thin, and follow the desired angle of the rotor, α_r . To provide a fully determined geometry with a circle on the pressure side and an ellipse on the suction side of the rotor, there is a need to slightly offset the local tip angles by $\Delta\alpha$ (see Fig. 5.7). Since this value is arbitrary, it was checked whether changing this to the minimum possible value of 6° and an equally larger value of 14° would affect the turbine efficiency. The results in Table 5.1 show that this is not the case, confirming that it is fair to define the effective tip angle of the rotor in this way. For the tip radius of the rotor blade, R_t , the larger value of 0.4 mm shows a slightly better efficiency. This could be insignificant, just as the value for 0.0 mm, since it is still in range of the measurement error. A possible explanation of why this bigger radius would be advantageous is because of the production technique. For a smaller radius the tip is more blunt, while a smooth radius will be more efficient, as was found by changing the tips of the guide vanes from blunt to smooth for some early stage prototypes (increasing the turbine efficiency by 4 %).

With the topics specific to the implementation in this work treated, more general design variations can be investigated. In the remainder of Table 5.1, the influence of the rotor length, thickness of the rotor blade, and the number of rotor blades is presented. First of all, the variations with changing number of blades is found to be insignificant. This is also the case for changing the chord length of the rotor blade, L_r , which is in contrast to what has been found in OWCs.^{115,135} For the same relative variation in rotor chord length, an insignificant influence is found here, while a 1 % and 3 % reduce in turbine efficiency was found in OWCs. However, this was measured for turbines with self-pitching guide vanes, which makes a direct comparison invalid. The same holds true for the thickness of the rotor, which was again varied in relatively the same amount, but found to be of less influence. Nevertheless, there is a slight reduction in turbine efficiency measured for the smallest and largest rotor thickness, while a t_r of 2.9 mm could be slightly better than the reference design. To summarize the results presented in Table 5.1, it can be said that it is hard to draw the conclusion that specific design changes are better or worse, since the measured differences are so small. This does, however, not mean that the results are not trustworthy, but simply that the presented design changes do not have a big influence on the turbine perfor-

Table 5.1: For several geometry variations, the difference of the maximum turbine efficiency with respect to the reference geometry, $\Delta\eta_{t,max}$, is given. For each case, only a single design parameter is changed, with all other dimensions the same as given in Fig. 5.7.

	Reference design	Current design	$\Delta\eta_{t,max}$ [%]
$\Delta\alpha$	10°	6°	-0.1
		14°	-0.1
R_t	0.2 mm	0.4 mm	+0.7
		0.0 mm	+0.3
L_r	10.6 mm	8.6 mm	-0.2
		12.6 mm	+0.2
t_r	3.3 mm	3.7 mm	-0.7
		3.5 mm	+0.1
		2.9 mm	+0.7
		2.6 mm	-0.9
#blades	29	31	+0.1
		30	+0.1
		27	+0.2

mance. In the remainder of this design study, it will be shown that similarly small design changes can have a more significant influence on the turbine efficiency.

Table 5.2 presents the maximum turbine efficiency for varying guide vane and rotor angles with respect to the reference case, where both angles are 60° . For fixed guide vanes and in oscillating flow, the 60° design was found to be significantly better than 50° or 70° in OWCs.^{115,136} To narrow down on this range, turbines with matching guide vane and rotor blade angles of 55° and 65° are investigated here. Table 5.2 shows that the 55° turbine is 2.0 % less efficient than the reference design, while the 65° turbine is 1.2 % less efficient. This shows that when keeping the guide vane and rotor angle the same, a 60° turbine design is also superior for thermoacoustic conditions. However, what is also interesting from Table 5.2, is that a larger rotor angle for the 55° guide vane is more efficient, while a smaller rotor angle for the 65° guide vane is significantly less efficient. Note that such an influence was not reported for OWCs,^{115,136} but this does show that the turbine with a 60° guide vane could be made more efficient. In the bottom part of Table 5.2, this is indeed shown to be the case. Combining the 60° guide vane with a 65° rotor provides the highest turbine efficiency, while around 70° the efficiency equals that of the reference design again. Increasing the rotor angle further decreases the turbine efficiency, indicating that the optimum rotor angle for the 60° guide vane is indeed around 65° . The positive effect of a larger rotor angle can be explained with the velocity diagram, which was presented

in Sec. 4.2. Increasing the rotor angle for a given guide vane causes the flow to the inlet of the rotor to be at a worse angle, but the outlet flow towards the downstream guide vane at a better angle. The inverse is true for a smaller rotor angle for the same guide vane, which is shown to cause a decrease in turbine efficiency. It therefore seems that optimizing for the absolute flow angle at the downstream guide vane is more important for the turbine performance than the relative inlet angle to the rotor. In other words, the losses for a mismatch in flow angle are more dominant at the downstream guide vane than at the inlet of the rotor.

Table 5.2: For varying guide vane and rotor angles, the difference of the maximum turbine efficiency with respect to the reference geometry, $\Delta\eta_{t,max}$, is given. All other dimensions are the same as given in Fig. 5.7.

α_{gv}	α_r	$\Delta\eta_{t,max}$ [%]
55°	55°	-2.0
	60°	-1.3
	65°	-1.1
65°	55°	-3.6
	60°	-2.1
	65°	-1.2
60°	55°	-1.3
	60°	(reference)
	65°	+1.2
	70°	+0.1
	73°	-0.1
	75°	-1.3

5.6 Conclusions

The implementation of a bidirectional impulse turbine under thermoacoustic conditions is optimized to the extent that the maximum turbine efficiency is measured to be 38 %. This is now in the same range as reported for similar turbine designs in OWCs. The largest increase in the turbine performance was found by minimizing the tip clearance, increasing the efficiency from 25 % to 35 % for an unshrouded rotor where the tip clearance ratio was reduced from 3.8 % to 1.2 %. For nearly all investigated tip clearance ratios, adding a shroud ring around the rotor blades is found to increase the turbine efficiency, especially when the tip clearance is relatively large. Only for a load resistance of 4.7 Ohm at the smallest tip clearance, it is found that an unshrouded rotor performs better than a shrouded rotor. The break-even tip clearance ratio for this case is around 1.4 %. However, reducing the thickness of the shroud ring from 1 mm to 0.5 mm is shown to further increase the

turbine efficiency by 1.8 %. For the current design, it can therefore be concluded that shrouded rotors will be the most efficient for the entire range that is investigated. Only when scaling to large enough sizes, where the relative tip clearance can be made small enough, the unshrouded rotors can become more efficient than the shrouded ones.

In contrast to OWCs, it is found that the axial spacing between the guide vanes and the rotor has a significant influence on the turbine efficiency, especially for a shrouded rotor with a relatively large tip clearance. For an increasing axial spacing, both the acoustic frequency as well as the turbine load start to have an effect on the efficiency. The deciding factor for this influence is identified as the displacement amplitude of the working fluid, where a significant decrease in turbine efficiency is measured when the displacement amplitude approaches the length of the axial spacing. For large enough displacement amplitudes with respect to the spacing, the maximum turbine efficiency is found to be independent of this measure, therewith providing a range of operating conditions in which the turbine should be used for optimal performance.

For a shrouded rotor with a small tip clearance and spacing, a design study for the geometry of the guide vane and rotor blades is performed. The main purpose of the shape design study is to identify whether the optimal design for OWCs is also the best under thermoacoustic conditions. Since most design changes were found to only have a small effect on the turbine efficiency, it can be said that the reference design is in the right range, but it is hard to conclude that it is indeed optimal. For this purpose, future work could focus on more drastic design changes, or perform similar work for turbines at a larger scale, where the investigated design details will not be in the same order of magnitude as the accuracy of the production technique.

For the angle of the rotor and guide vanes, the measurements from the design study did show a significant influence on the turbine efficiency. From the results it can be concluded that a 60° turbine outperforms the ones with 55° and 65° angles. Furthermore, using a 60° guide vane along with a 65° rotor provides an additional increase in the turbine efficiency. When this design change is combined with further reducing the tip clearance, which should be possible for turbines at a larger scale, the results in this work show that a turbine efficiency of 40 % should be achievable under thermoacoustic conditions.

6

Implementation

A thermoacoustic model is described and used to efficiently implement the optimized turbine from Chapter 5 into a thermoacoustic refrigerator. Experiments are done for several gas types and mean pressures to identify their influence on the turbine efficiency. For all operating conditions, it is investigated whether scaling with the thermoacoustic input coefficient from Sec. 4.4 still holds. Furthermore, the ratio of acoustic power absorbed by the turbine over the total amount of available power is examined for varying conditions. Finally, the results are used to present a case study in which the turbine is used to drive the fluid pumps of the device. The remaining acoustic power is used for cooling, thus providing an off-grid thermoacoustic refrigerator that works purely with low-grade heat as an input.

6.1 Introduction

In Chapter 5, the bidirectional impulse turbine has been optimized to reach an efficiency of around 36 %, which is in the same order as found for OWCs.¹¹⁵ For both the lab experiments and the OWCs, the turbine is used under atmospheric conditions with air as a working fluid. However, thermoacoustic devices generally work with helium or other noble gases at elevated pressures between 10 bar and 40 bar. Initial indications have shown that a bidirectional impulse turbine can reach an efficiency of 76 % when used in a thermoacoustic device at a mean pressure of 10 bar.⁶² In this chapter, this claim is investigated by implementing the same bidirectional turbine as used

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in Chapter 5 in a thermoacoustic refrigerator. The mean pressure and the type of gas will be varied to examine their influence on the turbine efficiency. Furthermore, it is investigated whether the scaling derived in Sec. 4.4 for air at 1 bar is still valid for different pressures and working fluids.

Details about the thermoacoustic refrigerator and the measurement procedure are given in Sec. 6.2. A thermoacoustic model is presented in Sec. 6.3, which is used to efficiently implement the turbine in the refrigerator in Sec. 6.4. This will mainly focus on the performance of the device as a whole, while implementing the electricity production besides the refrigeration. For the identified configuration, the performance of the bidirectional turbine is presented in Sec. 6.5.1 for varying pressures and gases, while the performance of the complete device is treated in Sec. 6.5.2. This includes a case study in which the turbine is used to drive the pumps of the device, while the remaining acoustic power is used for refrigeration, therewith investigating the possibility of a completely off-grid thermoacoustic refrigerator.

6.2 Experimental set-up

Following the experiments from Chapter 5 under lab conditions, the exact same turbine prototype will be implemented in a thermoacoustic device here. This turbine has a rotor shroud ring of 1 mm thickness, a tip clearance of 0.3 mm, and an axial spacing between the guide vanes and the rotor of 1 mm. The shaft power produced by the turbine is converted into electricity by an electric motor (Hacker A10-13L) that is used as a generator. The three-phase power produced by the generator is dissipated in an electrical load consisting of sets of three precision resistors. To vary the turbine load for different operating conditions, four resistances can be used for the electrical load, namely $R = 0.75 \text{ Ohm}$, 2.0 Ohm , 4.7 Ohm , and 10 Ohm . The efficiency of the generator is calibrated for a varying turbine RPM with all electrical loads. The shaft power of the turbine, P_m , is calculated by dividing the measured electric power with the generator efficiency. More details on the turbine design, calibration, and measurement can be found in Chapter 3.

The turbine is implemented in a 4-stage thermoacoustic refrigerator that is shown schematically in Fig. 6.1. The two vessels that produce acoustic power are denoted as the engines, and the two vessels that provide cooling are depicted as the heat pumps. In each of the vessels, there are two heat exchangers that enclose the regenerator material in which the thermoacoustic effect occurs. In the engines, thermal oil is used as the heat source to produce the acoustic power. The input temperature, T_{in} , is measured and can reach up to 200°C for the presented experiments. In the heat pumps, a water-glycol mixture is used as the medium to transfer the cold. Besides the hot- and cold heat exchangers, each vessel has an ambient heat exchanger that uses water as a working fluid. The acoustic power is produced in the di-

Figure 6.1: Schematic of the thermoacoustic refrigerator with implemented bidirectional turbine (to scale). The pressure sensor locations (P1-P4) and tube dimensions are given in mm.

rection from the ambient heat exchanger to the hot heat exchanger, resulting is the acoustic power traveling through the device as depicted in Fig. 6.1.

The turbine is implemented in the feedback tube between the second heat pump and the first engine, as shown in Fig. 6.1. In Sec. 6.4, this configuration is shown to provide the best performance in terms of produced acoustic power for the entire device. Note that for practical reasons, the turbine is implemented in a feedback tube with a 700 mm straight part (denoted as long), which is in contrast to the standard tubes with a 600 mm straight part (denoted as normal). To acoustically compensate for this extra length, a long feedback tube is also used between engine #1 and engine #2. All feedback tubes have a diameter of 54 mm, which is slightly less than the 60 mm diameter of the tube section in which the turbine is housed. The latter tube is used since this is a section of the lab set-up, which ensures that the turbine performance can be compared for an identical turbine and its mount. To connect the tubes with a different diameter, a rounded transition is made in the flange that connects the sections. In Sec. 6.5.1, it is shown that this transition causes some unwanted acoustic losses, which are accounted for in a single experiment to show the influence thereof.

To measure the input power to the turbine, the acoustic power is measured before and after it. This is done by using four pressure sensors (P1-P4) that have a range of 0 bar to 20 bar to deal with the varying pressure experiments. Using sets of two pressure sensors at the given distance, the local acoustic power is calculated as described in Sec. 3.3. As shown in Fig. 6.1, a thermocouple is used to measure the temperature in the middle of the pressure sensors, such that the local gas properties for the acoustic power calculation can be accurately determined. Even though the turbine is placed at the ambient side of engine #1, the difference in temperature between both sides

of the turbine can easily reach 5 °C. With the measurement of the acoustic power difference over the turbine, ΔE_2 , and the shaft power produced by the turbine, the efficiency is calculated as: $\eta_t = P_m / \Delta E_2$. Following Sec. 4.4, a function that uniquely determines the turbine efficiency is found by scaling with the thermoacoustic input coefficient, C_{ta} , as follows:

$$C_{ta} = \frac{\Delta E_2^*}{\phi^3 + \phi^2}, \quad (6.1)$$

where ΔE_2^* is the dimensionless acoustic power drop over the turbine and ϕ is the flow coefficient.

During the experiments under varying operating conditions, it is found that the pressure amplitude ranges from about 2 kPa to 20 kPa. Especially for the lowest amplitudes, this is quite small with respect to the resolution of the pressure sensors. This was found to be an issue for the phase difference between two adjacent pressure sensors, which initially varied significantly, resulting in an inaccurate calculation of the acoustic power. To deal with this issue, acoustic waveforms are sampled for 4 s instead of 1 s, to more accurately determine the phase information. Furthermore, by increasing the input temperature of the hot oil relatively slowly, enough phase data is acquired to make an accurate fit of the phase difference between the pressure sensors as a function of the measured amplitude. This phase information is then used to calculate the acoustic power with sufficient accuracy. The latter is concluded from the fact that the results were found to be repeatable and measuring with either increasing or decreasing oil temperature resulted in no significant difference when measuring slow enough. As an indication, each experiment presented in this work takes approximately one hour, in which the oil temperature is slowly raised from about 100 °C to 200 °C. The measurement points presented in this work are an average of twenty segments acquired by 4 s of sampling. The standard deviation of these averaged results is typically no more than 0.4 % in turbine efficiency, which also decreases for increasing pressure amplitudes due to the relatively better resolution of the measurements. A typical measurement with this standard deviation as an error bar will be presented as the first result in Sec. 6.5.1.

6.3 Thermoacoustic model

In the previous section, it is shown where the bidirectional turbine is eventually implemented in the thermoacoustic refrigerator. Since it is not straightforward to do this in thermoacoustic devices without harming its performance, this location has been determined by using a thermoacoustic model that can predict the performance of a system for a given design. In this section, details about the thermoacoustic model are presented. In Sec. 6.3, the model is validated with experiments and subsequently used to determine the aforementioned location for the bidirectional turbine. In other work, a more

Figure 6.2: A cascade of two acoustic two-ports in series, where the incident waves are denoted by i , and the reflected waves by r . The s -parameters of the segments (a) and (b) fully determine the transmission and reflection of each two-port.

elaborate description of the model¹³⁷ and its use for a practical application¹⁹ can be found.

A successful model should be able to predict the pressure and flow rate in the entire device as a function of the temperature at the heat exchangers, where the latter can be regarded as boundary conditions. This can be done by splitting the entire system into smaller segments, for which a local representation is determined. This is often done by using an electrical analogy, where the pressure is regarded as a voltage and the flow rate as a current. For simple systems, each segment can then be described as a combination of electrical components, such as resistors, capacitors, and inductors. By combining all segments into an electrical network, the system can be solved to find the pressure and velocity at all locations. However, the usefulness of such a method declines with the increased complexity of a system, especially when the segments are large with respect to the acoustic wavelength, such as for the feedback tubes.¹³⁷

Similar to the electrical analogy, the model used in this work uses a lumped system of circuit elements, but here the individual segments are modeled with acoustic two-ports. The advantage of such a description for a segment is that only the transmission and reflection of the acoustic wave are of importance, while the inner workings can be regarded as a black box. This means that even if not all sections can be modeled analytically, the overall system can still be solved by experimentally measuring the transmission and reflection of the unknown segments individually. The latter is enough to fully characterize the elements, and this approach remains valid for segments that are large with respect to the acoustic wavelength.¹³⁷

To describe the use of acoustic two-ports in more detail, two of them are depicted in Fig. 6.2 along with the local wave components. The acoustic two-ports are fully described by the four complex s -parameters which are frequency dependent. For each acoustic wave reaching a two-port (from either side), the s -parameters determine how much of the wave is transmitted and reflected. For example, for the incident wave at the first two-port (i_1), the parameter s_{11a} determines the reflected part and s_{21a} the transmitted part. Similarly, for reflected wave (r_2) reaching this two-port from the right hand side, s_{22a} determines the reflected part and s_{12a} the transmitted part. By combining these effects, the local wave components around two-port (a)

are described by a simple system of equations:

$$r_1 = s_{11a}i_1 + s_{12a}r_2, \quad (6.2)$$

$$i_2 = s_{21a}i_1 + s_{22a}r_2. \quad (6.3)$$

By rewriting these equations, they can be given in matrix-vector form as follows:

$$\begin{bmatrix} i_1 \\ r_1 \end{bmatrix} = \frac{1}{s_{21a}} \begin{bmatrix} 1 & -s_{22a} \\ s_{11a} & -\Delta s_a \end{bmatrix} \begin{bmatrix} i_2 \\ r_2 \end{bmatrix} = [\mathbf{S}_a] \begin{bmatrix} i_2 \\ r_2 \end{bmatrix}, \quad (6.4)$$

where $\Delta s_a = s_{11a}s_{22a} - s_{21a}s_{12a}$. This matrix-vector form is convenient because it allows one to simply cascade multiple two-ports. By applying the same method for two-port (b), the incident and reflected waves on both sides of the cascade are described by:

$$\begin{bmatrix} i_1 \\ r_1 \end{bmatrix} = [\mathbf{S}_a] [\mathbf{S}_b] \begin{bmatrix} i_3 \\ r_3 \end{bmatrix} = [\mathbf{S}_{ab}] \begin{bmatrix} i_3 \\ r_3 \end{bmatrix}, \quad (6.5)$$

where $[\mathbf{S}_{ab}]$ follows from matrix multiplication and determines the properties of an equivalent two-port for the entire cascade. Since this procedure can be repeated for an arbitrary amount of two-ports, the entire thermoacoustic device can be modeled by cascading all segments of the system. Note that from the components of the determined incident and reflected waves, the local pressure and flow rate can be found at all junction planes.¹³⁷

To solve the system of two-ports that models the thermoacoustic device, only the s-parameters for all segments have to be known. For simple elements they might be determined analytically, but since the s-parameters hold a physical meaning, they can also be measured.^{137,138} For this work, only the segment with the bidirectional turbine has to be added to the existing model of the thermoacoustic refrigerator. This has been done by implementing the turbine two-port as a function of the frequency and the acoustic power absorbed by the turbine, as measured during the lab experiments. With all the complex s-parameters known, and the given boundary conditions in the form of temperatures at all regenerators, the system can be solved. This is done by iteratively determining the correct pressure and flow rate amplitudes for which the equivalent two-port of the entire system is in balance. In other words, the operating point is found for which the acoustic power that is added to the system at the engines stages, balances with the energy subtracted from the system by the heat pumps, the turbine, and the acoustic losses. Note that this is a solution to the thermoacoustic system, thus it relates the acoustic power to the internal temperatures. By using the measured performance of the gas-fluid heat exchanger as a function of the acoustic velocity and fluid flow rate through it, this is converted into the external temperatures of the fluid circuits. In this latter form, the model calculates the performance of the system in parameters that are practically relevant and can be measured for the actual thermoacoustic device.

6.4 Turbine implementation

This section will focus on validating the thermoacoustic model with experimental results, and subsequently using the model to implement a bidirectional turbine in a thermoacoustic refrigerator. The success of implementation is quantified by looking at the performance of the complete device. For a given input temperature of the hot oil, a higher level of acoustic power running through the machine is regarded as a better implementation of the turbine. In Sec. 6.5, the performance of the turbine itself will be presented for the best implementation that is identified in this section.

As a benchmark for the turbine implementation, the performance of the refrigerator without a turbine is calculated using the thermoacoustic model and measured experimentally. The resulting acoustic loop power as a function of the input temperature of the oil is presented in Fig. 6.3. Just as for all other cases in this section, the given results are for air with a mean pressure of 10 bar. The first thing that can be seen for the case with no turbine is that the model matches the experimental results relatively well. The two things that should mainly be examined are the onset temperature at which the first acoustic power is produced, and the slope of the acoustic power curve for subsequently increasing temperature. Both of these characteristics also hold a physical meaning, where the onset temperature indicates how well the device is acoustically matched, while the slope is a measure for the acoustic losses in the system. The latter also includes the acoustic power that will be absorbed by the turbine, which is why the slope of the curves with a turbine will not be as steep as the ones without. However, ideally the acoustic matching for a device with a turbine is such that the onset temperature of 70 °C for the case with no turbine is still reached.

For initial tests, the 700 mm feedback tube including the bidirectional turbine is implemented with the turbine near the second heat pump (see HP #2 in Fig. 6.1). All other feedback tubes have been kept at the normal length of 600 mm. Using this configuration, it is found that the bidirectional turbine can indeed produce electric power inside the thermoacoustic refrigerator, yet only at a low level. It can be seen in Fig. 6.3 that both the model and experiments show that the onset temperature of the device is significantly increased due to the implementation of the turbine. The experimentally measured 125 °C onset temperature is indicative of bad acoustic matching, and should be improved by implementing the turbine differently. What is interesting to see from the experimental results though, is that the slope of acoustic power increase is similar to the case of no turbine at the moment when the turbine is not running yet. Once there is sufficient power such that the turbine self-starts, the acoustic loop power drops and the slope decreases since the turbine is absorbing acoustic power (and converting it into electricity). The slope is now approximately equal to the predicted value by the model, which doesn't account for starting behavior of the turbine.

Figure 6.3: Acoustic power in front of the turbine section as a function of the oil input temperature. The solid lines represent results from the thermoacoustic model, while the markers denote experimental results. The bullets (●) represent a working turbine and the square markers (■) are for the turbine either not running yet or not being present at all. The legend denotes the turbine location, which is just after the second heat pump (HP #2) or in front of engine #1, as depicted in Fig. 6.1. The legend further describes the length of the feedback tube between engine #1 and #2 (without turbine), which is either the normal 600 mm or the longer 700 mm length.

With some good indications that the thermoacoustic model can be used to predict trends in the performance of the device, it is used to search for a better implementation of the turbine. An evident attempt to improve the acoustic matching is to account for the extra tube length introduced by the turbine. It can be seen in Fig. 6.3 that the predicted onset temperature is significantly lower when a longer feedback tube is introduced between engine #1 and engine #2. Besides this, the location of the turbine and the extra feedback tube have been varied in the model to find the optimal configuration. The latter was found by keeping the additional longer tube in place, while moving the turbine to the position just before engine #1, as it is depicted in Fig. 6.1. It is experimentally shown that this implementation is indeed significantly better than the initial attempt, since the onset temperature can now be extrapolated to be approximately 85 °C. Note that this is not yet on par with the case of no turbine, which is most likely caused by the presence of the turbine in only one of the long tubes, still resulting in some acoustic mismatch. This might be solved by using a dummy load in the feedback tube between engine #1 and #2, but this will also introduce additional acoustic losses that lower the slope. Therefore, the configuration of the device is kept as shown in Fig. 6.1 in the remainder of this work, which will ensure enough acoustic power for

the turbine to be investigated for a large range of operating conditions.

The results presented in this section indicate that the thermoacoustic model is a useful tool in predicting the performance of a device in question. Even though the absolute value of the acoustic power can still vary significantly, it is shown that the predicted trends are in accordance with experimental results. The latter makes the model sufficiently useful, since the main purpose of the model is only to guide in designing and optimizing thermoacoustic devices. Another useful property of the model is to predict the operating frequency of the machine. For the given turbine implementation, the frequency with air as a working fluid is calculated to be 23.4 Hz, which is confirmed by the experiments. Predicting the operating frequency is especially useful to determine the ratio of gas mixtures, without having to accurately measure how much gas is added. Since the speed of sound is different for air, helium, and argon, which will be used in this work, it is possible to determine the mixture ratio of two gases from the experimentally measured frequency. The gas mixtures that are used in the following section are determined in this manner. Finally, it is worth noting that both the model and experimental results for different gas types provide the same conclusions about the turbine implementation as presented for air in this section.

6

6.5 Results

In this section, the performance of the implemented turbine is investigated for a wide range of operating conditions. The pressure of the working fluid will be varied from 10 bar down to 3 bar, which is the limit at which there is still sufficient acoustic power available for the turbine. Sweeps of increasing oil temperature are performed for several generator loads at each pressure. Furthermore, helium, argon, and an air-helium mixture are used as a working fluid besides air. As described in the previous section, the ratio of the air-helium mixture is determined from the measured frequency of 36.5 Hz, yielding a mass ratio of 40 % air and 60 % helium. Note that for the full helium and argon measurements, there is still air at ambient pressure left in the device when filling, which accounts for approximately 10 % of the mass. This is accounted for in calculating the acoustic power, but for clarity the mixtures are simply referred to as pure helium and argon. For the varying operating conditions, Sec. 6.5.1 will start with examining the turbine efficiency as a function of the acoustic power absorbed by the turbine. Subsequently, it is investigated whether scaling the performance using Eq. 6.1 still holds for varying gas types and pressures. In Sec. 6.5.2, the refrigerator and turbine combination is considered as a whole, looking at the amount of acoustic power absorbed by the turbine and thus how much is left over for refrigeration purposes.

Figure 6.4: Turbine efficiency as a function of the acoustic power absorbed by the turbine. The legend depicts the operating conditions, where the corrected results represent turbine efficiency as calculated when incorporating a correction for the acoustic losses at the tube transitions. The error bars for the normal results depict the standard deviation of averaging twenty measurement points for each given turbine efficiency.

6.5.1 Turbine performance

Fig. 6.4 presents the turbine efficiency for a measurement with helium at the given pressure and generator load, along with the standard deviation of the averaged results as an error bar. It can be seen that the maximum turbine efficiency is approximately 32 %, which is significantly less than the 36 % that was measured under lab conditions for air at 1 bar. This difference could be caused by the different operating conditions, but later in this section it is shown that this is not the case. The fact that the measured efficiency is lower is actually caused by the transition from a 54 mm tube to a 60 mm tube in the flanges around the turbine section (see Fig. 6.1), which were not present in the lab set-up. This transition introduces some additional acoustic losses, which are incorrectly ascribed to the absorbed acoustic power of the turbine since the measurement is done in the 54 mm tubes. To identify the magnitude of these losses, the experiment is repeated without a turbine using the same set-up. The loss in acoustic power between the upstream and downstream measurement is found to linearly increase as a function of the acoustic loop power, and reach approximately 2 W at a loop power of 100 W. By subtracting the acoustic losses as a function of the current loop power, the turbine efficiency is recalculated and shown in Fig. 6.4 as well. With this correction, the maximum turbine efficiency is now just above 36 %, which is in the same range as the previous measurements again. Nevertheless, the

Figure 6.5: Turbine efficiency as a function of the acoustic power absorbed by the turbine. The legend depicts the operating conditions for the different measurements with helium.

6

results in the remainder of this work are presented without such a correction, since it is not deemed necessary for comparing the turbine efficiency as a function of varying operating conditions.

Following the previous measurement with helium, it is interesting to see how increasing and decreasing the mean pressure in the device will influence the turbine efficiency. In Fig. 6.5, the turbine efficiency is given for measurements with three different mean pressures using helium. It can be seen that the maximum turbine efficiency remains constant at 32 % when decreasing the mean pressure from 7.2 bar to 4.8 bar. The maximum turbine efficiency also doesn't vary when further increasing the mean pressure and changing the load to the generator. From this it can be concluded that the mean pressure in the device does not affect the maximum turbine efficiency. This is in contrast with the rising efficiency for increasing mean pressure shown in on other work (and Fig. 2.7),⁶² where a 76 % turbine efficiency is reached at 10 bar. Correspondence about the measurement procedure from this other work has provided additional details that are not available in literature. This revealed that the acoustic power in front of the turbine was not directly measured, but estimated from the pressure amplitudes in the vessels using a model similar to the one presented in Sec. 6.3. Furthermore, assumptions are made about the amount of acoustic power that is absorbed by the turbine, while this is heavily dependent on the operating conditions, as will be shown in Sec. 6.5.2. Due to these estimations, it is likely that the measurements in the current work provide more realistic results, since the acoustic power is directly measured on both sides of the turbine. Therefore,

Figure 6.6: Turbine efficiency as a function of the acoustic power absorbed by the turbine. The legend depicts the operating conditions for the different measurements with helium, air, argon, and an air-helium mixture.

it can be said that the maximum turbine efficiency is actually not dependent on the mean pressure of the working fluid.

Since it is determined that there is no influence of the generator load and the mean pressure, it is allowed to vary these in an attempt to reach a maximum in turbine efficiency, while still making a fair comparison between different working fluids. The results for such a study are given in Fig. 6.6, where the turbine efficiency is given for measurements with helium, air, argon, and the air-helium mixture for varying generator loads and mean pressures. It can be seen that, despite the wide range in operating conditions, the maximum turbine efficiency is still around 32 % for all cases. Therefore, it can be concluded that the type of working fluid also has no influence on the maximum turbine efficiency. Note that the acoustic frequency varies from 22 Hz for argon to 53 Hz for helium, thus showing that the frequency also has no influence on the maximum turbine efficiency, confirming what was shown in Sec. 4.3 for a single gas type and pressure.

Besides the maximum turbine efficiency, these lab results showed that over the entire measurement range, the turbine efficiency can be scaled uniquely with the thermoacoustic input coefficient, as given in Eq. 6.1. Since the present work also varies the gas type and the mean pressure, next to the acoustic frequency and generator load, it is interesting to see whether this scaling still holds. Fig. 6.7 presents the turbine efficiency as a function of the thermoacoustic input coefficient for all results that have been presented so far, thus including the measurement for which the acoustic power is corrected. This latter result is interesting since it can be compared with the

measurement from Chapter 5 under lab conditions, which is also shown in Fig. 6.7. Focusing first on the results from this work, it can be seen that scaling with the thermoacoustic input coefficient works relatively well. Considering that for all operating conditions there is a wide spread in turbine efficiency as a function of ΔE_2 , using C_{ta} as a performance indicator provides an almost unique function of the turbine efficiency. However, it is hard to conclude whether this scaling exactly holds, since a discrepancy of a few percent in C_{ta} is clearly visible between the different results. By checking additional measurements that are not shown in this work, no consistent trend in either gas type or pressure was found. This leaves the acoustic power losses in the tube transition as a possible explanation for the slight differences in scaling. Comparing the measurement with corrected acoustic power to the lab results, quite a good agreement is found in turbine efficiency, especially since the gas type, mean pressure, generator load, and acoustic frequency are all different for these measurements. This shows that the thermoacoustic input coefficient can likely still be used to find the maximum turbine efficiency for any operating conditions. More confidence in such a conclusion could be provided by similar measurements that do not need any correction for the acoustic power.

Figure 6.7: Turbine efficiency as a function of the thermoacoustic input coefficient. Details about the operating conditions of the different measurements can be found in the legends of Fig. 6.4, 6.5, and 6.6. The blue and dotted line with diamond markers ($\diamond$) represents the measurement with air at 1 bar and 10 Ohm generator load from Chapter 5.

6.5.2 Combined performance

The results presented so far have purely focused on the turbine, without regarding the entire device under varying operating conditions. For the latter, not only the efficiency of the turbine is of importance, but also the amount of acoustic power that is absorbed with respect to the total amount available. This is especially true for a combination such as the present device, since any acoustic power that is absorbed by the turbine is not available for refrigeration purposes anymore. In the following, the acoustic power absorption is given for a varying load, mean pressure, and gas type, after which the power distribution in the entire device is examined under typical thermoacoustic conditions.

In Fig. 6.8(a), the ratio of the absorbed acoustic power over the total amount of loop power is given for a varying load to the generator. First of all, it can be seen that for a given load, relatively more acoustic power is absorbed as the loop power increases. Secondly, for an available amount of acoustic loop power, the turbine absorbs more as the resistance to the generator is increased. Note that increasing the resistance means that the load to the turbine is actually reduced, thus increasing the RPM for given operating conditions. To explain the aforementioned two effects in acoustic power absorption, the real part of the normalized turbine impedance is presented in Fig. 6.8(b) for the same measurements. The real part of the turbine impedance can be regarded as the flow resistance, and it can be seen that it has a clear correlation with the amount of acoustic power that is absorbed. In Sec. 4.5, it was shown that increasing the turbine RPM causes a larger real part of the turbine impedance, and therefore more acoustic power absorption. Since the RPM will increase for both a larger acoustic loop power and an increased load resistance for a given loop power, the larger associated impedance explains why more acoustic power is absorbed by the turbine for such conditions.

This section started by presenting results for a varying generator load, since this is the main way to vary the absorbed acoustic power during operation. This is typically necessary when the thermoacoustic device does not operate at a single point, for example due to fluctuations in the (low-grade) heat source or the ambient temperature. However, as can be seen in Fig. 6.8(a), the difference in absorbed acoustic power by the turbine is only a few percent between the largest and smallest load. Furthermore, to ensure a high efficiency, the flow conditions should be such that the turbine operates around the right thermoacoustic input coefficient (see Fig. 6.7). Due to this limited range of tuning possibilities during operation, it should be made sure that the acoustic power absorption is already around the required value. When large changes are needed for the latter, the turbine can be placed in an entirely different impedance region of the device, such as a perpendicular tube section often referred to as a stub. Furthermore, the gas type and

(a)

(b)

Figure 6.8: The ratio of absorbed acoustic power over input power (a) and the real part of the normalized turbine impedance (b) as a function of the acoustic loop power in front of the turbine. Results are presented from the point where the turbine absorbs at least 5 W acoustic power. The legend depicts the generator load for the measurements with helium at 10.0 bar mean pressure.

Table 6.1: The real part of the normalized turbine impedance, $\text{real}(z_t)/z_0$, and the ratio of absorbed acoustic power over input power, $\Delta E_2/E_{2,in}$, for varying operating conditions. Results are given at 50 W acoustic loop power for a varying gas type, mean pressure, P_m , and generator load, R.

Gas	P_m [bar]	R [Ohm]	$\text{real}(z_t)/z_0$	$\Delta E_2/E_{2,in}$
Helium	10.0	0.75	0.09	0.14
	10.0	2	0.10	0.15
	10.0	4.7	0.12	0.16
	10.0	10	0.14	0.17
Air	9.6	2	0.18	0.19
	7.1	2	0.21	0.21
	5.4	2	0.24	0.24
	3.4	2	0.26	0.28
Air-Helium	9.8	2	0.14	0.18
Argon	9.8	2	0.19	0.21

the mean pressure can also have a large influence on the absorbed acoustic power, as will be shown for a single turbine position in the following.

Table 6.1 presents the absorbed acoustic power ratio and real part of the normalized turbine impedance for varying generator load, mean pressure and gas type. The results are given at an acoustic loop power of 50 W, which is chosen since it is the largest power that could be reached for 3.4 bar air. The values for the varying generator loads at 10 bar helium correspond to the measurements presented in Fig. 6.8. Just as for these results with a varying load, it can be seen from Table 6.1 that changes in the absorbed acoustic power for the other measurements again correlate with the real part of the turbine impedance. When reducing the mean pressure of air from 9.6 bar to 3.4 bar, the ratio of absorbed acoustic power increases from 0.19 up to 0.28. This shows that the mean pressure of the device has a big influence on the relative amount of acoustic power that is absorbed, while in Fig. 6.5 it is shown that there is no influence on the maximum turbine efficiency. Note that although thermoacoustic devices will generally not operate at these smallest pressures, they do often run at a mean pressure significantly larger than 10 bar, which would result in a smaller amount of absorbed acoustic power. The remaining results that are given in Table 6.1 are for varying gas types with 2 Ohm generator load and a mean pressure of approximately 10 bar. The results show that the ratio of absorbed acoustic power increases from 0.15 for helium up to 0.21 for argon, with the values for the air-helium mixture and pure air in between. Since helium has the smallest characteristic impedance (z_0), followed by air and finally argon, this indicates that relatively more acoustic power is absorbed as the characteristic impedance of the working fluid increases.

Now that the acoustic power absorption and the performance of the turbine has been identified for a wide range of operating conditions, there is enough information to apply this to practical use cases. As an example, a case study is presented in the following for the thermoacoustic refrigerator with electricity production that is treated in this work. The amount of electricity produced by the turbine will be set such that it can exactly power the three pumps that are needed for the fluid circuits of the heat exchangers. This provides an off-grid system, for which in this case the remaining acoustic power will all be used to provide refrigeration in the heat pumps. Note that, similar to the current study, one can also use the remaining acoustic power to (partly) produce additional electricity.

The case study is based on the measured values for the air-helium mixture at 9.8 bar mean pressure with a 2 Ohm generator load. To provide an energy balance of the system, the input power of the heat source, Q_{in} , and the cooling power, Q_{cold} , have to be known. Since these are not measured during the turbine experiments, the input temperature and acoustic loop power are used to estimate these from measurements for a similar, yet significantly larger, 4-stage thermoacoustic refrigerator. For the cooling power, this is done by using the acoustic power that is left after the turbine has used enough to produce electricity for the pumps. Since the latter is around 20 W, there is 62 W of acoustic power needed for the turbine when assuming a generator efficiency of 90 % and a turbine efficiency of 36 %. From the measurement with the air-helium mixture, it is found that 31 W of acoustic power is absorbed by a single turbine when the input temperature is around 200 °C. Therefore, using two turbines will provide exactly enough electricity for the pumps at this operating point, while using two turbines will also have the added benefit of a better acoustic matching (as described in Sec. 6.4). For the other input temperatures that are considered in the case study, it is assumed that one or several turbines can be used such that they absorb 62 W of acoustic power as well.

Table 6.2 presents the results of the case study, where the performance of the off-grid thermoacoustic refrigerator is given as a function of the input temperature of the heat source. The first thing to notice is that an input temperature of at least 163 °C is needed to run the system. For this input temperature and corresponding input power, just enough acoustic power is produced such that the turbines can power the pumps. Since the pump power does not change, increasing the input temperature from this point onwards will provide a surplus of acoustic power that can be used for refrigeration. A way to quantify the refrigeration performance is to look at the ratio of cooling power over the input power, which is defined as the coefficient of performance (COP) here. For 175 °C input temperature the COP is only 0.12, but it can be seen that this quickly rises with increasing temperature. As a comparison, it is shown in Table 6.2 that for a 200 °C input temperature a COP of 0.53 is acquired when all acoustic power is used for refrigeration

Table 6.2: Performance of the thermoacoustic refrigerator as a function of the input temperature of the heat source. The acoustic loop power, $E_{2,in}$, is derived from the measurements for the air-helium mixture at 9.8 bar with a 2 Ohm load. With 62 W of loop power absorbed by the turbine, 20 W of electricity is produced to run the pumps. The remaining acoustic power is converted into cooling power, Q_{cold} . With the input power from the heat source, Q_{in} , the coefficient of performance (COP) is defined as Q_{cold}/Q_{in} .

T_{in} [°C]	Q_{in} [W]	$E_{2,in}$ [W]	Q_{cold} [W]	COP
163	249	62	0	0
175	285	83	34	0.12
200	360	120	93	0.26
225	435	158	154	0.35
250	510	195	213	0.42
200	360	120	192 ¹	0.53 ¹

(in which case external power is of course needed for the pumps). In the off-grid version, approximately half of this performance is still reached at 200 °C. Furthermore, as the input temperature is further increased, the pump power becomes less relevant, and the COP of the off-grid version will converge towards the refrigeration only version. What will remain is the added capital costs of the machine due to the additional components producing electricity. So depending on the specific use case, it should be determined whether the added cost and complexity of the off-grid system is worth it. This might especially be the case for remote locations where the electricity grid is not stable or not even present. For such cases, where the possibility of running off-grid is critical, a refrigerator working purely on a heat source could be much preferred over conventional systems.

6.6 Conclusions

To implement a bidirectional impulse turbine into a thermoacoustic refrigerator, a model is presented that can predict the acoustic power in the device as a function of the input temperature. The thermoacoustic model is experimentally validated for configurations both with and without a turbine. Subsequently, the model is used to implement the turbine at a good location in the device, hereby focusing on the acoustic matching quantified by the onset temperature where oscillations start. For the identified turbine implementation, sufficient acoustic power can be generated by the device to investigate the turbine under a wide range of operating conditions. In future work, more effort should be taken to implement the turbine with a better acoustic matching, which can for example be done by adjusting more feedback tube sizes or

¹Value when no turbine is present and the pumps are driven by external power.

introducing a second turbine to form a matching pair.

For the implemented turbine with helium as a working fluid, it is shown that electric power can indeed be produced in the thermoacoustic refrigerator. After correcting for acoustic losses at two flange connections, the maximum efficiency of the turbine is found to be 36 %, which is equal to the measured efficiency for the same turbine in previous lab experiments.

The mean pressure in the device is varied, which resulted in no significant change in the maximum turbine efficiency. This is found to be in contrast with other work, where the efficiency drastically increases as a function of the mean pressure. It is concluded that the results from this work are most likely correct, since the acoustic power to the turbine was more correctly measured in a direct manner. Besides the mean pressure, the generator load and the gas type were also varied. For helium, air, argon, and an air-helium mixture, the maximum turbine efficiency was found to be approximately the same for all pressures and generator loads. This constant maximum efficiency is convenient when designing a thermoacoustic engine, since the gas type and mean pressure can be chosen such that they are optimal for the rest of the device. Furthermore, this shows that the bidirectional turbine can be further optimized under lab conditions, after which the same performance can be expected with actual thermoacoustic conditions.

For all measurements under varying operating conditions, it is shown that the turbine efficiency is approximately a unique function of the thermoacoustic input coefficient. Furthermore, for the measurement with corrected acoustic power, a good agreement is found with the results from Chapter 5 under drastically different conditions. This indicates that this scaling can be used to identify the point at which the turbine operates at its maximum efficiency. Besides the most efficient point, it is shown how much acoustic power the turbine absorbs for the varying operating conditions. The acoustic power absorption is shown to be directly correlated with the real part of the normalized turbine impedance. By taking both the measures for the maximum turbine efficiency and the absorbed acoustic power into account, the turbine can be correctly implemented in a thermoacoustic device.

A case study is given which shows such an implementation, where the turbine is used to produce enough electric power to drive the fluid pumps, while the remaining acoustic power is used for refrigeration. The threshold for operation of the given device is found to be 163 °C for the temperature of the heat source. As the input temperature is increased above this value, the amount of cooling power and the coefficient of performance rise, while the acoustic power needed for the turbine to drive the pumps becomes increasingly less relevant. The given case study shows that a thermoacoustic refrigerator running purely on a heat source is a viable option, especially when there is a need for off-grid operation.

7

Conclusions and recommendations

The work in this thesis concerns the use of a bidirectional impulse turbine for thermoacoustic devices. A review on all acoustic to electric transducers has shown that the turbine is a promising option, but that there is a lack of literature on the application in thermoacoustics. Therefore, it is attempted in this work to build a solid foundation for the use of bidirectional turbines in thermoacoustic devices. For this purpose, the turbine performance is characterized and optimized under lab conditions, after which it is implemented in an actual thermoacoustic device and tested for a wide range of operating conditions.

In this chapter, a summary of the conclusions from the individual topics, along with some more general conclusions, are given in Sec. 7.1 through Sec. 7.4. After this, suggestions for future work are given in Sec. 7.5, and the potential areas where this work can be applied are discussed in Sec. 7.6.

7.1 General acoustic to electric

A review is given on the main methods to convert thermoacoustic power into electricity. For each option, design aspects, operating characteristics, and methods to calculate and optimize the performance are given. To efficiently implement any transducer in a thermoacoustic device, it is shown to be of vital importance to ensure good acoustic matching. The conclusions of the individual methods to convert thermoacoustic power into electricity are as follows:

- Linear alternators, or more generally electromagnetic transducers, are the most widely used and mature way to produce electricity in thermoacoustics. The conversion efficiency can reach up to 75 %, and power output up to the kW range has been shown. However, scaling to a larger power output is shown to be troubling, and the linear alternators are expensive in comparison with the rest of a thermoacoustic device.
- Piezoelectric transducers can operate under relatively high frequencies, which is useful for compact thermoacoustic devices. The maximum efficiency is around 20 %, while most practical implementations have only shown a very small power output ($\sim$ mW) in thermoacoustics.
- Magnetohydrodynamic devices have no mechanically moving parts, which potentially makes them reliable and suitable for inaccessible environments. Although the numerically predicted performance is fairly good, there is a lack of experimental confirmation. The main problem in practice is the efficient power transduction across the gas-liquid interface of the transducer.
- Bidirectional turbines are relatively cheap and simple to implement, while scaling to a power output in the MW range is shown to be possible in oscillating water columns (OWCs). Initial indications have shown that a bidirectional turbine can be as efficient as a linear alternator, but there is a lack of experimental validation to confirm this.

Since an axial impulse turbine is potentially the most attractive alternative to a linear alternator, for both increasing size and reducing costs, it is studied in both lab conditions and a thermoacoustic device to confirm it is indeed a viable option.

7.2 Turbine characterization

To properly characterize the performance of a bidirectional turbine, especially for small changes in operating conditions, accurate measurements are desired. By including the atmospheric conditions and frequency dependent phase information of the pressure sensors, the experimental results show a good repeatability and reproducibility. For a turbine with 25 % efficiency, the maximum error is estimated to be around 0.5 %, which is sufficiently small for the studies performed in this work.

The characterization of the bidirectional turbine is started by examining its velocity diagram. Due to the oscillating nature of the acoustic flow and the need for guide vanes on both side of the rotor, the flow angles are far from optimal during most of the period. Besides this, it is shown that the turbine efficiency does not uniquely depend on the flow angles, and therewith the flow coefficient, which is in contrast to turbines in steady flow or OWCs. The

main reason for this discrepancy is the variable phase difference between the pressure and velocity of the acoustic flow. Furthermore, there is also a phase difference of these properties over the turbine itself. Both these phase differences suggest that a quasi-steady approximation of the system, as is used to study the bidirectional turbine in OWCs, might not be valid. This raises the need for a characterization of the turbine that is specific for thermoacoustic conditions.

A performance indicator that incorporates the phase difference between pressure and velocity is the acoustic power drop over the turbine, which is shown to be a good measure to examine the turbine efficiency for varying operating conditions. By changing the load to the turbine, a different turbine efficiency as a function of the acoustic power drop is found, but the maximum efficiency is around 25 % for all cases. For the acoustic frequency, no influence on the turbine efficiency is found in the range of 50 Hz to 80 Hz. This independence means there is no need to match the frequency of a thermoacoustic device with the turbine for efficient performance. This is an advantage over linear alternators, for which it can be difficult to reach a high efficiency since it has to be driven around its mechanical resonance frequency.

In an effort to find a unique function of the turbine efficiency which is valid for all operating conditions, a dimensional analysis is performed. The relevant dimensionless groups are found to be the dimensionless acoustic power drop, flow coefficient and Reynolds number, of which the latter is experimentally found to be of a negligible influence. By combining the former two, a thermoacoustic input coefficient is defined, which is shown to provide a unique function of the turbine efficiency for all investigated operating conditions. Finally, the complex impedance of the turbine is presented, such that it can be used for implementing the turbine in a real thermoacoustic system.

7.3 Turbine optimization

Several design alterations are tested in an attempt to increase the efficiency of the bidirectional turbine. Adding a 1 mm thick shroud ring around the rotor blades, while keeping a constant tip clearance, increases the efficiency from 25 % to 28 %. By reducing the ring thickness to 0.5 mm, more blade area is created at a large radius, therewith further increasing the efficiency by 2 %. Shrouded rotors are more efficient than the unshrouded ones since they block the radial leakage flow over the tips of the blades. This effect is shown to be especially big for a large tip clearance, where the shrouded rotor provides an efficiency that is almost twice as high. For a small tip clearance, the efficiency of both unshrouded and shrouded rotors increases significantly, reaching approximately 37 % for the most efficient turbine. When reducing the tip clearance there will be a break-even point, at which the unshrouded rotor will become more efficient, as is measured for one case in this work.

This cross over is due to the reduced leakage flow for the unshrouded rotor, while the tip radius of the blades is larger than for the shrouded rotor. While for the current work the shrouded rotors were more efficient in nearly all cases, this shows that for large systems where the relative tip clearance can be made smaller, an unshrouded rotor can be more efficient. Note that the conclusions for the tip clearance measurements with and without a shroud ring concur with experiments done for turbines in steady flow.

Varying the axial spacing between the rotor and the guide vanes has a big influence on the turbine efficiency, which is in contrast to measurements done for OWCs. Especially for a shrouded rotor with a large tip clearance the effect is large, which is explained by the reduced leakage flow between the shroud ring and the outer tube. For a large axial spacing, the maximum turbine efficiency is not constant anymore for a varying acoustic frequency and load, as was found when characterizing the turbine at 1 mm spacing. The deciding factor for this influence is identified as the displacement amplitude of the working fluid, which is a function of the operating conditions. A significant decrease in turbine efficiency is measured when the displacement amplitude approaches the length of the axial spacing, while the maximum efficiency is constant for large enough displacement amplitudes. This shows that, as a function of the operating conditions, there is an upper bound to the axial spacing that should be used for efficient performance.

Starting from the optimum design for OWCs, a study varying the guide vane and rotor blade geometry is performed. No large variations in turbine efficiency are found, suggesting that the current design is also near an optimum for thermoacoustic conditions. The largest influence was found for the angle of the blades, where 60° is identified as an optimum when using the same angle for the guide vanes and the rotor. By increasing the rotor angle to 65° for the same guide vanes, the turbine efficiency can be slightly improved.

When incorporating all measures for optimization given in this work, it is possible to reach a turbine efficiency of 40 %, which is in line with what is achieved for OWCs with a similar turbine design.

7.4 Turbine implementation

With the turbine optimized under lab conditions, it is implemented in a thermoacoustic refrigerator to investigate whether it can work in a real device and reach the same efficiency. To determine the optimum location of the turbine in the thermoacoustic refrigerator, a model based on acoustic two-ports is introduced, and the turbine is implemented with the complex impedance that was measured in the lab experiments. By focusing on the onset temperature of oscillations, which is a measure for the acoustic matching, a good turbine implementation is identified. However, since the onset temperature is not back to that for the thermoacoustic refrigerator without a turbine, there

is room for improvement.

Initial tests show that the bidirectional turbine can indeed produce electricity alongside refrigeration for the chosen implementation. By quantifying and correcting for acoustic losses at two flange connections, the turbine efficiency is found to be 36 %, which equals the results from the lab experiments for the same turbine.

With a correctly implemented turbine, the operating conditions are varied to investigate the influence thereof. While other work has indicated that the turbine efficiency will increase significantly as a function of the mean pressure, the more accurate measurements in this work show that the maximum turbine efficiency is actually constant for such changes. Furthermore, measurements with air, helium, argon, and an air-helium mixture all show the same maximum turbine efficiency as well. Since a different gas type will cause a change in acoustic frequency, and the generator load also shows no influence on the maximum turbine efficiency, the results from the characterization during the lab experiments are confirmed in practice. This independence of the maximum turbine efficiency for gas type, mean pressure, and generator load is very convenient when designing a thermoacoustic device, since this provides some flexibility that will not be present when using a linear alternator. Additionally, further optimizing the performance of the bidirectional turbine can be performed under much more accurate lab conditions, after which the same performance can be expected in the thermoacoustic device.

With more widely varying operating conditions, it is shown that the thermoacoustic input coefficient still provides an approximately unique function of the turbine efficiency. This shows that it can be used to ensure efficient performance of the turbine, *e.g.* by varying the generator load during operation. Besides the point of optimum efficiency, the amount of acoustic power absorbed by the turbine is given as a function of the operating conditions. The real part of the turbine impedance, which can be regarded as the flow resistance, is shown to directly correlate with the acoustic power absorption. For the different operating conditions, between 14 % and 28 % of the acoustic loop power is absorbed. This is an interesting difference with the lab experiments, where nearly all of the input acoustic power is used, because there is a mostly standing wave pattern downstream of the turbine due to the closed end. Despite the turbine placement in standing wave or traveling wave conditions (for the thermoacoustic refrigerator), the performance characteristics of the turbine are found to agree. This confirms that the acoustic power and thermoacoustic input coefficient can be used as general performance indicators for turbines in thermoacoustic devices.

Depending on the use case, a different amount of acoustic power should be absorbed by the turbine. While setting this amount by changing the implementation and operating conditions, it should be made sure that the turbine is driven around its most efficient point, as described by the thermoacoustic

input coefficient. As an example as such an implementation, a case study is presented in which two turbines are used in the thermoacoustic refrigerator. The produced electricity is exactly enough to drive the fluid pumps at a 163 °C input temperature of the heat source. For higher input temperatures, there is a surplus of acoustic power that can be used for refrigeration. At 200 °C a COP of 0.26 is reached, which rises to 0.42 at 250 °C input temperature. Since the required pump power stays the same, the amount of acoustic power needed for the turbine becomes relatively less for increasing input temperatures, resulting in a COP that converges to the case where no turbine is present. The results from this case study show that it is possible to run a thermoacoustic refrigerator purely on a heat source, therewith providing completely off-grid cooling power.

7.5 Future work

The use of bidirectional turbines in thermoacoustic devices is relatively new, and there is only a scarce amount of literature available. Therefore, this work has provided a full story from proof of concept and characterization, up to the implementation of an optimized turbine in a real thermoacoustic device. This should provide a basis for using the bidirectional turbines in thermoacoustics, but due to the big steps that are taken, there is plenty of additional work that can be done for all topics treated in this thesis. In the following, several suggestions for such future work are given.

Computational fluid dynamics (CFD) simulations can be done to further optimize the design of the bidirectional impulse turbine. Initially this was also planned for the current work, but due to the lack of experimental literature it would be hard to validate the simulations. Especially the presence of a rotation domain and turbulence in a high frequency oscillating flow will provide uncertainty in the simulation results. With the large amount of experiments that are performed in this work, there should be enough results in the data publications to validate CFD simulations with. Once an accurate model is present, the design study from Sec. 5.5 should be repeated numerically with larger variations for the blade geometry, such that it can be confirmed that the current design is indeed optimal. Furthermore, in contrast to OWCs, there is a known frequency and amplitude of the wave in thermoacoustics. By numerically studying the flow field in detail, it might be possible to use this fact to invent new and more efficient turbine designs for thermoacoustics. Note that it is undesirable to achieve such goals with elements such as pitching guide vanes, since this will significantly decrease the durability of the turbine due to the high operating frequency. Therefore, passive solutions should be sought that rely on the known acoustic wave properties. If more efficient designs are identified, they should be tested experimentally under lab conditions before implementation, since these measurements can more

accurately confirm whether the new design is indeed better.

A question that has not been completely resolved is whether the acoustic flow for the turbine can be assumed to be quasi-steady or not. This can be further investigated experimentally by having a set-up that can have larger variations in acoustic frequency, starting from steady flow up to a few hundred Hz. However, it is easier to study this numerically, which will also provide the flow field in more detail. Furthermore, since the phase difference between pressure and velocity is also of importance, this can be varied numerically by using a time-domain impedance boundary condition. In this way, the domain can be terminated to ensure a pure standing wave, traveling wave, or anything in between, which will be much harder to do experimentally for varying frequencies.

During the error estimation of the experiments in Sec. 3.5, the largest variations in turbine efficiency were found when using different 3D printing materials and techniques. Since the other materials also ensured a higher efficiency, it is interesting to investigate this in further detail. A special focus should be given to the surface roughness of the materials, since this can for example have an influence on the acoustic losses and flow separation. Besides the material type, it is desired to test turbine prototypes at a larger scale, possibly with a different production technique. At such a size it is possible to reduce the relative tip clearance in an effort to further increase the turbine efficiency. Additionally, it can be checked whether the scaling rules determined in Sec. 4.4 still hold for larger turbines.

This work has only focused on axial impulse turbines, since they seemed the most promising and are relatively easy to implement. As an alternative, the radial version of the impulse turbine can also be studied. The radial impulse turbine might introduce acoustic streaming in looped thermoacoustic devices, but if this issue is resolved, *e.g.* by using a jet pump, it should be able to compete with the axial impulse turbine. Besides this, the reaction based Wells turbine can be used. The downsides of the Wells turbine are that it is not self-starting and it operates at a high RPM, therewith producing more noise. However, it is shown in OWCs that the Wells turbine can outperform an impulse turbine, albeit in only a small range of operating conditions. Initial experiments with a Wells turbine have resulted in an efficiency of only 2 %. Since the scaled design from an OWCs resulted in a really thin 3D print, a thicker version was designed which resulted in a maximum RPM of 15000 and a turbine efficiency of 15 %. This is still not on par with the impulse turbine, but it shows that with additional work this might be achieved. Furthermore, it is interesting that a thicker Wells turbine performed significantly better. This indicates that the inertia of the rotor might be important, which should be investigated in future work. With the impulse turbine, for which the designs were about three to four times heavier than the final Wells turbine, no effects of inertia were noticed.

The implementation of the impulse turbine in the thermoacoustic refriger-

erator should be improved, such that the onset temperature is back to the level where it was without a turbine. Since a single turbine in a 4-stage device causes an inherent asymmetry, this might be resolved by using two or preferably four turbines (one in each feedback tube). The thermoacoustic model shows that this implementation will ensure better acoustic matching, which should be experimentally verified.

By using four bidirectional turbines, the produced electric power is significantly increased. If this is still not sufficient, *e.g.* for a device in which only electricity should be produced, the turbine can be scaled and placed in a larger tube radius. However, since the acoustic velocity will be lower in this case, it might not be possible to reach the maximum turbine efficiency. Alternatively, multiple stages of bidirectional impulse turbines can be placed in a row. In this way, the velocity amplitude is still sufficient, while more power is generated. It should be investigated whether such a multi-stage turbine can work under thermoacoustic conditions, and subsequently if the same conversion efficiency can still be achieved.

The case study presented in Sec. 6.5.2 shows that it should be possible to make a completely off-grid thermoacoustic refrigerator. In future work, this should be confirmed by building such a system. For this purpose, at least a second turbine is necessary, and the electric power should be converted such that it can be used to drive the pumps. Furthermore, a battery is needed to initially start the system, which should also be charged by some additional electric power during operation. With such a system in place, it is interesting to see whether it is possible to continuously run the thermoacoustic refrigerator purely on heat, and whether the generator load can be automatically varied to efficiently run the turbine according to the thermoacoustic input coefficient.

7.6 Practical application

In this section, suggestions are given about where the current work, possibly after future improvements, can best be used in practical applications. First some words are spent on the usefulness of varying device configurations, after which promising areas of application are discussed.

In this work the bidirectional turbine was used in a refrigerator, but it is also possible to make a thermoacoustic engine that produces electricity only. In remote locations where a sufficient heat source is available, such a system might be desired, but in most situations other options probably offer a more efficient or economical way to produce electricity. However, when combined with a useful product such as cooling, the application potential increases significantly. As an example, a 4-stage device can produce about 0.5 kW cooling from 1 kW of heat at 200 °C, while there is 0.3 kW of acoustic loop power. In purely converting the loop power to electricity, only 0.12 kW

of electric power is produced with a turbine efficiency of 40 %. This example shows that it is hard to justify the capital cost for purely producing electricity, while for cooling power the outcome is more positive. Whether for such a thermoacoustic refrigerator a bidirectional turbine should be added to make an off-grid version, will be a case dependent decision. In places with cheap electricity, it is probably beneficial to drive the pumps externally. In locations with a high electricity price such as remote areas, where e.g. a genset is used to produce electric power, the added capital cost of the off-grid version might be earned back relatively quickly (low ROI). In case the turbine is not always needed, for example in locations with an unstable electricity grid, it is an interesting option to place a turbine in a side branch (stub) of a the thermoacoustic device, such that it can be opened only when electric power production is necessary.

When producing electric power besides refrigeration, this inherently reduces the cooling power that is produced, which should be taken into account when calculating the feasibility of a device. Since the linear alternator has a higher efficiency than the bidirectional turbine, it might seem that this is a better option. However, since the linear alternator is approximately an order of magnitude more expensive than the turbine for the same power, it is economically much better to use a turbine. Especially when the input temperature is large, say above 250 °C, the case study in Sec. 6.5.2 has shown that the reduction in cooling power becomes less important. Furthermore, that case study was performed for a relatively old 4-stage thermoacoustic refrigerator. For a newer system with improved heat exchangers, along with a better acoustic matching, the system will already be viable for lower input temperatures.

The most prominent energy source is likely waste heat, since it is widely available and in a way can be regarded as free. When capturing solar heat, e.g. by the solar collectors shown in Fig. 1.1, additional capital costs are needed to provide the heat. Compared with using waste heat, this could double the ROI time in using a thermoacoustic refrigerator instead of an electrically driven vapor compression system. So even though using solar collectors is the most sustainable way of producing cooling from the given options, it is likely the most expensive option if no changes are made in the way energy is priced in our society.

The area in which the most (waste) heat is available is the industry. Since this generally concerns large amounts of heat and the relatively large size of thermoacoustic systems is usually not a big issue, this is a very promising area of application. The capital costs of a system should also not be a problem when the ROI is small enough when comparing with electrically driven cooling. The best locations to introduce a (potentially off-grid) thermoacoustic refrigerator are places where besides available heat there is a cooling need nearby. Examples are metal forgeries and industrial bakeries, but cooling can also be provided to neighboring companies with refrigeration needs. Further-

more, generator sets are often used in industry to have a mobile system of electricity production. With the waste heat from such systems, an off-grid thermoacoustic refrigerator can pre cool the air entering the generator set, therewith significantly increasing its efficiency.

For individual residencies, the current thermoacoustic devices are likely still too big and expensive (although the turbine is compact enough). Even if the price of thermoacoustic coolers can be linearly scaled down with the power, the costs of installing such small systems will not decrease in the same manner. Therefore, the main application is seen in larger systems, such as for apartment buildings where there is sufficient space available and the capital costs are also shared. With that said, further advancements into more compact devices and lower prices once the technology matures, will ensure that smaller residencies will become increasingly more attractive.

For the automotive industry, some waste heat might be available, but the thermoacoustic systems seem just too heavy and bulky. Even with future advancements, it is probably not economically viable to include a thermoacoustic device, since transporting the extra weight will cost additional fuel. For ships this extra weight is less critical, and since enough heat is available from the engines, this is a viable option. The main difficulty to overcome is the lack of available space. It can be seen from the system in Fig. 1.3, which has a diameter of 4.5 m, that the area of a thermoacoustic refrigerator can be quite big. However, the internal volume is not that large, so if the system is neatly integrated into the engine room, the application into ships is a feasible option.

Due to the lack of mechanically moving parts in a thermoacoustic refrigerator, locations where maintenance is hard or impossible are generally interesting. Examples include systems where no down-time can be present or extreme environments such as outer space. Extensive reliability testing should be done to confirm a bidirectional turbine can be included in such system if electricity production is an additional requirement.

With some main topics treated in this section, it should be kept in mind that there are significantly more potential areas of application for the thermoacoustic devices. Most heat sources, even if only 100 °C above ambient, can be utilized by thermoacoustic devices since they already produce power for small temperature gradients and can work for varying ambient and input temperatures. This flexibility makes thermoacoustic devices an interesting option for many areas of application.

List of symbols

Abbreviations

CFD	Computational fluid dynamics	
COP	Coefficient of performance	
DeltaEC	Design Environment for Low amplitude- Thermoacoustic Energy Conversion	
HP	Heat pump	
MHD	Magnetohydrodynamic	
ORC	Organic Rankine cycle	
OWC	Oscillating water column	
ROI	Return on investment	
RPM	Rotations per minute	[1/min]

Roman letters

A	Cross-sectional area	[m ²]
c	Speed of sound	[m/s]
c	Absolute velocity	[m/s]
C_A	Input coefficient	[-]
C_{ta}	Thermoacoustic input coefficient	[-]
D	Diameter	[m]
E_2	Acoustic power	[W]
$E_{2,in}$	Acoustic loop power of thermoacoustic refrigerator	[W]
ΔE_2	Acoustic power drop over turbine	[W]
ΔE_2^*	Dimensionless acoustic power drop over turbine	[-]
L_r	Rotor length	[m]
P	Pressure	[Pa]
P_a	Pressure amplitude	[Pa]
ΔP_a	Pressure drop amplitude over turbine	[Pa]
P_e	Electric power	[W]
P_m	Shaft power	[W]
Q	Heat	[W]
Q_{cold}	Cooling power	[W]
Q_{in}	Heat source input power	[W]

R	Resistance	[Ohm]
R_0	Outer tube radius	[m]
Re	Reynolds number	[-]
R_{hub}	Turbine hub radius	[m]
R_t	Rotor blade tip radius	[m]
R_{tip}	Turbine tip radius	[m]
s	Complex s-parameter of acoustic two-port	[-]
T	Temperature	[K]
T_{in}	Oil input temperature	[°C]
t_a	Rotor gap length	[m]
t_r	Rotor blade thickness	[m]
u_a	Velocity amplitude	[m/s]
U	Flow rate	[m ³ /s]
U	Blade speed	[m/s]
V	Voltage	[V]
w	Relative velocity	[m/s]
x_s	Axial spacing	[m]
z_0	Characteristic specific acoustic impedance	[Pa·s/m]
z_t	Specific acoustic turbine impedance	[Pa·s/m]

Greek letters

α	Absolute flow angle	
α_{gv}	Guide vane blade angle	
α_r	Rotor blade angle	
$\Delta\alpha$	Rotor tip angle offset	
β	Relative flow angle	
η	Efficiency	[-]
μ	Dynamic viscosity	[Pa·s]
ρ	Density	[kg/m ³]
ϕ	Flow coefficient	[-]
ϕ_{pU}	Phase angle between pressure and flow rate	

Subscripts

1	Amplitude of oscillation
2	Time-averaged value
$a2e$	Acoustic to electric
gen	Generator
$h2e$	Heat to electric
m	Mean
t	Turbine
x	In axial direction

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Curriculum vitae

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Journal publications

- M. A. G. Timmer, K. de Blok, and T. H. van der Meer, "Implementing a bidirectional impulse turbine into a thermoacoustic refrigerator," J. Acoust. Soc. Am., current status: to submit.
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