A THERMOACOUSTIC ENGINE REFRIGERATOR SYSTEM FOR SPACE EXPLORATION MISSION

by

SUDEEP SASTRY

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Dissertation advisor: Dr. Jaikrishnan R. Kadambi

Department of Mechanical and Aerospace Engineering
CASE WESTERN RESERVE UNIVERSITY

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सरस्वति नमस्तुभ्यं वरदे कामरूपिणि ।
विद्यारम्भं करिष्यामि सिद्धिर्भवतु मे सदा ॥

कर्मण्येवाधिकारस्ते मा फलेषु कदाचन ।
मा कर्मफलहेतुर्भूर्मा ते सङ्गोस्त्वऽकर्मणि ॥

- Bhagavad Gītā

To,
My parents,
and, my brother

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A Thermoacoustic Engine Refrigerator System for Space Exploration

Mission

Abstract

By

SUDEEP SASTRY

Unique cooling systems have to be designed to cool the electronic components of space exploration rover, especially in places like Venus, which has harsh surface conditions. The atmospheric pressure and temperature on the surface of Venus are 92 bars and 450 °C respectively, which make operation of electronic devices and sensors very difficult. An exploration rover sent to operate at an altitude of 40 km above Venus' surface will also need active refrigeration of its electronic components as the temperature can be around 145 °C. Conventional cooling methods are currently deemed unfeasible due to the short life span of moving parts of the refrigerator systems at high temperatures. Furthermore, alternate energy sources such as solar power are not an option on Venus, since the cloud layer consisting of concentrated sulfuric acid droplets is thick and the cloud layer reduces the solar intensity at the surface to about 2% of the intensity above the atmosphere. Therefore, developing alternate method of power and cooling systems are essential for Venus surface operation of any robotic rover. The advantages of using thermoacoustic systems are that there are no

moving parts, and they have efficiencies comparable to conventional systems. This work discusses the development and optimization of a standing wave thermoacoustic engine refrigerator system to be used as a cooling device for the electronic components. The effects of various parameters such as gas mixture ratio, pressure, stack material, etc. is discussed. The system designed provides 150 W of cooling power while operating between 170 °C and 50 °C. The surface cooling temperature drop of 400°C is too large to be achieved by a single unit. Hence, multiple units are staged in series to obtain the required cooling temperature on the surface.

Nomenclature

Symbol	Parameter	Units
a	Speed of sound	m/s
a_n	n^{th} term of a series	
A	Cross sectional area	m^2
C	Specific heat capacity	J/kg.K
COP	Refrigerator coefficient of performance	
D	Diameter	m
dE	Change in internal energy of the system	J
dQ	Heat added into the system	J
dW	Work done by the system	J
dS	Change in entropy of the system	J/K
f	Frequency	Hz
f	Thermoviscous function	
f_f	Friction factor	
$\dot{H}_2$	Time averaged energy flux	W
h	Enthalpy per unit mass	J/kg
i	$\sqrt{-1}$	
$\text{Im}[]$	Imaginary part of a complex variable	

k	Wave number	$1/m$
K	Thermal Conductivity	$W/m.K$
K_L	Minor loss coefficient	
l	Half thickness of the stack plate	m
L	Length	L
$\dot{m}$	Mass flow rate	Kg/s
M	Acoustic Mach number	
p	Pressure	Pa
P_A	Pressure amplitude	Pa
p_m	Mean pressure	Pa
P_r	Prandtl Number	
q	Heat generation	W/m^3
$\dot{q}$	Rate of heat transfer	W
Q_H	Heat exchanged by a system with a high temperature reservoir	J
Q_L	Heat exchanged by a system with a low temperature reservoir	J
r	Common ratio in a geometric series	
r_h	Hydraulic ratio	m
$Re[]$	Real part of a complex number	
s	Entropy	$J/kg.K$
s_m	Mean entropy	$J/kg.K$
S_n	Sum of nth term	

S_{gen}	Entropy generated	J/K
t	Time	s
T	Temperature	K
T_{H}	Hot side temperature	K
T_{L}	Cold side temperature	K
u	Component of velocity in x direction	m/s
U	Velocity	m/s
v	Component of velocity in y direction	m/s
$\vec{v}$	Velocity	m/s
V	Volume	m^3
x	Coordinate in the direction of motion of acoustic oscillation	m
y	Coordinate in the direction perpendicular to the motion of acoustic oscillation in gas medium	m
y'	Coordinate in the direction perpendicular to the motion of acoustic oscillation in gas medium	m
z	Coordinate in the direction perpendicular to the motion of acoustic oscillation and perpendicular to the plane of paper	m
y_0	Half distance between two parallel plates	m

Greek Letters

β	Thermal expansion coefficient	1/K
δ_k	Thermal boundary layer thickness	m

δ_v	Viscous boundary layer thickness	m
Δx	Stack length	m
γ	Ratio of the isobaric to isochoric specific heats	
ϵ	Internal energy of the gas	J
ϵ_s	Stack heat capacity ratio	J/kg.K
ξ_1	Gas displacement amplitude (In the direction of motion of the gas)	m
λ	Wavelength	m
η	Engine efficiency	
ϱ	Density	kg/m ³
ϱ_m	Mean density of fluid	kg/m ³
μ	Dynamic viscosity	kg.m/s ²
ω	Angular frequency	1/s
ν	Kinematic viscosity	m ² /s
ζ	Bulk viscosity	kg.m/s
Π	Perimeter of the stack material	m

Subscripts

1	First order
2	Second order
crit	Critical
E	Engine
Exit	Exit conditions

H	High
Inlet	Inlet conditions
k	Thermal
L	Low
m	Mean value
R	Refrigerator
s	Solid parameters
v	Viscous

Other Symbols

Overdot	Time rate
Overbar	Time average
$\sim$	Complex conjugate

Chapter 1

Introduction

Thermoacoustics is the branch of science that deals with the study of conversion of thermal energy into acoustic energy and vice versa. An acoustic wave is, essentially, a pressure wave that is oscillating in space. Associated with the acoustic wave are temperature fluctuations, which are an inherent part of the acoustic wave. At larger scales, these temperature fluctuations are generally negligible, but when the gas travels through small channels and/or at high pressures, the thermoacoustic effect becomes quite pronounced. In fact, this effect if controlled efficiently can be used to build a range of prime movers, refrigerators and heat pumps.

1.1 Motivation

The Venus surface mission is one of the priorities for future missions as per the National Academies of Science Space Studies Board's study, *New Frontiers in the Solar System: An Integrated Exploration Strategy* [1]. The environment of Venus can be termed as "among the most enigmatic in the solar system" and understanding its atmosphere, climate, geology, and history could shed considerable light on our understanding of our own planet.

Even though Venus is the closest planet to Earth and the second brightest object in the night sky (after the Moon), there was, and still is, a lack of information about Venus due

to its thick opaque cloud cover. The cloud cover concealed the planet's surface from view so effectively that it was only when unmanned space probes were sent to explore the planet at close range that some of its secrets were revealed. From 1961 to 1989, the US and USSR launched more than 30 spacecraft towards Venus. Some reached their destination, while others did not [2].

The first successful flyby was made in 1962 by Mariner 2, an American probe which confirmed high surface temperatures. This was followed by the Venera and the Vega missions which helped in establishing data about the thickness and chemical composition of the cloud layers as well as surface conditions. The USSR's Venera missions had multiple landers which relayed important surface information back to Earth, albeit, surviving anywhere from a few minutes to a few hours on the surface of Venus. Venera 7 was the first probe to land, surviving for few minutes before succumbing to the intense pressure and high heat. Venera 9 sent back the first images from the surface, while Venera 13 obtained first color panoramic views of the planet.

To pierce Venus' opaque clouds orbiting spacecraft such as National Aeronautics and Space Administration's (NASA) Magellan probe (launched 1989) have carried out radar surveys to map its surface (Figure 1.1), circling the planet 8 times per day. Currently Venus Express – launched by the European Space Agency in 2005 - is orbiting the planet [3]. Mainly, the aims of the Venus Express mission are to investigate how the global atmospheric circulation, the cloud chemistry, surface – atmosphere physical and chemical interactions together to produce a climate starkly different from Earth's [3].

Venus has undergone runaway greenhouse warming, whereby trapped solar radiation has heated the planet's surface to an average temperature of 450 °C. Corrosive atmospheric conditions exist due to the presence of sulfur compounds, including clouds which contain

sulfuric acid which precipitate as an acid rain called virga, which evaporates before reaching the surface. The dense cloud layer exists from an altitude of about 45 km to 65 km above the surface of Venus.

Figure 1.1: Surface of Venus as mapped by Magellan.
(Source: www.nasa.gov)

Despite the atmospheric pressure and temperature at the surface of Venus being 92 bars and 450°C , respectively, Venus bears the closest resemblance to the Earth, among all other celestial objects in our solar system. The mass of Venus is about 0.8 Earth masses and the surface gravity of Venus is 0.9 times the gravity of Earth. Venus has dense atmosphere

composed primarily of Carbon dioxide (96.5%) and, Nitrogen (~3.5%) [4]. Solar intensity incident on Venus is about 1.9 times the solar intensity incident on Earth, though most of it is reflected back due to the optical nature of clouds covering Venus [4]. Given their similarities and the relative distance from the Sun, Venus and Earth possibly had similar surface conditions early in their formation, but Venus subsequently evolved differently than Earth, developing an environment unsuitable for life. A basic goal is to realize how and why Venus, even though very similar to Earth in a lot of aspects, has progressed to such a different state.

Venus has active geochemical cycles and dynamic environments in the clouds and near surface that are not yet clearly understood. Similar to Earth, Venus has also been geologically active within the past billion years, yet its surface is very different from Earth's and exhibits no similar plate tectonics [1]. Furthermore, the composition of the lower atmosphere of Venus is unknown. In order to fully understand and develop an origination theory of Venus's atmosphere compositional measurements of the atmosphere, specifically the noble gases are required. This information is vital for comparisons of the factors that affect climate on Earth and on Venus, such as photochemistry, cloud chemistry, and surface-atmosphere interaction.

Most electronic devices and sensors cannot operate at 450 °C. However, the upper atmospheric regions of Venus have temperatures and pressures very similar to Earth [5-7] (Figure 1.2 and Figure 1.3). The upper reaches of atmosphere at about 55km have a temperature of about 27 °C at a pressure of 50 kPa, 50 km have a temperature of about 75 °C at a pressure of 100 kPa and at 40 km have temperatures of about 145 °C at pressures of about 350 kPa [5]. Previous missions to Venus have been short lived – a few hours, due to the extreme conditions. Majority of the information obtained about Venus has been from

spacecraft flybys such as the Mariner and Galileo or orbiting satellites such as Venera 15, Venera 16 and Magellan [8]. The primary drawback of using these satellites and flybys is the dense cloud cover of Venus. The cloud cover prevents obtaining detailed information about the planet. However if the Venus exploratory Venus mission vehicle is inserted below the cloud cover, i.e., at an altitude of about 40km, it can be obtain a great wealth of information. Various studies [6, 9-12] have been conducted on different designs of long endurance probes to explore the Venus atmosphere. The operating conditions the exploration mission vehicle will be subjected to at this altitude will still be harsh, but not as severe as the surface conditions and one can plan the vehicle to be under operation for months rather than hours.

Figure 1.2: Variation of temperature with altitude on Venus

Figure 1.3: Variation of pressure with altitude on Venus

Any robotic mission to Venus will be tasked with making compositional and isotopic measurements of the atmosphere. A core sample can be obtained at the surface and ascended to altitude to perform further geochemical and mineralogical analysis. Scientific data obtained by this mission will assist in grasping the history and stability of the greenhouse effect on Venus and the recent geologic history, including resurfacing [1]. A few key measurements which will provide important data are [1, 10]:

- Visual imagery and mapping (including magnetic field) of the surface below the clouds to provide advanced geological interpretation of the surface.
- Global atmospheric dynamics explored in detail with long-lived instruments including in situ pressure and temperature measurements.
- Noble gas and other trace compounds measurements made with a simple Venus atmosphere sample-return mission.

- Detect, if any, biogenic gases as presence of life.
- Communication with surface-landers.

The mission's purpose would be to shed light on an enduring mystery about how a planet so similar to our own in size, mass, and composition has evolved so differently over the last 4.6 billion years. Closely studying the atmospheric composition and the geologic structure of Venus can help to predict under what conditions the runaway greenhouse effects occur. This data can then be compared to terrestrial climate computer models to predict how changes in Earth's atmospheric composition will impact the overall terrestrial climate.

The Venus exploratory mission vehicle is required to operate for long durations in these conditions. The Venus exploratory mission vehicle that is operational at an altitude of about 40 km will be able to image the Venusian surface and make other lower atmospheric measurements without being impeded by the dense cloud cover of Venus. Since most electronic components do not operate at 145 °C, power and cooling systems are required for Venus surface operation.

Solar power is not an option on Venus, since the cloud layer consisting of concentrated sulfuric acid droplets is thick and the surface does not get a direct view of the sun, resulting in a solar intensity at the surface of about 2% of the intensity without cloud cover. The light intensity is equivalent to the light intensity during a rainy day on Earth [13]. NASA has selected nuclear energy to power the rover, but the efficiency of current technology, which uses the electric current produced by a thermocouple, is only four percent efficient. It is so low that the weight of nuclear fuel required to produce sufficient power becomes significantly large; hence NASA plans to use a radioisotope power source for these

missions. Landis and Mellot [13], involved in the design of a radioisotope power system to provide electrical power for a probe operating on the surface of Venus, conclude that for a mission duration of substantial length the use of thermal mass, which acts as a heat sink, to maintain an operable temperature range is impractical, and active refrigeration may be required to keep components at a temperature below ambient Venus temperatures. Their radioisotope Stirling power converter design produces a thermodynamic power output capacity of 478.1 watts, with a cooling power of 100 watts. The overall efficiency is calculated to be 23.36 %.

The duration of the trip to Venus creates a serious fatigue issue for such cooling devices. The moving parts associated with most engines can result in potential cyclical fatigue failure which could leave the rover without power. Now, the question is, how can one create an efficient engine/ refrigerator system without cranks, bearings, or compressors, which tend to wear down? One potential device is a thermoacoustic refrigerator that executes a thermodynamic cycle using an acoustic wave as an input. The principle of the thermoacoustic engine and refrigerator have been illustrated by Garret and Backhaus [14], Yazaki et al [15] and Swift [16]. The cycle described by Garret and Backhaus and Swift uses a speaker (as power source) to create a pressure gradient, which in turn produces a temperature gradient. This pressure gradient can be used to drive a motor, in the case of a Stirling engine, or to remove energy from a cold source and add it to a hot source, as in the case of refrigeration. Such a system drastically decreases the system weight and number of moving parts and hence increases the reliability of the refrigerator system. Based upon the thermoacoustic principles, one can use the radioisotope energy source to create the pressure pulses for driving the thermoacoustic power generation and cooling (refrigeration/air conditioning) system.

Thermoacoustic refrigerators and engines can be run using acoustic power generated by a heat source. Therefore, the total efficiency of the refrigeration is equal to the product of the heat to sound conversion efficiency and the efficiency of the refrigerator itself (Figure 1.4(a)). The earlier system (Figure 1.4(b)) used a heat source to run an engine which powers a generator which produces electricity in order to run a refrigerator. Therefore, the total refrigeration efficiency is the product of engine efficiency, electric conversion efficiency, and refrigerator efficiency. Since the thermoacoustic refrigerator reduces the number of elements that must act in series, it raises the total efficiency of the refrigeration. The combination of reducing elements acting in series, increasing engine and refrigerator efficiencies, and reducing the number of moving elements makes the thermo acoustic refrigerator a good candidate for providing the cooling system for NASA's Venus exploration mission.

Figure 1.4: Efficiencies involved in a (1.4(a)) thermoacoustic engine refrigerator system and (1.4(b)) conventional refrigeration device using a heat source as energy input

A detailed study is undertaken to develop an appropriate design to use thermoacoustic device for refrigeration on a Venus exploratory mission. Design Environment for Low-Amplitude ThermoAcoustic Energy Conversion (DeltaEC) [17], a program developed at the Los Alamos National Laboratory, is used in obtaining the system parameters. The current work involves investigating the effect of various parameters such as geometry of the system, material, and the fluid dynamics on the development of an efficient and compact system. The proposed objective, which follows, is based upon the results and observations from the current work.

1.2 Objectives

This work is focused on the design and development of a thermoacoustic cooling system for the Venus and other space exploration missions where it is necessary to cool down electronic component from high temperatures (145°C) to reasonable temperatures (50°C) so that the electronic components can function efficiently. The objectives include, addressing key areas of design of a thermoacoustic cooling system. This comprises of investigating and optimizing the effect that various parameters such as geometry, gas mixture ratios, pressure, frequency, etc. have on the functioning of the thermoacoustic system. The work involves development of an efficient and coupled thermoacoustic engine (pressure pulse engine)—thermoacoustic refrigerator system to suit the Venus exploration requirements and a design of a prototype device. A further objective of the study is to design a system which will be able to cool electronic components on the surface of Venus down to operable temperature levels, i.e., from 450°C to 50°C .

1.3 Outline

The work presented subsequently, in this dissertation, is organized such that Chapter 2 deals with background; a brief review of development of the research in the area of thermoacoustics through the ages. Chapter 3 details the basic theory of thermodynamic and thermoacoustic principles upon which the work in the manuscript is based. Chapter 4 deals with the design parameters, modeling and a parametric study of the thermoacoustic engine refrigerator system. Results are discussed in Chapter 5. The design details of a prototype device are provided in Chapter 6, and, finally, Chapter 7 deals with conclusions of this study and further recommendations for future work.

Chapter 2

Literature Review

Though the thermoacoustic phenomenon was first observed and studied more than two hundred years ago, greater understanding of the phenomena has been developed over the past four decades. The underlying principle and the governing equations were developed resulting in accelerating the research in this field. In this chapter, a brief review of earlier work and development of the research in the field is presented.

2.1 Early History

Putnam and Dennis [18], in their review of organ-pipe-combustion-oscillations-related phenomena described the work of Higgins [19] in 1777, who made the first recorded observation of a thermoacoustic phenomena when he excited organ pipe oscillations by appropriately positioning a Hydrogen flame inside a tube. These were also known as “singing flames”.

Figure 2.1: Glass Blowing: A hot glass bulb at the end of a cold tube.
Source: www.science.nasa.gov

For centuries, glass blowers noticed that during glass blowing (Figure 2.1) when a hot glass bulb is joined to a cold tube, sound is emitted from the tube. Based on this, in 1850, Soundhauss [20] became the first person to do quantitative research on thermoacoustics when he performed experiments in a hollow glass tube with one end open and a closed bulb at one end (Figure 2.2). Soundhauss noted that heating the closed end (bulb) produced acoustic oscillations in the tube in the audible range and the frequency of sound was based on the length of the tube and the volume of the bulb. He conducted experiments for various tube inner diameters, length and bulb volume. He also noted that higher heat input caused more powerful sound output.

Figure 2.2: Soundhauss Tube. Heat supplied at the bulb end of the unit generates sound wave at the open end.

Rijke [21], in 1859, discovered that placing a heated wire mesh in the bottom half of an open tube produced sound waves. The highest intensity was obtained when the mesh was placed at one-fourth the length of the tube as measured from the bottom. Rijke also observed that convective transport of air through the tube was necessary to produce the thermoacoustic effect. The Rijke tube (Figure 2.3) is an extension of Higgins work.

Figure 2.3: Rijke Tube. A heated mesh in an open tube generates sound waves.

The Soundhauss tube and the Rijke tube are seen as the precursors of the present day standing wave and travelling wave thermoacoustic machines, respectively. Feldman [22, 23] reviewed the work done on Soundhauss and Rijke tubes and concluded that though the geometry of Soundhauss oscillator could be optimized from the available experimental data and stated that further work was required to optimize the oscillators.

In his work on sound, Lord Rayleigh [24], discussed the experiments by Soundhauss and Rijke and qualitatively explained the thermoacoustic principle. It can be summarized as: the generation of sound waves is encouraged if the phase between fluid motion and heat transfer to the fluid is appropriate. When density of the oscillating fluid is highest, it receives heat, and when density of the oscillating fluid is least, heat is rejected from the oscillating fluid.

2.2 Development of the thermoacoustic theory

Taconis [25] arrived at a similar conclusion as Rayleigh when he observed spontaneous acoustic oscillations, known as “Taconis oscillations”, reaching high amplitudes when a gas filled tube at room temperatures approaches cryogenic temperatures. Until Kramers [26] tried to provide a mathematical explanation to these “Taconis oscillations”, the thermoacoustic phenomena had only been qualitatively described. However, Kramers work was mismatch between theoretical and experimental results due to incorrect assumptions.

Major advances in the understanding of the thermoacoustic phenomena quantitatively came with the seminal studies made by Rott [27-32]. Rott, in a series of papers, developed the theory behind the thermoacoustic phenomena; deriving the equations for pressure, motion and time averaged energy transport, considering the acoustic oscillations in long tubes and varying diameters (large and small) in relation to the thermal penetration

depth of the working fluid. Rott's theory has been validated by various studies including Yazaki et al [33-35] and Wheatley et al [36-38]. Rott's theory has also been successfully used to design and predict the performance of thermoacoustic devices [17].

A broad perspective of thermoacoustic engines and components thereof is given by Swift [39]. The work included theoretical illustration of the thermoacoustic concept on a single plate and extending it to the stack and other parts of the thermoacoustic engine: heat exchangers, resonators, etc. Pulse tube refrigerators and Stirling engines is also discussed. Wheatley et al [38] carried out extensive work in understanding the thermodynamic aspects of a thermoacoustic device by building and carrying out experiments on simple thermoacoustic devices.

Improved understanding has led to studies including, thermoacoustics effects on a single plate where Wetzel and Herman [40, 41] experimentally observed the thermoacoustic effect through temperature measurements. They also implemented an evaluation procedure that accounts for the change in the refractive index due to acoustic pressure variations and measured the heat fluxes. Studies have also been conducted to experimentally validate numerical studies; Bailliet et al [42] conducted acoustic flow measurements using Laser Doppler Anemometry (LDA) in the resonator of a thermoacoustic refrigerator and compared the result to analytical calculations. Yazaki and Tominaga [43] have conducted experiments to measure the pressure and velocity in an acoustic resonator. Siddiqui and Nabavi [44] used Particle Image Velocimetry (PIV) to experimentally determine planar velocity fields of an acoustic standing wave in a rectangular channel. At certain locations, their experimental and theoretical results differed by 2.4%. Babaei and Siddiqui [45] developed a system for optimum design of thermoacoustic devices considering different

parameters involved. Qiu et al [46], discuss the optimum packing factor involved in the construction of a refrigerator stack material in the design of a thermoacoustic refrigerator.

2.3 Thermoacoustic Engines

As the phenomenon became better understood, people realized that useful work could be extracted from heat generated acoustic oscillations. This led to the invention of thermoacoustic engines which would convert the generated acoustic power into electricity. In 1951, the first patent [47] for a device to use thermoacoustic principle was issued. The device used heat input to generate acoustic pressure pulses – similar to the Soundhauss tube – which was then converted to electrical energy using an acoustical-electrical transducer. Another patent [48] issued in 1958 similarly had two acoustic engines, mounted in opposition for vibration balance, was used to convert heat to acoustic energy to electricity. These devices were inexpensive, reliable and had no moving parts. Novel though the concept was at the time to generate electricity using a device without any moving parts, the glaring disadvantage of these devices were their low efficiency (<10% for conversion of heat into acoustic power).

Ceperley [49] in 1979 realized that higher efficiencies could be obtained if the acoustic wave underwent phasing. The cycle the wave followed in these circumstances were similar to the Stirling thermodynamic cycle. Using this philosophy, the first Stirling cycle based thermoacoustic engine was developed [50]. The device was in effect a working Stirling engine without any moving parts i.e. the inertia of the acoustic oscillations transferred the energy instead of piston, crankshaft and other moving parts associated with a regular Stirling device. Unfortunately, the device did not amplify the acoustic power. Yazaki et al [51], in 1998, constructed the first working model of the thermoacoustic Stirling engine. The

efficiencies were lower than anticipated due to unaccounted thermal and viscous losses. Later on, Backhaus and Swift [52] built the Thermoacoustic Stirling hybrid engine (TASHE) using a more rigorous design criteria. TASHE had an efficiency of 0.30, corresponding to 41% of Carnot efficiency.

Recently, Telesz [53] built a thermoacoustic power converter consisting of a thermoacoustic-Stirling engine interfacing with a pair of linear alternators to produce 100W of electricity from a heat input at 1200⁰ F. Helium at 450 psig was used as the working fluid. The heat to acoustic power conversion has an efficiency of 26.3% while the efficiency of conversion of heat to electricity is 16.8%.

2.4 Thermoacoustic Refrigerators

Even though the concept of producing acoustic oscillations by providing a temperature gradient is centuries old, it was not until Gifford and Longworth [54], that people started researching the reverse effect of a thermoacoustic engine i.e. generating a temperature gradient using acoustic wave as input. Gifford and Longworth [54] built a “pulse tube” refrigerator which pumped heat along the inner surface of a closed chamber by maintaining a low frequency pressure pulse generated by oscillating gas. Merkli and Thomann [55], observed the thermoacoustic cooling effects in an acoustically resonant tube at a velocity antinode. Once the refrigeration aspect of thermoacoustics was discovered, it sparked intense research at the Los Alamos National Laboratory (LANL) during the 1980s. Wheatley and Cox [56] discuss the convertibility of a heat pump into a thermoacoustic engine upon the application of a temperature gradient in the absence of an imposed acoustic wave.

Hofler [57] designed and constructed a functioning thermoacoustic refrigerator using a loudspeaker as an acoustic source. A 12% COP relative to the Carnot COP was achieved. The lowest measured ratio of cold temperature to that of the ambient temperature was 0.66. Further work on loudspeaker driven thermoacoustic refrigerator was carried out by Tijani [58]. A lowest temperature of -67°C was reported to be achieved. The effect of Prandtl number on the efficiency was studied using different helium-noble gas mixtures. A maximum COP relative to Carnot of 17% was reported when using a Helium-Xenon mixture containing 30% Xenon.

2.5 Standing Wave and Traveling wave systems

Thermoacoustic systems can be broadly classified into two divisions: the standing wave systems and the traveling wave systems. The earliest examples of a standing wave thermoacoustic device and a traveling wave thermoacoustic device are the Soundhauss and Rijke tubes (Figures 2.1 and 2.2) respectively. The primary difference between standing wave thermoacoustic systems and traveling wave thermoacoustic system is that standing wave thermoacoustic systems follow the Brayton cycle (Figure 2.4) whereas traveling wave thermoacoustic systems follow the Stirling cycle (Figure 2.5). The Brayton cycle consists of two isobaric processes – heat addition and heat rejection, and two adiabatic processes – expansion and compression [59]. The thermoacoustic Brayton cycle is discussed in detail in the subsequent chapter (Section 3.2). The ideal Stirling cycle consists of two isochoric processes – heat addition and heat rejection, and two isothermal processes – an expansion process and a compression process. However, in a thermoacoustic Stirling cycle, neither the heat addition and rejection processes are isochoric, nor the expansion and compression processes isothermal.

Figure 2.4: Brayton Cycle

Figure 2.5: Stirling Cycle

Further, a standing wave thermoacoustic engine can be described as a device in which the velocity of gas oscillations and the acoustic pressure are 90° out of phase with each other. The stack spacing is of the order of the thermal boundary layer and the two ends of the stack are maintained at a sufficiently large temperature gradient. As the gas oscillates, it expands and contracts exchanging heat with stack and producing acoustic power. Imperfect thermal contact between the oscillating gas and the solid stack is required to enable the thermal expansion and contraction of the gas to be in phase with the oscillating pressure and out of phase with the velocity [60].

In the traveling wave thermoacoustic engine, the velocity of the gas and the oscillating pressure are in phase, which implies that good thermal contact is required between the gas and the regenerator. The gas exchanges heat with regenerator while undergoing expansion and contraction. A good thermal contact is generated by keeping the channel sizes in the regenerator smaller than the thermal penetration depth.

2.6 Thermoacoustic Engine-Refrigerator Systems

The concept of using thermally generated acoustic oscillations to drive a thermoacoustic refrigerator evolved from the individual thermoacoustic engine and thermoacoustic refrigerator systems. Thermoacoustic engines produce acoustic oscillations and thermoacoustic refrigerators need acoustic oscillations to function. Coupling both the systems effectively results in a thermoacoustic engine-refrigerator system which produces a cooling effect using a heat source as input.

Hofler [61, 62] built a thermoacoustically driven thermoacoustic refrigerator (TADTAR). The unit had a total COP of 15% and delivered a cooling power of 91 W across

a temperature span of 25°C with the hot side heat addition temperature being 450°C . Helium-Argon mixture was used as the working fluid.

A thermoacoustic cooler using solar power to generate acoustic oscillations was built by Chen [63]. The device used a solar collector to focus the radiation from sun to the engine side of the device to generate acoustic waves. The acoustic waves were then utilized to produce the refrigeration effect. The goal to freeze water was not met due to leakage and losses. Nevertheless, the device functioned well enough to demonstrate the effect.

A predominant area of application of thermoacoustic engine refrigerator systems has been cryogenics. Jin et al [64] used sound from a thermoacoustic prime mover to drive a pulse tube refrigerator to achieve cooling down to 120 K. Effects of using different gas mixtures were also discussed. Dai et al [65] constructed a travelling wave thermoacoustic refrigerator driven by a travelling wave thermoacoustic engine. The device reached a lowest temperature of -66°C . 250 W of cooling at -22°C was obtained when using Helium gas at 3 MPa mean pressure.

The available literature shows that most of the research carried out in the area of thermoacoustic refrigeration, has been to produce the cooling effect from a maximum of Earth ambient conditions and to temperatures in the range of cryogenic temperatures. Research and development in the area of standing wave thermoacoustic engine refrigerator system for use in cooling a high temperature system is nearly non-existent. The work, presented in the succeeding chapters is an attempt to address the issue, by developing a standing wave thermoacoustic engine refrigerator system for a space exploration system.

Chapter 3

Thermoacoustics: Concepts and Theory

The thermodynamics and concepts associated with thermoacoustic are presented in this chapter. The discussion includes thermodynamic efficiencies of engines and refrigerators, principle of the thermoacoustic theory, governing equations and important parameters in thermoacoustics.

3.1 Thermodynamics

The two pillars of thermoacoustics are thermodynamics and acoustics. While acoustics deals with the dynamic properties of gas oscillations such as type of gas, pressure, velocity, phase, etc., thermodynamics deals with the energy conversion, efficiency and heat transfer. The energy conversion from one form to another (heat to sound and vice versa) follows the first law of thermodynamics which states that energy can neither be created nor destroyed. It states that overall change in the total energy of a system is equal to the difference of the algebraic sum of the heat flow and work done by the system [60]. Mathematically,

$$dE = dQ - dW \quad [3.1]$$

The second law of thermodynamics talks about inequalities involved in a system. The second law states that for a system, change in entropy for a process is given by

$$dS = \frac{dQ}{T} + S_{\text{gen}} \quad [3.2]$$

Additionally, the second law also states that

$$S_{\text{gen}} \geq 0 \quad [3.3]$$

A thermodynamic engine or a prime mover is a device that produces work by receiving heat (Q_H) from a high temperature source (T_H) and rejecting heat (Q_L) to a low temperature sink (T_L). And similarly, but on the reverse principle, a heat pump or a refrigerator is a device that absorbs heat (Q_L) from a low temperature (T_L) source and rejects heat (Q_H) to a high temperature (T_H) sink using up work. The first law for an engine and a refrigerator becomes (from Equation [3.1]):

$$Q_H = Q_L + W \quad [3.4]$$

For an engine the second law becomes (from Equations [3.2] and [3.3]):

$$\frac{Q_L}{T_L} - \frac{Q_H}{T_H} \geq 0 \quad [3.5]$$

Similarly for a refrigerator:

$$\frac{Q_H}{T_H} - \frac{Q_L}{T_L} \geq 0 \quad [3.6]$$

Let us consider a thermodynamic engine operating between a heat source at temperature T_{HE} and a heat sink at a temperature T_{LE} producing W_E watts of work and a thermodynamic refrigerator operating between a heat source at temperature T_{HR} and a heat sink at a temperature T_{LR} , using W_R watts of work (Figure 3.1).

The efficiency of a heat engine is defined as the ratio of useful work delivered to that of heat supplied to the system.

$$\eta = \frac{W_E}{Q_{HE}} \quad [3.7]$$

Combining Equations [3.4] and [3.5], and eliminating Q_L , we get

$$\eta = \frac{W_E}{Q_{HE}} \leq \frac{T_{HE} - T_{LE}}{T_{HE}} \quad [3.8]$$

Figure 3.1: Thermodynamic Engine and Refrigerator

The ratio $\frac{T_{HE} - T_{LE}}{T_{HE}}$ is the Carnot efficiency. Carnot efficiency is the maximum efficiency

limit for all heat engines working between two temperatures T_{HE} and T_{LE} .

For a refrigerator its coefficient of performance (COP) is given by combining Equations [3.4] and [3.6], and, eliminating Q_{HR} .

$$\text{COP}_R = \frac{Q_{LR}}{W_R} \leq \frac{T_{LR}}{T_{HR} - T_{LR}} \quad [3.9]$$

where $\frac{T_{LR}}{T_{HR} - T_{LR}}$ is the Carnot coefficient of performance. It is the maximum performance limit for a given refrigerator operating between two temperatures. By combining these two devices, we can create a system, which uses heat to remove heat, or in other words, the work generated by the engine is used by the refrigerator to do work, i.e., extract heat from a body at a lower temperature and pump it to a body at a higher temperature.

For such a combined system which incorporates a thermoacoustic engine and a thermoacoustic refrigerator, the overall COP is defined as the ratio of heat removed from the cold temperature reservoir by the refrigerator to that of heat added from the high temperature reservoir into the thermoacoustic engine. Mathematically,

$$\text{COP}_{\text{overall}} = \frac{Q_{LR}}{Q_{HE}} \quad [3.10]$$

The overall Carnot COP is given by

$$\text{Carnot COP}_{\text{overall}} = \left(1 - \frac{T_{LE}}{T_{HE}}\right) \left(\frac{T_{LR}}{T_{HR} - T_{LR}}\right) \quad [3.11]$$

where T_{LE} is the heat rejection temperature of the engine and T_{LR} is the heat rejection temperature of the refrigerator. It follows that,

$$\text{COP}_{\text{overall}} < \text{Carnot COP}_{\text{overall}} \quad [3.12]$$

3.2 The thermoacoustic effect

As mentioned earlier, acoustic waves are pressure oscillations in space, which contain temperature fluctuations associated with the compression and expansion of the pressure waves. These temperature fluctuations in acoustic waves are negligible in everyday phenomenon. For example, speaking causes a pressure variation of about 10^{-6} psi and a

temperature fluctuation of 10^{-4} °C. And at 120 dB, the auditory pain threshold, the pressure fluctuation in an acoustic wave is 10^{-2} psi and the temperature fluctuation is about 10^{-2} °C [14]. It can be inferred that these variations are too small to be practically used, as is. One way to extract useful work from the acoustic waves is to put a solid with high specific heat capacity and a large surface area in contact with the oscillating gas. Due to their higher specific heat capacity compared to gas, the solids can store and exchange heat with the gas without a considerable change in temperature. Therefore, during expansion of a gas volume in an oscillation cycle, heat is absorbed by the gas and during the compression heat is absorbed by the solid, keeping the overall temperature of the gas stable.

The thermoacoustic principle is best illustrated by the following example (Figure 3.2). Consider a long tube filled with gas containing a solid material of high specific heat capacity and low thermal conductivity known as the stack. The stack geometry is such that it has pores (similar to a honeycomb structure) and this creates numerous channels for the acoustic wave to travel. To generate the thermoacoustic effect, a sufficiently large temperature gradient is applied across the ends of the stack (Figure 3.2(a)) by placing two heat exchangers – one high temperature and one low temperature – in contact with the ends of the stack material. Using the Lagrangian approach, we follow a parcel of gas, which we define as our control volume, as it oscillates in the system. The motion of the gas parcel is sinusoidal, but for sake of clarity, it is described here in a step by step motion. At the hot end, the gas parcel absorbs heat from the neighboring stack material at a constant pressure (Step 1). As a consequence, its temperature rises and the parcel moves to the right (towards the colder end).

Figure 3.2: Schematic diagram showing the heat transfer process by thermoacoustic oscillations in the stack.

During this step, the gas parcel expands adiabatically (Step 2) and pressure and temperature in the gas parcel are lowered. However, at the end of Step 2, the temperature of the gas is still higher than that of the adjacent stack material. This results in an isobaric heat transfer from the gas parcel to the stack material (Step 3). Finally, the gas parcel adiabatically

compresses back to its original state (Step 4) and thereafter the cycle continues. As the gas parcel undergoes the cycle of exchanging heat between different regions of the stack material it oscillates at the acoustic frequency generating work in the form an acoustic wave. This is the primary thermoacoustic effect.

The reverse outcome, i.e., producing the refrigeration effect, can easily be achieved, by using oscillating sound waves through a stack material in a tube (Figure 3.2(b)). As the sound waves resonate back and forth in the chamber, the gas is compressed as it is shifted to one side (Step 1). As it is compressed, the temperature of the gas parcel increases and it becomes higher than that of the neighboring stack material. During the next step it transfers heat to the solid (Step 2). Then, as it oscillates back in the other direction, the gas parcel expands and cools down sufficiently during the process so that its temperature is less than that of the adjoining stack material (Step 3). This leads the gas parcel to absorb heat from the stack material (Step 4). The effect is to pick up heat from a solid at a lower temperature and transfer it to a solid at a higher temperature. It must be noted that, despite the fact that an individual parcel of gas transfers only a small amount of heat, overall, all the gas parcels, combined, act as a “bucket brigade” [14] transferring heat from the cold end to the hot end. Thus, the thermoacoustic refrigeration effect is induced.

The temperature variations of the gas are due to two reasons. One, due to the adiabatic compression and expansion of the acoustic wave itself, and second, due to its interaction with the adjacent stack material. It is important to note that the distance, perpendicular to the direction of acoustic wave propagation, over which the heat transfer takes place within the gas in contact with the solid stack material is known as the thermal penetration depth (δ_k). The thermal penetration depth depends on the frequency of the acoustic wave and also on the properties of gas. For optimal heat transfer between the gas

and the solid, in the stack, the spacing in the stack should be about twice the thermal penetration depth, as there are two solid surfaces (top and bottom) for the gas to interact with. The gas parcels that are farther away than the thermal penetration depth in the stack material or, those present in the resonator undergo simple adiabatic acoustic expansion and compression without experiencing heat transfer. It is also important to note that the temperature gradient plays a great role in determining whether a system can function as a prime mover or as a heat pump, since both systems are interchangeable. A small to moderate temperature gradient is an essential condition for a heat pump; whereas a relatively higher temperature gradient is requisite for a prime mover or an engine.

3.3 Relevant Acoustic Concepts

A few of the important acoustic concepts that govern the thermoacoustic effect are discussed in this section. Since the device under study is a standing wave thermoacoustic device, the discussion will be appropriately limited to standing wave thermoacoustic devices. Let us consider a half-wavelength resonator of length L , which contains two heat exchangers exchanging heat with a stack material (Figure 3.3) generating an acoustic wave. Here, the nodes are points in space of a standing wave where the amplitude of a particular parameter is at a minimum; while antinodes are points where the amplitude of the parameter is at a maximum. An antinode occurs midway between two nodes.

For a 1 dimensional wave oscillating with an angular frequency $\omega=2\pi f$, where f is the frequency of oscillation, the equations for pressure (Equation [3.13]), velocity (Equation [3.14]), temperature (Equation [3.15]), density (Equation [3.16]) and entropy (Equation[3.17]) can be expressed, respectively, as:

$$p = p_m + \text{Re}[p_1(x)e^{i\omega t}] \quad [3.13]$$

Where x is the distance along the direction of motion of the acoustic wave. The acoustic pressure is dependent only in the x -direction. The subscript 1 denotes first order expansion of the complex terms.

Figure 3.3: A standing wave thermoacoustic engine of resonator length L , depicting the location of the velocity and pressure nodes and antinodes

$$u = \text{Re}[u_1(x, y, z)e^{i\omega t}] \quad [3.14]$$

Where x is the distance along the direction of motion of the acoustic wave, y is the direction perpendicular to the motion of acoustic wave, but in the plane of paper, while, z is the direction perpendicular to the plane of paper (Figure 3.3). The mean velocity, u_m , is considered to be zero assuming there is no mean flow.

$$T = T_m + \text{Re}[T_1(x, y, z)e^{i\omega t}] \quad [3.15]$$

Similar to the temperature, T , the equations for density, ρ and entropy s , are:

$$\rho = \rho_m + \text{Re}[\rho_1(x, y, z)e^{i\omega t}] \quad [3.16]$$

$$s = s_m + \text{Re}[s_1(x, y, z)e^{i\omega t}] \quad [3.17]$$

The term $e^{i\omega t}$ represents the oscillatory part associated with an acoustic wave. The subscript m denotes the mean values of the variables.

The x-component acoustic pressure and acoustic velocity in the resonator are given by [39]:

$$p_1(x) = P_A \cos kx \quad [3.18]$$

$$u_1(x) = \frac{P_A}{\rho_m a} \sin kx \quad [3.19]$$

Where $k = \omega / a$ is the wave number, P_A is pressure amplitude, ρ_m is mean density of the fluid, a is the speed of sound and x is the distance along the direction of motion of the acoustic wave.

Ideally, for standing wave devices, the acoustic pressure and velocity are out of phase by 90 degrees, as is evinced by Equations [3.18] and [3.19] (illustrated in Figure 3.3). This is due to the time lag between the heat transfer from the solid stack material to the gas and the gas motion. In other words, the gas temperature does not change instantly, but takes time, thereby creating a phase difference between the heat transfer and pressure and the acoustic velocity.

3.4 Principle of Thermoacoustics

In this section, basic equations that govern the thermoacoustic phenomena are discussed. A detailed description has been developed by Rott [27, 29-32] , Wheatley et al. [36] and Swift [39].

To assist in our discussion of the basic principles of thermoacoustics, we consider a simple parallel plate stack in a gas filled resonator as illustrated in Figure 3.4 [39, 58]. A sustained one dimensional acoustic wave is transmitted through the system. The following assumptions are made:

- Steady state conditions exist.
- The plates are rigid and stationary.
- The length of the plates is relatively small compared to the length of the resonator.
- The acoustic pressure is x -dependent only and is constant over the entire cross section of the plates.
- Higher order effects such as turbulence and acoustic streaming are ignored.
- Radiation is neglected.
- The temperature difference across the stack is relatively small compared to the absolute value of temperature. Viscosity is assumed independent of temperature.
- The average fluid velocity is zero.

The co-ordinate system is defined as the x -axis being along the direction of acoustic vibration and the y -axis perpendicular to the plane of the parallel plates (Figure 3.4). The distance between the plates is $2y_0$ while the thickness of the plates is $2l$. Two different set

of axes are defined for the system. The location $y = 0$ is defined at the center of the gas layer in between two plates. This sets $y = y_0$ at the boundary of gas and the plate and $y' = 0$ is defined at the center of the plates, which sets $y' = 1$ at the gas-solid boundary.

Figure 3.4: A simple short stack thermoacoustic engine with stack spacing $2y_0$ and plate thickness $2l$.

The 3 basic thermoacoustic equations are the wave equation (as derived by Rott [27-29]), the energy equation and the acoustic power equation. The first order expansion in the acoustic amplitude is used for all variables. (Example: $\varrho = \varrho_m + \varrho_1 e^{i\omega t}$) [39].

The continuity equation is

$$\frac{\partial \varrho}{\partial t} + \nabla \cdot (\varrho \vec{v}) = 0 \quad [3.20]$$

Neglecting higher order terms, we get,

$$i\omega\varrho_1 + \frac{\partial}{\partial x}(\varrho_m u_1) + \varrho_m \frac{\partial v_1}{\partial y} = 0 \quad [3.21]$$

Where u_1 is the x component of the velocity and v_1 is the y component of the velocity and ϱ is the density of the fluid.

The momentum equation is

$$\varrho \left[\frac{\partial \vec{v}}{\partial t} + (\vec{v} \cdot \nabla) \vec{v} \right] = -\nabla p + \mu \nabla^2 \vec{v} + \left(\zeta + \frac{\mu}{3} \right) \nabla (\nabla \cdot \vec{v}) \quad [3.22]$$

Where $\vec{v}$ is velocity, ϱ is density, p is pressure, μ is dynamic viscosity and ζ is bulk viscosity.

The x-component, first order expansion of the momentum equation gives us:

$$i\omega\varrho_m u_1 = -\frac{dp_1}{dx} + \mu \frac{\partial^2 u_1}{\partial y^2} + \left(\zeta + \frac{\mu}{3} \right) \frac{\partial}{\partial x} (\nabla \cdot \vec{v}_1) \quad [3.23]$$

Where u_1 is the x component of the velocity.

Now, $\partial / \partial x$ is of the order of $1/\lambda$ and $\partial / \partial y$ is of the order of $1/\delta_v$. This is because we know that the characteristics length associated in the x-direction is λ and the characteristic length associated with the y-direction is δ_v . But $\delta_v \ll \lambda$ which implies that $\partial / \partial x$ may be neglected when compared to $\partial / \partial y$ terms. Therefore, we can neglect the $\partial / \partial x$ term.

Thus the x component of the momentum equation is:

$$i\omega\varrho_m u_1 = -\frac{dp_1}{dx} + \mu \frac{\partial^2 u_1}{\partial y^2} \quad [3.24]$$

The boundary condition for Equation [3.24] is $u_1=0$ at $y=0$. The solution for u_1 is [39]:

$$u_1 = \frac{i}{\omega\varrho_m} \left[1 - \frac{\cosh \left[\frac{(1+i)y}{\delta_v} \right]}{\cosh \left[\frac{(1+i)y_0}{\delta_v} \right]} \right] \frac{dp_1}{dx} \quad [3.25]$$

where

$$\delta_v = \sqrt{\frac{2\mu}{\rho_m \omega}} \quad [3.26]$$

The energy equation is

$$\frac{\partial}{\partial t}(\rho \epsilon + \frac{1}{2} \rho |\vec{v}|^2) = -\nabla \cdot [-K \nabla T - \vec{v} \cdot \Sigma + (\rho h + \frac{1}{2} \rho |\vec{v}|^2) \vec{v}] \quad [3.27]$$

where ρ is density, $\vec{v}$ is velocity, p is pressure, μ is dynamic viscosity, K is the gas thermal conductivity, ϵ is the internal energy and h is the enthalpy per unit mass, and Σ is the viscous stress tensor, given by

$$\Sigma_{ij} = \mu \left(\frac{\partial v_i}{\partial x_j} + \frac{\partial v_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial v_k}{\partial x_k} \right) + \zeta \delta_{ij} \frac{\partial v_k}{\partial x_k} \quad [3.28]$$

Where ζ is the bulk viscosity.

From the conduction equation, the temperature of in the solid stack is given by:

$$\rho_s C_s \frac{\partial T_s}{\partial t} = K_s \left(\frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} + \frac{\partial^2 T_s}{\partial z^2} \right) + q \quad [3.29]$$

where K_s is the thermal conductivity of the solid, C_s is the specific heat capacity of the solid, ρ_s is the density of the solid material, and q is the heat generation term.

Neglecting the heat generation term, we can rewrite the above equation as:

$$\frac{\partial T_s}{\partial t} = \frac{K_s}{\rho_s C_s} \nabla^2 T_s \quad [3.30]$$

The first order approximation of Equation [3.30] gives us the following equation for the temperature of the solid material,

$$i\omega T_{s1} = K_s \frac{\partial^2 T_{s1}}{\partial y'^2} \quad [3.31]$$

The solution for Equation [3.31] is given by [39],

$$T_{s1} = T_{b1} \frac{\cosh\left[(1+i)y' / \delta_s\right]}{\cosh\left[(1+i)l / \delta_s\right]} \quad [3.32]$$

Here, δ_s is the solid's thermal penetration depth and T_{b1} is the temperature amplitude at the solid-gas interface at $y' = l$.

And the temperature of the gas is given by (derived from Equation [3.27])

$$\rho T \left(\frac{\partial s}{\partial t} + v \cdot \nabla s \right) = K \nabla \cdot (\nabla T) + (\text{higher order terms}) \quad [3.33]$$

Where s is the gas entropy, ρ is the density of the gas and K is the thermal conductivity of the gas.

Also,

$$ds = \left(\frac{C_p}{T} \right) dT - \left(\frac{\beta}{\rho} \right) dp \quad [3.34]$$

Where C_p is the specific heat capacity of the gas per unit mass and β is the thermal expansion coefficient.

Using Equation [3.34], Equation [3.33] becomes

$$\rho_m C_p \left(i\omega T_1 + u_1 \frac{dT_m}{dx} \right) - i\omega \beta T_m p_1 = K \frac{\partial^2 T_1}{\partial y^2} \quad [3.35]$$

At the solid-gas interface, the gas and solid temperatures are the same. Therefore, the boundary conditions translate into,

$$\begin{aligned} T_1(y_0) &= T_s(l) = T_b \\ K \left(\frac{\partial T_1}{\partial y} \right)_{y_0} &= -K_s \left(\frac{\partial T_s}{\partial y'} \right)_1 \end{aligned} \quad [3.36]$$

Using these boundary conditions (Equation [3.36]), the solution to Equation [3.31] is,

$$T_{s1} = \left[\frac{T_m \beta}{\rho_m C_p} P_1 + \frac{1}{\rho_m \omega^2} \frac{dT_m}{dx} \left(\frac{dp_1}{dx} \right) \left(\frac{1}{P_r - 1} \right) \left(1 - \frac{f_v}{P_r f_k} \right) \right] \frac{\cosh((1+i)y'/\delta_s)}{\cosh((1+i)l/\delta_s)} \quad [3.37]$$

Where

$$f_v = \frac{\tanh((1+i)y_0/\delta_v)}{(1+i)y_0/\delta_v} \quad [3.38]$$

$$f_k = \frac{\tanh((1+i)y_0/\delta_k)}{(1+i)y_0/\delta_k} \quad [3.39]$$

$$\epsilon_s = \frac{\sqrt{K \rho_m C_p} \tanh((1+i)y_0/\delta_k)}{\sqrt{K_s \rho_s C_s} (1+i)l/\delta_s} \quad [3.40]$$

$$\delta_k = \sqrt{\frac{2K}{\rho C_p \omega}} = \sqrt{\frac{2\kappa}{\omega}} \quad [3.41]$$

$$\delta_s = \sqrt{\frac{2K_s}{\rho_s C_s \omega}} \quad [3.42]$$

$$P_r = \left(\frac{\delta_v}{\delta_k} \right)^2 \quad [3.43]$$

where δ_v is the viscous penetration depth, ν is the kinematic viscosity of the gas, ρ is the density of the gas, μ is the dynamic viscosity, δ_k is the thermal penetration depth, δ_s is the solid's thermal penetration depth, K is the thermal conductivity, C_p is the specific heat capacity, κ is the thermal diffusivity of the gas, ω is the angular frequency of the acoustic wave and P_r is the Prandtl number for the gas mixture. f is a function known as Rott's function. It is dependent on the geometry. The Rott function for various geometries have been derived and reported in different studies [60].

Now, using the boundary conditions from Equation [3.36], substituting for u_1 from Equation [3.25] and solving Equation [3.35] for T_{s1} , we get

$$T_1 = \frac{\beta T_m}{\varrho_m C_p} p_1 - \frac{1}{\varrho_m \omega^2} \frac{dT_m}{dx} \frac{dp_1}{dx} \left(1 - \left[\left(\frac{P_r}{P_r - 1} \right) \frac{\cosh[(1+i)y / \delta_v]}{\cosh[(1+i)y_0 / \delta_v]} \right] \right) \quad [3.44]$$

$$- \left(\frac{\beta T_m}{\varrho_m C_p} p_1 + \frac{1}{(P_r - 1)\varrho_m \omega^2} \frac{dT_m}{dx} \frac{dp_1}{dx} \left(1 + \frac{\varepsilon_s f_v}{f_k} \right) \right) \frac{\cosh[(1+i)y / \delta_k]}{(1 + \varepsilon_s) \cosh[(1+i)y_0 / \delta_k]}$$

Rott's wave equation can be derived by combining first order continuity equation, x derivative of the momentum equation and using the expressions for both u_1 and T_1 .

Combining Equation [3.21] and the x derivative of the momentum equation [3.24],

$$-\omega^2 \varrho_1 - \frac{dp_1}{dx^2} + \frac{\partial}{\partial x} \left(\mu \frac{\partial^2 u_1}{\partial y^2} \right) + i\omega \varrho_m \frac{\partial v_1}{\partial y} = 0 \quad [3.45]$$

Using the thermodynamic equation that relates ϱ_1 to T_1 and p_1 ,

$$\varrho_1 = -\varrho_m \beta T_1 + \left(\frac{\gamma}{a^2} \right) p_1 \quad [3.46]$$

Where γ is the ratio of the isobaric to isochoric specific heats and a is adiabatic speed of sound. Substituting equation [3.46] in equation [3.45], we obtain,

$$\varrho_m \omega^2 \beta T_1 - \frac{\omega^2}{a^2} \gamma p_1 - \frac{d^2 p_1}{dx^2} + \frac{\partial}{\partial x} \left(\mu \frac{\partial^2 u_1}{\partial y^2} \right) + i\omega \varrho_m \frac{\partial v_1}{\partial y} = 0 \quad [3.47]$$

Now, using the expressions for u_1 (Equation [3.25]) and T_1 (Equation [3.44]) in Equation [3.47] and integrating it with respect to y between the limits 0 and y_0 , we arrive at

Rott's wave equation [39]:

$$\left(1 + \frac{(\gamma - 1)f_k}{(1 + \varepsilon_s)} \right) p_1 + \frac{a^2 \varrho_m}{\omega^2} \frac{d}{dx} \left(\frac{(1 - f_v)}{\varrho_m} \frac{dp_1}{dx} \right) + \frac{a^2 \beta}{\omega^2} \frac{(f_k - f_v)}{(P_r - 1)(1 + \varepsilon_s)} \frac{dT_m}{dx} \frac{dp_1}{dx} = 0 \quad [3.48]$$

The above equation gives an expression for the acoustic pressure p_1 in a stack given a mean temperature distribution $T_m(x)$ and other thermophysical properties of both the solid and the gas media.

Next, we will develop the governing equations for the time averaged energy flux $\dot{H}_2$ [39].

From conservation of energy,

$$\frac{\partial}{\partial t} \left(\frac{1}{2} \rho v^2 + \rho \epsilon \right) = -\nabla \cdot \left[\rho v \left(\frac{1}{2} v^2 + h \right) - K \nabla T - v \cdot \Sigma \right] \quad [3.49]$$

Where ϵ is the internal energy per unit mass and h is the enthalpy per unit mass and Σ is the viscous stress tensor, whose components are given by Equation [3.28]. In Equation [3.49], the terms on the right hand side denote the divergence of energy flux density, consisting of three terms: mass transfer by fluid motion, heat transfer and energy flux dissipation, where as the term on the left denotes rate of change of energy per unit volume [58].

The time averaged energy flux is independent along x . Neglecting the third order terms and integrating with respect to y from $y = 0$ to $y' = 0$ time averaging yields [39]:

$$\frac{d}{dx} \left[\int_0^{y_0} \overline{\rho u h} dy - \int_0^{y_0} K \frac{\partial \overline{T}}{\partial x} dy - \int_0^1 \overline{K_s \frac{\partial T_s}{\partial x}} dy' - \int_0^{y_0} \overline{(v \cdot \Sigma)_x} dy \right] = 0 \quad [3.50]$$

Here ρ is the density, u is x -direction velocity, h is the enthalpy per unit mass, K is the thermal conductivity. The overbar above the parameters indicates time average. The above quantity in square brackets is the time averaged energy flux per unit perimeter along x and is given by,

$$\frac{\dot{H}}{\Pi} = \int_0^{y_0} \overline{\rho u h} dy - \int_0^{y_0} K \frac{\partial \overline{T}}{\partial x} dy - \int_0^1 \overline{K_s \frac{\partial T_s}{\partial x}} dy' - \int_0^{y_0} \overline{(v \cdot \Sigma)_x} dy \quad [3.51]$$

Where Π is the perimeter of the stack material.

Using the relations from Equations [3.13] - [3.17] and expanding to second order, we get,

$$\int_0^{y_0} \overline{\rho u h} dy \approx \int_0^{y_0} \left(\rho_m \overline{u_1 h_m} + \rho_m \overline{u_2 h_m} + \rho_1 \overline{u_1 h_m} + \rho_m \overline{u_1 h_1} \right) dy \quad [3.52]$$

Since $\overline{u_1} = 0$, the first term on the right is zero. Also, the integrals of the second and third terms sum to zero because the second order time averaged mass flux is zero [39].

$$\int_0^{y_0} (\varrho_m \overline{u_2} + \overline{\varrho_1 u_1}) dy = 0 \quad [3.53]$$

Using,

$$dh = T ds + \frac{dp}{\varrho} = C_p dT + \frac{1}{\varrho} (1 - \beta T) dp \quad [3.54]$$

We obtain,

$$\overline{\varrho u h} \simeq \varrho_m \overline{u_1 h_1} = C_p \varrho_m \overline{T_1 u_1} + (1 - \beta T_m) \overline{p_1 u_1} \quad [3.55]$$

From Equation [3.51], only the zero order terms are significant. Therefore,

$$-\int_0^{y_0} K \frac{\partial \overline{T}}{\partial x} dy - \int_0^1 K_s \frac{\partial \overline{T_s}}{\partial x} dy' \simeq -(y_0 K + l K_s) \frac{dT_m}{dx} \quad [3.56]$$

Again from Equation [3.51],

$$\int_0^{y_0} \overline{(v \cdot \Sigma)_x} dy / \int_0^{y_0} \overline{\varrho u h} dy \sim \frac{v}{\lambda a} = \frac{1}{2} \frac{\delta_v^2}{\lambda^2} \ll 1 \quad [3.57]$$

So that the viscous term $v \cdot \Sigma$ is negligible. Hence, Equation [3.51] becomes,

$$\frac{\dot{H}_2}{\Pi} = \int_0^{y_0} [\varrho_m c_p \overline{T_1 u_1} + (1 - T_m \beta) \overline{\varrho_1 u_1}] dy - (y_0 K + l K_s) \frac{dT_m}{dx} \quad [3.58]$$

Substituting Equation [3.44] for T_m and Equation [3.25] for u_1 and integrating, we get [39]:

$$\begin{aligned}
\dot{H}_2 = & \frac{\Pi y_0}{2h\varrho_m} \text{Im} \left[\frac{d\tilde{\varrho}_1}{dx} \varrho_1 \left(1 - \tilde{f}_v - \frac{T_m \beta (f_k - \tilde{f}_v)}{(1 + \epsilon_s)(1 + P_r)} \right) \right] \\
& + \frac{\Pi y_0 C_p}{2h^3 \varrho_m (1 - P_r)} \frac{dT_m}{dx} \frac{d\varrho_1}{dx} \frac{d\tilde{\varrho}_1}{dx} \times \text{Im} \left[\tilde{f}_v + \frac{(f_k - \tilde{f}_v)(1 + \epsilon_s f_v / f_x)}{(1 + \epsilon_s)(1 + P_r)} \right] \\
& - \Pi(y_0 K + lK_s) \frac{dT_m}{dx}
\end{aligned} \tag{3.59}$$

In the above equation, tilde denotes the complex conjugate of the individual parameter, while $\text{Im}[\]$ denotes the imaginary part. This equation gives the energy flux along the direction of wave propagation in terms of pressure, mean temperature, material properties and the geometry of the device. Rott [32] obtained the result for an ideal gas and $\epsilon_s = 0$.

Additionally defining,

$$p_x = \frac{dp_1}{dx} \tag{3.60}$$

We also need a set of equations to express the acoustic power for a thermoacoustic system. Since the acoustic waves are longitudinal, the acoustic power is generated/absorbed only in the x-direction. Let us select a portion of length dx in the thermoacoustic system in along the x-direction (Figure 3.4). The average power generated/absorbed in this length is given by [39]:

$$\dot{W}_2 = \Pi y_0 \left[\left(\overline{p_1 u_1} \right)_x - \left(\overline{p_1 u_1} \right)_{x+dx} \right] \tag{3.61}$$

Using Taylor series expansion of $\left(\overline{p_1 u_1} \right)$, and neglecting higher order terms, we get,

$$\dot{W}_2 = \Pi y_0 \Delta x \frac{d}{dx} \left(\overline{p_1 u_1} \right) \tag{3.62}$$

Now, we know that both p_1 and u_1 are complex quantities and the time averaged product of these variables is:

$$\left(\overline{p_1 u_1}\right) = \frac{1}{2} \text{Re} \left[p_1 \tilde{u}_1 \right] \quad [3.63]$$

Where $\text{Re}[\]$ denotes the real part and the tilde sign denotes complex conjugate. Substituting Equation [3.63] in Equation [3.62], we get,

$$\dot{W}_2 = \frac{1}{2} \Pi y_0 \Delta x \left[p_1 \frac{d \tilde{u}_1}{dx} + \tilde{u}_1 \frac{dp_1}{dx} \right] \quad [3.64]$$

In the above equation the 2nd term can be dropped since $dp_1 / dx = i\omega \rho_m u_1$ and the real part of the term is nonexistent. Using Equation [3.25] and Equation [3.48], we get the following expression for the 1st term in Equation [3.64]:

$$\frac{d \tilde{u}_1}{dx} = \frac{-i\omega}{\rho_m a^2} \left[1 + \frac{(\gamma-1)}{(1+\epsilon_s)} f_k \right] p_1 + \frac{(f_k - f_v)}{(1-\text{Pr})(1+\epsilon_s)(1-f_v)} \beta \frac{dT_m}{dx} u_1 \quad [3.65]$$

Substituting Equation [3.65] into Equation [3.64], we get,

$$\dot{W}_2 = \frac{1}{2} \Pi y_0 \Delta x \left[\frac{-i\omega}{\rho_m a^2} \left[1 + \frac{(\gamma-1)}{(1+\epsilon_s)} f_k \right] p_1 + \frac{(f_k - f_v)}{(1-\text{Pr})(1+\epsilon_s)(1-f_v)} \beta \frac{dT_m}{dx} u_1 \right] p_1 \quad [3.66]$$

Equations [3.48], [3.59] and [3.60] are a set of five coupled equations with five variables: $\text{Re}(p_1)$, $\text{Im}(p_1)$, $\text{Re}(u_1)$, $\text{Im}(u_1)$ and T_m . These variables can be used for the design, analysis and optimization of thermoacoustic devices.

3.5 Critical Temperature Gradient

Equation [3.35] is,

$$\rho_m C_p \left(i\omega T_1 + u_1 \frac{dT_m}{dx} \right) - i\omega p_1 = K \frac{\partial^2 T_1}{\partial y^2} \quad [3.67]$$

Assuming:

- The temperature variation reduces to zero at the boundary; $T_1 = 0$
- The temperature variation has a finite value as the perpendicular distance from the solid boundary increases; ($T_\infty = \text{finite}$) [63],

We get,

$$T_1 = \left(\frac{p_1}{\rho_m C_p} - \frac{u_1}{i\omega} \frac{dT_m}{dx} \right) \left[1 - e^{-(1+i)y/\delta_k} \right] \quad [3.68]$$

From Equation [3.68] it is gathered that the temperature oscillation of a gas parcel is due to the adiabatic pressure fluctuation, and therefore the temperature variation and the motion equal the local temperature gradient dT_m / dx . This implies that the temperature at a fixed location is independent of time. This gives rise to the critical temperature gradient, which is obtained by equating the term in parentheses in Equation [3.68] to zero.

$$\nabla T_{\text{crit}} = \frac{i\omega |p_1|}{\rho_m C_p |u_1|} \quad [3.69]$$

For an inviscid standing wave engine $|dT_m / dx| > \nabla T_{\text{crit}}$ whereas for an inviscid standing wave refrigerator $|dT_m / dx| < \nabla T_{\text{crit}}$.

Using $p_1 = P_A \cos \frac{2\pi x}{\lambda}$, $u_1 = i \frac{P_A}{\rho_m a} \sin \frac{2\pi x}{\lambda}$ and $\omega = 2\pi a / \lambda$, we get,

$$\nabla T_{\text{crit}} = \frac{2\pi a^2}{\lambda} \cot\left(\frac{2\pi x}{\lambda}\right) \quad [3.70]$$

Substituting,

$$a^2 = \gamma \frac{P_m}{\rho_m} = T_m C_p (\gamma - 1) \quad [3.71]$$

$$\nabla T_{\text{crit}} = \frac{2\pi T_m C_p (\gamma - 1)}{\lambda} \cot\left(\frac{2\pi x}{\lambda}\right) \quad [3.72]$$

∇T_{crit} is the largest temperature gradient that can develop over a stack and it is larger nearer a pressure antinode and at the pressure node it is smaller. This implies that larger temperature gradients can be sustained at velocity nodes (where displacement is lower), than at velocity antinodes (where displacement is higher).

Chapter 4

Design: Parametric Study and Modeling

In this chapter, the design of the overall system is presented. The numerical modeling is done using pre-existing software, specifically written for development of thermoacoustic systems. Further, a discussion of a parametric study of the system components and variables carried out to suit the objectives is presented. The effect of different parameters on the thermoacoustic system is also discussed.

4.1 Method of Analysis

Analysis and optimization of such a thermoacoustic refrigerator is performed using the “Design Environment for Low-Amplitude ThermoAcoustic Energy Conversion” (DeltaEC) computer code [17] developed at the Los Alamos National Laboratory. DeltaEC can be used for predicting the performance of a given thermoacoustic apparatus. It also allows the user to design an engine-refrigerator system to achieve a desired performance. The code of DeltaEC is compiled using FORTRAN-77 while the graphical user interface is developed using Python programming language [66]. The code solves the wave equation for a given configuration including geometry, operating conditions, material properties, fluid properties provided by the user. The user constructs the geometry of the thermoacoustic device in DeltaEC using appropriate segments inbuilt in the software. The different

segments generally begin and end with boundary conditions. In between, they involve the geometry of the resonator and, various other sections of a thermoacoustic system such as stack materials, speakers, transducers, heat exchangers, ducts, compliances, etc. Care must be taken to enter these segments of the thermoacoustic device in the sequentially in the program from one end to the other as the location of each component with respect to the other components plays a vital role in the functioning of a thermoacoustic system.

Two coupled first order equations for pressure (p_1) and velocity (U_1) are used by DeltaEC to describe the acoustic wave equation:

$$\frac{dp_1}{dx} = -\frac{i\omega Q_m}{A} U_1 \quad [4.1]$$

$$\frac{dU_1}{dx} = -\frac{i\omega A}{Q_m a^2} p_1 \quad [4.2]$$

Equation [4.1] is derived from the momentum equation and Equation [4.2] is derived from the continuity equation [66]. The direction of propagation of the acoustic wave is discretized into small sections such that

$$p_2 = p_1 + dx \frac{-i\omega dQ_m p_1}{A} U_1 \quad [4.3]$$

$$U_2 = U_1 + \frac{-i\omega A}{Q_m a^2} p_1 \quad [4.4]$$

DeltaEC uses continuity of oscillating pressure, oscillating volumetric velocity, and the mean temperature to match the solution between two adjacent segments [17]. Inside individual segments, the program solves the wave equation appropriate for that particular segment. Knowing p_1 and U_1 from boundary conditions p_2 , U_2 , etc. can be calculated. A

detailed discussion on DeltaEC is provided by Ward W.C, Clark, J, and Swift, G. [66].

The wave equation solution proceeds as a sequence of segments from one end of the system to the other. Power sources such as electro-mechanical drivers, gas combustion, and heating elements can be modeled into the software. Using a shooting method, the software computes the solution to the wave equation for each segment by matching the complex acoustic pressure, complex volume velocity and temperature at the individual junctions. It allows the user to specify target and guess parameters. Target parameters are boundary condition parameters that the DeltaEC code attempts to converge on. Guess parameters are initial values the program uses as a starting point. These guess parameters are recalculated by DeltaEC for each and every iteration. The parameters that are typically included as the guess vectors are the frequency, beginning temperature, phase of the wave, heat into the hot heat exchanger, and heat extraction from the cold heat exchanger. Examples of the parameters considered as in the target values are heat input into the thermoacoustic system, temperature of the hot heat exchanger, temperature of the cold heat exchanger, acoustic impedance at the end conditions, etc. Examples of parameters that can be used as guess values, if so desired, are dynamic pressure, heat removed from the system, initial temperature.

4.2 Design Considerations

The objective of this study is to develop and design a cooling system to meet the cooling system requirements of Venus atmosphere and surface temperatures. To that effect there are certain target parameters which have to be met. These include, primarily, cooling power requirements to be obtained from the thermoacoustic unit and the cooling temperature to be achieved. There are also, certain parameters such as the heat addition temperature, which cannot be exceeded and the ambient heat rejection temperature that are

constraints. The heat input into the system is provided by radioisotope power source. Each power source unit provides 200 W. Multiple radioisotope power sources can be used to achieve the required heat input. Table 1 lists the design conditions used in the modeling of the thermoacoustic engine refrigerator system.

Parameter	Value
Cooling power required (W)	300
Cooling temperature to be achieved (K)	323 (50 °C)
Maximum heat addition temperature (K)	1223 (950 °C)
Ambient atmospheric heat rejection temperature (K)	443 (170 °C)
Ambient surface heat rejection temperature (K)	723 (450 °C)
Radioisotope Heat addition increments (W)	200

Table 4. 1: Design conditions for the Thermoacoustic Engine-Refrigerator

4.3 Design of the Thermoacoustic Engine-Refrigerator System

The thermoacoustic engine refrigerator system is designed to meet the targets in Table 4.1. As the system has both engine and refrigerator units, there are two different stacks present, which in turn involve two different parameters for measurement of performance. The engine section has its efficiency and the refrigerator section has its coefficient of performance (COP). These two contribute to the overall COP of the system. Therefore for a given COP, the system may have larger engine efficiency and a smaller refrigerator COP or vice versa.

A comprehensive optimization of the thermoacoustic system is undertaken to study the effect of parameters such as, pressure, geometry of the system, lengths of individual components, gas mixture ratio, stack material etc. The focus in the rest of the chapter will be the optimization of these parameters.

4.4 Length Scales

Important length scales to be considered when optimizing a thermoacoustic system are discussed. The viscous penetration depth (perpendicular to the direction of motion of the gas) and the thermal penetration depth (perpendicular to the direction of motion of the gas) as described in Equation [3.26] and Equation [3.41] respectively are important.

Another important parameter is the hydraulic ratio, r_h given by Equation (4.5):

$$r_h = \frac{y_o}{\delta_k} \quad (4.5)$$

Where, y_o is half the stack spacing and δ_k is the thermal boundary layer.

The target requirements such as cooling power, operating temperature are a function of the stack configuration and its location in the standing wave field. Some of the other influencing factors are input power, length of the stack, pressure, etc.

An indicator of suitable phasing in standing wave thermoacoustic systems is given by Rott's function (Figure 4.1). The imaginary components of the f-function are responsible for the proper phasing of the velocity and pressure components to produce thermoacoustic effects. If the hydraulic ratio is too small ($r_h \ll 1$), then the temperature fluctuations in the gas are isothermal. If the hydraulic ratio is large ($r_h \gg 1$), then the temperature fluctuations in the gas are adiabatic. In both these cases the heat transfer between the gas and solid will not generate the thermoacoustic effect. Hence, it is advisable that the optimal hydraulic ratio

[67], around 1.5-2, be maintained, other parameters are modified to achieve the maximum efficiency for desired conditions.

Figure 4.1: Plot of hydraulic ratio vs. f -function for different stack geometry.
Source: Swift GW. Thermoacoustics: A unifying perspective for some engines and refrigerators. Melville, NY: Acoustical Society of America 2002 [60].

Gas displacement amplitude (In the direction of motion of the gas)

$$|\xi_1| = \frac{|u_1|}{\omega} \quad (4.6)$$

Prandtl number:

$$P_r = \left(\frac{\delta_v}{\delta_\kappa} \right)^2 = \frac{\mu C_p}{k} \quad (4.7)$$

Where P_r is the Prandtl number and is of order unity for gases. The heat exchanger components in the thermo-acoustic system should have dimensions of the order of δ_v in order to exchange heat efficiently with the working gases.

The boundary layer thickness is vital, as the thermoacoustic effect is maximum within the thermoviscous boundary layer. The heat transfer and the generation of acoustic wave occur predominantly due to the oscillation of the gas volume within the boundary layer thickness. The thermal and viscous penetration depths are of the same order of magnitude as the boundary layer thickness. The thermal boundary layer is productive, in the sense that it leads to enhancement of heat transfer from the gas to the solid, but, on the other hand, the viscous penetration depth is dissipative and is counterproductive to the efficiency of the thermoacoustic system. The viscous drag that occurs in the viscous penetration depth generates heat due to momentum diffusion. Therefore, effort is made to reduce the viscous boundary layer. One way to achieve this is by varying the gas mixture. The choice of stack material is critical as it pre determines the optimal thermal penetration depth.

The gas displacement amplitudes $|\xi_1|$ are much larger than the thermal and the viscous penetration depths, but are quite smaller than the wavelength

Mathematically,

$$\delta_v, \delta_k \ll |\xi_1| \ll \lambda \quad (4.8)$$

4.6 Parametric Study of the system

There are numerous parameters which affect the working and performance of a thermoacoustic system. We will look at the most important ones and determine what role they play in the effective functioning of thermoacoustic device. Figure 4.2 shows a flow

chart of the design algorithm. It gives a list the various parameters that a thermoacoustic system is dependent upon. As the figure indicates all these parameters are linked to one another, some directly and others indirectly.

Figure 4.2: Design and optimization flow chart for the thermoacoustic engine refrigerator system.

For the current design, the target parameters are the constraints and are not modified. These are the cooling temperature to be achieved (323K), the cooling power required (300W), the heat addition temperature (1223K) and the heat rejection temperature (>418K for the atmospheric conditions and >723K for the surface conditions, as the heat rejection temperature has to be greater than ambient temperature on Venus). The operating conditions are various such as frequency, average pressure, dynamic pressure, type of gas,

stack material, stack geometry, stack spacing, resonator length etc. Many of these operating conditions are interdependent and have to be investigated to obtain the best performance from the system. Finally, the system has to be optimized to obtain the maximum COP and cooling power while at the same time minimizing the size.

4.7 Selection of Gas

The type of gas used as the working fluid in the standing wave refrigerator plays a very important role in its efficiency. Non-dimensional groups [68] for thermo-acoustic variables show that the thermo-acoustic power generally scales as $\rho_m a^2$ (Refer Equation (4.10)). Therefore, for a given mean pressure amplitude, high sound speeds yield high power per unit volume. The lighter gases have higher sound speeds. (E.g., Hydrogen, Helium, Helium-3 (^3He) and Neon). Light gases also have the advantage of having high thermal conductivity, which increases the thickness of thermal penetration depth, δ_x , leading to increased stack and heat exchanger dimensions and therefore stacks and heat exchangers with bigger inner spacing can be used. This is an advantage as it makes fabrication of the stack easier and more cost effective.

An important factor in selection criteria for working gases in any thermoacoustic device is the Prandtl number. We recall that Prandtl number is

$$P_r = \left(\frac{\delta_v}{\delta_k} \right)^2 \quad (4.9)$$

The Prandtl number affords a dimensionless estimate of relative effectiveness of momentum and energy transport by diffusion in velocity and thermal boundary layers [69]. The thermal boundary layer is dissipative and is productive for the thermoacoustic effect,

whereas the viscous boundary layer diffusion is counterproductive to the thermoacoustic effect. It is, therefore, beneficial to lower the Prandtl number in order to reduce the viscous dissipative effects. Adding a heavier gas has the effect of reducing Prandtl number of the mixture to below the typical value of $2/3$ for pure gases. An Argon mole fraction of 0.2 in a Helium-Argon gas mixture will reduce the Prandtl number to 0.4 and a 0.2 mole fraction of Xenon in a Helium-Xenon mixture will reduce the Prandtl number to 0.2 [60]. The addition of a heavy gas in a small amount to the lighter gas is favorable as it improves the overall efficiency of the thermodynamic system. The caveat associated with adding heavier gas to a lighter gas is that it increases the overall mass of the gas mixture, which reduces the speed of sound and hence results in a lower power density. The reduction in the speed of sound has one related beneficial effect as it reduces the size of the resonator.

Preliminary modeling (Appendix B) and other studies [58, 60] have shown that among all noble gas mixtures used as the working fluid, Helium-Xenon has the highest efficiencies. Therefore, Helium-Xenon mixture is selected as the working fluid. It is seen that as the Helium fraction in the gas mixture increases, the frequency increases (Figure 4.3). Plot of relative efficiency of the thermoacoustic engine section of the system is shown in Figure 4.4 and plot of COP of the thermoacoustic refrigerator section is shown in Figure 4.5. As discussed earlier, one can observe from these figures that there is an optimum value for He-Xe gas mixture ratio to obtain the maximum relative efficiency and coefficient of performance.

Figure 4.3: Helium-Xenon gas mixture ratio vs. frequency

Figure 4.4: Helium-Xenon gas mixture ratio vs. relative efficiency

Figure 4.5: Helium-Xenon gas mixture ratio vs. coefficient of performance

4.8 Dynamic Pressure

Dynamic pressure or the pressure amplitude is the fluctuating component of the pressure which contributes to the generation of an acoustic wave. Equation (4.10), derived by Swift [39], shows the power density per unit volume of a thermoacoustic device is proportional to:

$$\frac{\dot{H}_2}{V} \sim \frac{f}{2} \frac{T_m \beta}{(1 + \epsilon_s)} \frac{P_A^2}{\rho_m a^2} = \frac{f}{2} \frac{T_m \beta}{(1 + \epsilon_s)} \rho_m a^2 M^2 \quad (4.10)$$

Where P_A is the dynamic pressure, a is the speed of sound, f is the frequency of the oscillation, β is the thermal expansion coefficient, ϵ_s , stack heat capacity ratio, ρ_m is the mean density of the gas, M is the acoustic Mach number, and T_m is the mean temperature of the gas.

The acoustic Mach number is

$$M = \frac{P_A}{\rho_m a^2} \quad (4.11)$$

It is observed from Equation (4.10) that the power density is directly proportional to the square of the dynamic pressure. This implies that a higher dynamic pressure will deliver more power per unit volume.

To avoid non-linear effects, P_A should be limited to keep $M \leq 0.1$. In order to have high dynamic pressure and still limit the Mach number, $M < 0.1$, the term $\rho_m a^2$ should be as high as possible. Since,

$$\rho_m a^2 = \gamma p_m \quad (4.12)$$

Where γ is the ratio of specific heat capacity of gases and p_m is the average pressure. Therefore this implies that the average pressure should be as high as possible.

4.9 Average Pressure

Power density of any thermoacoustic system is directly proportional to average pressure of the oscillating gas (Equations (4.10) and (4.12)). Therefore, it is advantageous to have a high average pressure. A caveat to consider though is that the, thermal boundary layer thickness, δ_k is inversely proportional to the quadratic root of the average pressure. This is due to the density of the gas being inversely proportional to the square root of the thermal boundary layer thickness (Equation [3.41]). For an ideal gas, at a constant temperature, the density is directly proportional to the pressure of the gas (Equation [4.13])., according to the ideal gas law [70],

$$p = \rho RT \quad [4.13]$$

Where p is the pressure, ρ is the density, R is the universal gas constant and, T is the temperature of the gas. Therefore, selecting too high an average pressure will decrease δ_k to small values, which implies small stack spacing. Beyond a certain value, this can lead to labor and time intensive construction of the stack and sometimes, it may even be unfeasible.

Figure 4.6: Average Pressure vs. Dynamic Pressure

Figure 4.6 shows the variation of dynamic pressure vs. the average pressure. It is seen that as the average pressure increases, the dynamic pressure also increases linearly. Figure 4.7 shows the plot of efficiency vs. average pressure while, Figure 4.8 shows the plot of coefficient of performance vs. the average pressure. It is seen that there is an optimum value of average pressure to obtain the maximum COP for a given system. Higher average

pressure values lead to smaller and smaller thermal boundary layer thickness which in turn increases the losses for the given system.

Figure 4.7: Average Pressure vs. Relative Efficiency

Figure 4.8: Average Pressure vs. Coefficient of Performance

4.10 Frequency

Although it is not necessary to operate the thermo-acoustic refrigerator at a resonant frequency, it is desirable to do so in order to reduce the reactive forces on the system. Frequency is generally estimated starting from

$$f = \frac{a}{\lambda} \quad (4.14)$$

Where a is the speed of sound and λ is the wavelength. From the above equation it is clear that to determine the frequency, speed of sound and wavelength are required which in turn is dependent on the type of gas, boundary conditions and length of the resonator. When the system is in resonance, the whole length of the apparatus may typically be either half wavelength or quarter wavelength. A half wavelength resonator has two closed ends resulting in velocity nodes and pressure antinodes at each end of the resonator (Figure 3.3). On the other hand, a quarter wavelength resonator has one open end and one closed end. It is preferable to use a quarter wavelength resonator (Section 4.15). For a resonator with two closed ends, using Equation [3.19], and the boundary condition that the longitudinal velocity is zero at the resonator length L ,

$$\sin kL = 0 \quad (4.15)$$

From (4.14) and $k = \omega / a$,

$$k = \frac{2\pi}{\lambda} \quad (4.16)$$

Therefore, solution for Equation (4.15) is,

$$\lambda = \frac{2L}{n} ; (n=1,2,3...) \quad (4.17)$$

In the case of a quarter wavelength resonator, we have one closed end and one open end. The requisite at the open end is a pressure node. Therefore from Equation[3.18],

$$\cos kL = 0 \quad (4.18)$$

The solution is

$$\lambda = \frac{4L}{2n-1} ; (n=1,2,3...) \quad (4.19)$$

We see that in the half wavelength resonator and the quarter wavelength resonator, the fundamental modes have resonator lengths corresponding to half wavelength ($\lambda / 2$) and quarter wavelength ($\lambda / 4$) respectively.

There are two other important factors that determine the oscillating frequency of a thermoacoustic device. One is the power density. Power density is linearly proportional to the acoustic resonance frequency. Therefore it is desirable to have as large a frequency as possible. But the second factor, the thermal boundary layer thickness, δ_k , is inversely proportional to the square root of the frequency. So, to have larger stack spacing implies that the frequency should be smaller. An optimum frequency has to be selected keeping these relevant parameters in mind.

4.11 Optimization of the Stack

The stack material, being the core of any thermoacoustic device, is a key factor in determining its performance and efficiency. There are different parameters that must be given due consideration before a stack can be selected.

4.11.1 Stack Material

For the thermoacoustic effect to occur, the stack should provide local heat capacity for the thermoacoustic gas parcel, which is in thermal contact with the gas, allowing for heat transfer along the mean temperature gradient for a thermoacoustic engine. High local heat capacity ensures that adequate heat transfer occurs between the stack and the gas parcel. It is important that the stack provides the local heat capacity for the gas parcel while minimizing ordinary heat conduction along the temperature gradient and minimizing the viscous dissipation of acoustic power. The final term in energy expression (Equation [3.59])

$$-\Pi(y_0 K + l K_s) \frac{dT_m}{dx} \quad (4.20)$$

represents conduction of heat through the gas and stack material in the stack part. In the equation, Π is the perimeter of the stack, K is the thermal conductivity of the stack material, l is the half the stack plate thickness and y_0 is half the gap distance between stack plates. As can be seen this term has a negative effect on the energy flux. Solids generally have high heat capacity, therefore while selecting the material, the one with the least value of thermal conductance and highest possible specific heat capacity should be selected. The solid stack has a large cross sectional area so as to maintain modest thermal contact with the gas. Stacks can be made of different metals like stainless steel, ceramics, plastic or fiberglass depending on the application. If the stack material is highly conductive, then the heat transfer across the stack occurs through the stack material instead of the gas which results in the non generation of the thermoacoustic effect. As direction of heat transfer is against the mean temperature gradient in a thermoacoustic refrigerator, the stack material in thermoacoustic refrigerators have a lower thermal conductivity than the stack material in thermoacoustic engines.

4.11.2 Stack Location

The heat flow in the x-direction is dependent on the product of pressure and velocity. The pressure variation is needed to produce the temperature variation and the velocity is needed for momentum transfer. Considering a resonator of length $\lambda/2$ (Figure 4.9), velocity is zero at the two ends and at $x = \lambda/4$ pressure variation is small. The maximum heat flow is at the location $x = \lambda/8$ which is at $1/4^{\text{th}}$ the resonator length.

Figure 4.9: Pressure and velocity nodes and antinodes in a standing wave thermoacoustic system

Viscous and thermal losses are proportional to the local velocity (u_1^2) and local pressure (p_1^2), respectively, and the distribution of their sum is almost uniform over the length of a simple standing wave tube. These losses are quite high relative to the heat flow and can be minimized by confining the stack to the around the quarter length region of the tube. In other words, at the velocity anti-node, the viscous losses are at its highest since velocity is highest and at the pressure anti-nodes, the thermal losses are the highest.

4.11.3 Stack Geometry

Different stack geometries have been used in standing wave thermoacoustic devices [53, 58, 60, 63]. Some of them are parallel plate stacks, rectangular, hexagonal or circular pore stacks, spiral stacks and pin arrays etc. Any of these geometries can be used effectively to generate the thermoacoustic effect. The power for a stack material is given by the imaginary part of Rott's function, $\text{Im}[-f_k]$ (Equation [3.39]). Using Figure 4.1 as a reference for different kinds of stack geometry and their power density, we observe that the highest power density is obtained for the pin array stack geometry and the parallel plate geometry. The circular pore and rectangular pore geometry have lower power densities. Compared to the pin array stack geometry, parallel plate geometry is easier to manufacture and hence it selected.

4.11.4 Stack Spacing

As discussed in section 4.1 the optimal hydraulic ratio range is:

$$1.5 \leq r_h \leq 2 \quad (4.21)$$

Therefore,

$$y_o \cong 2\delta_k \quad (4.22)$$

Where y_o is the stack spacing and δ_k is the thermal boundary layer. This implies that the stack spacing should be twice the thermal boundary layer thickness for optimal performance (Figure 4.10).

Figure 4.10: Optimal stack spacing

4.11.5 Stack Length

Assuming that stack is short enough and the temperature spanned is small enough, the pressure drop across the stack is estimated by

$$\Delta p_1 \approx -\frac{i\omega Q_m \Delta x U_1}{A(1-f_v)} \quad (4.23)$$

Where Δx is the length of the stack, and the other values are evaluated at the stack midpoint.

Here we observe that increase in the length of the stack leads to a higher pressure drop across the stack, which lowers the performance of the system. Too small a Δx will not be enough to provide a good refrigeration effect. Therefore an optimal length of the stack has to be selected.

The numerical results obtained (Figures 4.11 and 4.12) validate the effect of engine stack length and the refrigerator stack length on the overall COP of the thermoacoustic device. It is seen that the maximum COP is obtained for an engine stack length of 0.16 m and a refrigerator stack length of 0.12 m.

Figure 4.11: Effect of engine stack length on the COP

Figure 4.12: Effect of refrigerator stack length on the COP

Additionally, since standing wave refrigerators pump heat proportional to the product $[p_1 U_1]$, they are least efficient at the pressure antinodes, where velocity is negligible, and also at pressure nodes. Therefore, the length of stack should not be such that it is contained between two antinodes (or two nodes). Additionally, high heat transfer rate requires high velocity and high efficiency requires low velocity. An optimum value, considering the above factors, puts the stack starting about $\lambda/20$ from the nearest pressure antinode (Typical value suggested by Swift [60]). Placing the other end at a distance of anywhere between $\lambda/10$ to $\lambda/6$ from the velocity antinode, an optimum length of the stack tubes is between $1/5^{\text{th}}$ to $1/15^{\text{th}}$ of the resonator length.

4.12 Heat Exchangers

Another critical component of any thermoacoustic device is the heat exchanger. The heat exchanger(s) transfer heat between thermoacoustic device; specifically, the stack, and the external surroundings. Depending upon the type of thermoacoustic device, there can be around two to four heat exchangers in the system. Appropriate design of the heat exchanger is essential, as inadequate heat transfer; due to an inefficient heat exchanger, can lead to lowered performance of the device.

Development of heat exchangers for heat transfer in oscillating flow is complicated. Only a few studies have been carried out to study heat transfer in oscillating flows [40, 41, 71], but otherwise, all the literature is geared towards steady state heat transfer.

There are four heat exchangers in the current design. The first one (also known as the engine hot heat exchanger) provides heat input into the device to the engine stack thereby facilitating the generation of acoustic energy by the stack. This heat exchanger is designed to be made from a heating wire and supply around 800W of heat input into the

system. The second heat exchanger (also known as the engine cold heat exchanger) rejects into the surroundings. The third heat exchanger known as the refrigerator hot heat exchanger exchanges heat with the refrigerator stack extracting heat from the stack and rejecting it to the surroundings. The second and third heat exchangers are designed to be fin and tube heat exchangers. The fourth heat exchanger known as refrigerator cold heat exchanger is the load on the refrigerator. Heating element is used to provide the load into the system. Swift [72] discusses that the length of the heat exchanger should be equal to the gas displacement amplitude $|\xi_1|$ (Equation (4.6)).

4.13 Heat Addition Temperature

Heat addition temperature is the temperature at which heat is input into the engine side of the thermoacoustic system. From the second law of thermodynamics, the higher the heat addition temperature, higher the efficiency of the thermoacoustic engine, but due to the system constraints, the higher limit of the heat addition temperature is 1223K.

However, the COP is a function of the heat addition temperature or the hot side heat exchanger temperature of the engine section. It is seen that the higher COP is obtained at lower temperatures and the COP decreases at higher temperatures (Figure 4.13). This is an expected trend as we are putting more energy into the system at higher temperatures for the obtaining the same output. However, if we observe the variation of cooling power obtained as a function of heat addition temperature (Figure 4.14) it is seen that the higher cooling power is initially obtained at higher temperatures. As the temperature is increased further, there is no related increase in cooling power. This is due to the fact that as we increase the heat addition temperature for a given system, the speed of sound and other temperature

dependent factors change. Beyond a certain threshold, the system properties are not optimal for the higher heat addition temperature.

As there is an inverse correlation between the COP and cooling power with respect to heat addition temperature, as a compromise between the COP and cooling power, an intermediate value of heat addition temperature should be selected for the hot side heat exchanger.

Figure 4.13: Effect of heat addition temperature on COP

Figure 4.14: Effect of heat addition temperature on cooling power

4.14 Heat Rejection Temperature

Heat rejection temperature is the temperature at which heat is rejected by the ambient heat exchangers of the thermoacoustic system. There are two heat exchangers which operate at the heat rejection temperature. One is the engine side ambient heat exchanger and the other one is the refrigerator side heat exchanger. The goal is to achieve a heat rejection temperature at least slightly higher than the ambient temperature of Venus (both for high altitude and surface operation), so that these ambient heat exchangers can reject heat into the atmosphere.

As with any thermodynamic refrigerator, lower the heat rejection temperature, higher the cooling power output. COP of the system also decreases at higher heat rejection temperatures (Figure 4.15).

Figure 4.15: Optimization of middle heat exchanger temperature

4.15 Resonator Geometry

The resonator geometry is also designed to yield maximum efficiency. The cross sectional area of the resonator is determined by the cross sectional area of the stack and the length of the resonator is determined by the resonant frequency. Typically, the resonator length is selected to be either $\lambda / 2$ (half wavelength resonator) or $\lambda / 4$ (quarter wavelength resonator). If the resonator has fixed or closed boundaries at both ends, the amplitude of the wave is forced to zero and velocity nodes are formed at both ends. These nodes occur at multiples of $\lambda/2$ ($\lambda/2$, λ , $3\lambda/2$, 2λ , and so on) and therefore these resonators are known as half wavelength resonators. On the other hand if the resonator has an open end, then maximum amplitude develops at the open end, or, an antinode is formed at open end. The first node occurs at $\lambda/4$ from the open end and therefore these resonators are called quarter wavelength resonators. Further nodes develop at additional half wavelength, i.e., at $3\lambda/4$,

$5\lambda/4$, $7\lambda/4$, and so on. Quarter wavelength resonators are preferred as the energy dissipated by the resonator is directly proportional to its surface area. Hence, a half wavelength resonator will dissipate twice as much energy as a quarter wavelength resonator.

Chapter 5

Results and Discussion

The final design is presented and discussed in this chapter. As mentioned earlier, the thermoacoustic system modeling software, DeltaEC, is used to design and optimize the thermoacoustic engine refrigerator system. The discussion also includes optimization of various parameters to meet the requirements.

5.1 Design of Venus-high-altitude system

Based upon the objectives of this study, a thermoacoustic engine refrigerator system is designed and optimized to suit a Venus mission. DeltaEC, a thermoacoustic system designing software, has been used to investigate various parameters including pressure, frequency, and type of gas, etc, to obtain optimum performance of the standing wave thermoacoustic engine refrigerator system for the desired conditions (Table 5.1). It is found that high pressures delivered higher efficiencies and power output, provided the stack material spacing is appropriately small. Also, light gases such as Helium when mixed with small quantities of heavier gas such as Xenon (10%-35%), provide higher efficiencies. Stack material selection is of vital importance as it is the crux of any thermoacoustic device. For the current design, ceramic stack material is chosen as it has a low thermal conductivity and high specific heat capacity which are requisites for a good stack material. Other commonly

used refrigerator stack materials such as Kapton® and Mylar® cannot be used in this design as they have low melting points. A nickel chromium super alloy - Inconel® - is used for the body of the resonator as it has a low value of thermal conductivity while being capable of withstanding extremely high pressures. The system has an overall coefficient of performance (calculated using Equation [3.10]) of 0.19, which is comparable to other thermoacoustic engine refrigerator systems (0.16)[63].

Parameter	Value
Heat Addition Temperature (K)	1223
Heat Rejection Temperature (K)	443
Cooling Temperature (K)	323
Heat Added (W)	800
Cooling Power (W)	150
Heat Rejected by Engine (W)	591
Heat Rejected by Refrigerator (W)	388
Total Heat Rejected (W)	979
Relative Engine Efficiency	0.24
Overall COP	0.19
Overall Length (m)	1.52
Volume of the System (m ³)	5.27E-02

Table 5.1: Parameters of the thermoacoustic engine refrigerator system

Figure 5.1: Schematic diagram of the high altitude thermoacoustic engine refrigerator system.

5.2 Overall Design

Figures 5.2 and 5.3 show the 3-D assembly of the designed thermoacoustic system drawn using SolidWorks™ (Dassault Systèmes, SolidWorks Corp., Vélizy, France) and the 2-D assembly drawing of the thermoacoustic engine refrigerator system respectively.

Figure 5.2: 3-D assembly of the Thermoacoustic Engine Refrigerator System

Figure 5.3: The Thermoacoustic Engine Refrigerator System (All dimensions are in inches)

5.3 Design of Venus-surface system

Another goal of this study is to determine whether a thermoacoustic system could be developed to cool down electronic components on the surface of Venus. The earlier mentioned design parameters were utilized to develop a system for a Venus-surface system. The details of the Venus-surface system are discussed in the subsequent sections.

5.4 Design constraints for Venus-surface system

For a thermoacoustic refrigerator to perform efficiently, it is important that the temperature difference across the stack material be modest. A large temperature drop, across the refrigerator stack leads to conduction of the heat through the stack material from the hot end to the cold end, thus leading to non-generation of the thermoacoustic effect or leads to the refrigerator stack acting as a thermoacoustic engine and generating acoustic waves. In such a design, the temperature drop of 400°C is not feasible for a standing wave thermoacoustic engine refrigerator system while delivering 300W of cooling. Therefore, it is decided to build the multiple units of thermoacoustic refrigerator system and then stage them efficiently to achieve the required cooling temperature and cooling power.

5.5 Overall Design of the Venus Surface System

The required cooling power of 300 W and required cooling temperature range from 450°C to 50°C could not be achieved with one unit. Three units in series are required to achieve the cooling temperature of 50°C . But each of these units generates a cooling power of 150 W. Therefore, a combination of 2 systems (each containing 3 units in series) in parallel is required to meet the estimate. For each system, based on the optimization carried out in Chapter 4, different parameters are selected accordingly. The engine stack material

and the refrigerator stack material are selected to be ceramic stack. The gas used as the working fluid is a mixture of Helium and Xenon. The average pressure is selected to be 6MPa.

The refrigerator section of unit 1 operates between the temperatures 323K and 443K generating 150W of cooling power (Figure 5.4).

Figure 5.4: Schematic diagram of the proposed unit 1 of the thermoacoustic engine refrigerator system.

The refrigerator section of unit 2 operates between the temperatures 443K and 580K generating 150W of cooling power (Figure 5.5).

Figure 5.5: Schematic diagram of the proposed unit 2 of the thermoacoustic engine refrigerator system.

Figure 5.6: Schematic diagram of the proposed unit 3 of the thermoacoustic engine refrigerator system.

The refrigerator section of unit 3 operates between the temperatures 580K and 743K generating 150W of cooling power (Figure 5.6).

Table 5.2 lists the important parameters for the three units. The heat addition temperature is maintained at the maximum permissible value, so as to obtain the maximum efficiency. The heat added into the engine section of the system is higher for the 3rd unit as it is the least efficient amongst the three units due to the unit operating between a smaller temperature difference between the heat source and the heat sink. The overall COP is calculated using Equation [3.10]. The relative engine efficiency is the ratio of the engine efficiency (η) to the Carnot efficiency.

Unit	1	2	3
Heat Addition Temperature (K)	1223	1223	1223
Heat Rejection Temperature (K)	443	580	743
Cooling Temperature (K)	323	443	580
Heat Added (W)	800	800	1200
Cooling Power (W)	150	150	150
Heat Rejected by Engine (W)	591	571	771
Heat Rejected by Refrigerator (W)	388	403	607
Total Heat Rejected (W)	979	974	1378
Engine Efficiency	0.24	0.21	0.14
Carnot Efficiency	0.64	0.52	0.39
Relative Engine Efficiency	0.37	0.39	0.36
Overall COP	0.19	0.19	0.13
Overall Length (m)	1.52	1.52	1.56

Table 5.2: Parameters of the three units of the thermoacoustic engine refrigerator system

This three unit arrangement can be successful in meeting the cooling temperature requirement if one does not consider the heat rejected by the thermoacoustic system itself. But each the thermoacoustic system needs to reject large quantities of heat ($\sim 1000\text{W}$); which is the heat utilized mainly by the thermoacoustic engine part of system to produce the thermoacoustic effect. This additional heat also needs to be removed to the ambient Venus conditions. The situation becomes complicated since it is realized that the first and the second thermoacoustic units reject heat at temperatures much lower than the Venus ambient temperature. This heat rejection temperature should be greater than the ambient temperature of Venus surface for this to occur. Therefore, further cooling units are required which will “cool down” this additional heat generated.

One Unit 1 system rejects a total amount of 1000 W of energy at 443K . As this is too big to be handled by a single unit, it is proposed that the rejected heat be split up into 7 equal chunks of approximately 150W each. These are then rejected at 580K using Unit 2 systems. The number of Unit 2 systems required is 7. Similarly, heat rejected by each of these systems is further divided into chunks of 150W and then rejected at 743 K using 7 systems of Unit 3 for each Unit 2 system. It is observed that the number of systems follow the geometric progression, given by the formula,

$$a_n = a r^{n-1} \quad (5.1)$$

where a_n is the n^{th} term of the series, a is the first term and r is common ratio and the sum of the series is given by

$$S_n = \frac{a(r^n - 1)}{r - 1} \quad (5.2)$$

In our case, $r = 7$ and $n = 3$. Therefore the number of units required to effectively produce 150 W of cooling between 450°C and 50°C is 57.

To produce a cooling power of 300 W, 114 units are required, or two assemblies of 57 systems in parallel. Figure 5.7 shows the schematic diagram of one of assemblies.

Figure 5.7: Schematic diagram of the heat flow in the overall system

800W of Heat is supplied to Unit 1 at 1223K, which generates the thermoacoustic wave and generates a cooling of 150W at 323K. The unit rejects overall about 1000W of heat at 443K. Now this cannot be directly rejected into the ambient Venus atmosphere as the ambient temperature is about 723K and the rejection temperature needs to be higher. Therefore, the heat rejected by Unit 1 is divided and sent into 7 different systems of Unit 2 thermoacoustic engine-refrigerator systems. Each of these systems can handle 150W of cooling. Each of the 7 different Unit 2 thermoacoustic systems is supplied with 800W of cooling. Each Unit 2 system rejects about 1000 W of energy as well, but at 580K. This in turn is cooled down by 7 Unit 3 thermoacoustic engine refrigerator systems. The Unit 3 systems also have cooling power wattage of 150W and they reject heat at 743K, which is higher than the ambient temperature of Venus of 723 K.

Figure 5.8 shows all 57 units used to cool the temperature of the electronics from 723K to 323K providing an overall 150W of cooling. To generate 300W of cooling, 2 such arrangements in parallel are to be used.

Chapter 6

Design of a Prototype System

In this chapter we will discuss the design of a prototype of the thermoacoustic engines-refrigerator system.

6.1 Considerations for fabrication

The optimized design discussed in the previous chapter is very sophisticated and expensive to manufacture. Therefore, a relatively less sophisticated model is developed which can be used as a prototype to verify the thermoacoustic refrigerator design using a thermoacoustic pulse engine, but at a lower cooling temperature and a lower cooling power output. The geometries of the two different designs are inherently similar, with the individual modifications being discussed below.

6.2 Components of the Engine-Refrigerator System

The following section describes the system components as modeled in DeltaEC. Overall, the components of the thermoacoustic engine refrigerator system are two stacks (one engine stack and one refrigerator stack), four heat exchangers and the resonator body.

6.2.1 Engine Stack Material

The engine stack material, Corning Celcor®, is selected and is readily and manufactured by Corning Incorporated, New York, NY. Corning Celcor® is generally used

as an automobile catalytic converter, but since it matches our need for a stack material, it is an inexpensive alternative. Other materials such as polymers are unsuitable because of their low melting point. Metals such as stainless steel, etc. have higher thermal conductivity than that of ceramic materials which leads to lower efficiencies.

The melting point for Corning Celcor® is 1200 °C for continuous usage but it can also withstand temperature spikes up to 1400 °C. This falls well within our temperature maximum. These substrates are manufactured at different cell densities ranging from 400 cells per square inch to 900 cells per square inch which determines the stack spacing. For smaller stack spacing, it is recommended to use substrates with higher cell densities. For our purpose we select the honeycomb ceramic stack material having a cell density of 900 per square inch which gives us a stack spacing of 0.425mm.

6.2.2 Refrigerator Stack Material

The refrigerator stack material can be fabricated from Kapton® (DuPont, Wilmington, Delaware) sheets. The Kapton® sheets are 0.0015” thick. Teflon coated wire 0.03” in diameter are glued on to the sheets and then the sheets are rolled into a spiral forming the refrigerator stack. These Kapton® polymer sheets have a thermal conductivity of 0.19 W/m-K.

6.2.3 Engine Hot Heat Exchanger

The engine hot heat exchanger utilizes a Nickel-Chromium (Ni-Cr) heating element capable of delivering 500 W of heating to the engine stack material. The heating element is a ribbon instead of a wire as it provides greater surface area for heat transfer to the engine stack. Grooves are designed to be cut into the engine stack on to which the Ni-Cr heating element is wound across in a serpentine fashion.

6.2.4 Middle Heat Exchangers

The two middle heat exchangers are designed to be made from copper tubing and fins similar to the procedure given in [73]. Alternatively, they could also be fabricated by welding or soldering the individual fins together (for lower temperature applications). These heat exchangers are responsible for taking heat out of the thermoacoustic-engine refrigerator system. Figure 6.1 shows a schematic diagram of the heat exchanger. These heat exchangers are made from copper fins with centers spaced 0.0444" apart. A 1/8" copper tubing is wound across the surface of these fins. A coolant is run through these tubes to reject heat (300 W) from the system.

Figure 6.1: Schematic diagram of the middle heat exchanger

6.2.5 Cold Heat Exchanger

The cold heat exchanger provides the load to the thermoacoustic refrigerator. It supplies the total amount of heat that needs to be eventually removed by the thermoacoustic refrigerator. As in the case of Hot Heat Exchanger, heating wire elements (Ni-Cr) are designed to be used as heat exchanger capable of delivering 50 W of heat into the refrigerator stack. They are wound on to the end of the refrigerator stack.

6.2.6 Resonator Geometry

The resonator geometry had to be modified as well to bring it within budgetary constraints. It is desired to build the unit out of a material which can withstand high temperatures and pressure without deforming. At the same time, the material should also have low thermal conductivity to prevent heat loss through its surface. Materials that meet these requirements are high performance alloys, and, in particular, Inconel® (Special Metals Corporation, New Hartford, NY). Inconel has been used previously [53, 63] in thermoacoustic devices, but a drawback is that Inconel is relatively expensive. Therefore an alloy of stainless steel (SS 304) is selected for the fabrication of the system.

The resonator bulb was designed to be a sphere to obtain maximum compliance. Again, manufacturing a metal sphere was deemed to be expensive and hence, the compliance was modified to be a cylinder and a standard pipe size is selected for the compliance.

Figure 5.2 shows the assembly of the prototype thermoacoustic engine refrigerator system and Figure 5.3 shows the schematic diagram with different parts of the system.

Figure 6.3: Schematic diagram of the prototype thermoacoustic engine refrigerator system.

6.3 Suggested Measurements

The following measurements are recommended to evaluate the performance of the system and validate the design.

- Temperature measurements at different locations along the thermoacoustic engine refrigerator system, including, various points along the engine and the refrigerator stack, heat exchangers and the ends of the system. The temperature measurements are to be made using thermocouples.
- Average and dynamic pressure measurements are to be made using a pressure transducer.
- Using power analyzers and power supplies, the power into the system can be controlled and monitored.

Chapter 7

Conclusions and Recommendations

This chapter summarizes the work carried out and presented in the previous chapters. Also, suggestions for future work and recommendations for improvement are reported.

7.1 Conclusions

As mentioned previously, most of the literature in the thermoacoustic area deals with either only a thermoacoustic refrigerator or a thermoacoustic engine. Studies in the combined thermoacoustic engine refrigerator systems area are only a handful and this work is the only one in the open literature to use standing wave thermoacoustic engine refrigerator systems to cool systems from high temperatures. This work is also focused on designing a system or a cascaded system which delivers the required cooling effect for extreme temperatures. Based upon the objectives of this study, a thermoacoustic engine refrigerator system is designed and optimized to suit a Venus high altitude mission with the secondary objective of designing an optimized system for a Venus surface mission.

Using DeltaEC, a thermoacoustic system designing software, different parameters such as pressure, frequency, type of gas, etc, are investigated to obtain optimum performance of the system for the desired conditions. It is found that high pressures

delivered higher efficiencies and power output, provided the stack material spacing is appropriately small. Also, light gases such as Helium when mixed with small quantities of heavier gas such as Xenon (10%-35%), provide higher efficiencies. Stack material selection is of vital importance as it is the crux of any thermoacoustic device. For the current design, ceramic stack material is chosen as it has a low thermal conductivity and high specific heat capacity which are requisites for a good stack material. Other commonly used refrigerator stack materials such as Kapton® and Mylar® cannot be used in this design as they have low melting points. A nickel chromium super alloy - Inconel® - is used for the body of the resonator as it has a low value of thermal conductivity while being capable of withstanding extremely high pressures.

The primary objective of designing a system to deliver a cooling power of 300 W and operate between the temperature of 145 °C and 50 °C is achieved by cascading two units (in parallel as each unit provides a cooling power of 150 W). The designed thermoacoustic unit is about 1.52 m long which is an acceptable dimension with respect to space mission constraints.

The designed Venus high altitude thermoacoustic engine-refrigerator system has an overall COP of 0.19. This is comparable to other energy conversion devices used for cooling such as standing wave thermoacoustic engine-refrigerator systems [63]. A combination Stirling cycle power converter [13] and a refrigeration system built for similar purposes had an overall COP of about .09. The power converter produced 400 W using a radioisotope heat source (efficiency 0.23) and was combined with a refrigerator system (efficiency 0.38) resulting in a COP of 0.09. It has to be kept in mind that the Stirling power converter was designed to operate at slightly different conditions than our system. For a thermoelectric power generation system, which converts heat into electricity, the efficiency is about 5% [13]

without including the efficiency involved in the refrigeration efficiency. A comparison (Table 7.1) shows the efficiencies of different systems.

Energy Conversion System	Overall COP
Venus high altitude thermoacoustic engine-refrigerator System	0.19
Standing Wave thermoacoustic system [63]	0.16
Stirling power converter system [13]	0.09
Thermoelectric energy conversion (Without Refrigeration) [13]	0.05

Table 7.1: Comparison of various energy conversion systems

The required cooling power of 300 W and required cooling temperature range from 450 °C to 50 °C for the Venus surface mission, however, could not be achieved with two units. So, in this case individual units were designed which achieve the result when cascaded in series to obtain the required temperature drop and in parallel to obtain the required cooling power. The units individually have an overall coefficient of performance of 0.19, 0.19 and 0.13 which is comparable to other thermoacoustic engine refrigerator systems [63]. Overall 114 systems are required to be cascaded in series and parallel to achieve the required temperature and power.

7.2 Recommendations for future work

There are some considerations that should be taken into account for future studies which could not be undertaken due to inherent limitations, in the DeltaEC software. For instance, radiation cannot be modeled into the design. The resonator is assumed to be ideal whereas in reality it is not. At high temperature radiation effects increase and the losses due

to radiation may become significant. Dissipative effects due to convection and radiation also cannot be modeled. Boundary conditions are generally assumed ideal, where as in reality, that is not the case. Development of more sophisticated stack materials suitable for such space applications is recommended.

A cost effective scaled down prototype has been designed. Testing should be carried out on the prototype system to obtain and verify the operational parameters. Based on the results obtained from testing the prototype model, design modifications should be made to the actual design. And then, an actual unit which is designed for the Venus system should be built and tested as that will give a better estimate of the performance.

Appendix A – Ambient Heat Exchanger

Calculations

The two middle heat exchangers (Section 6.2.4) are tasked with rejecting heat to the ambient. Each heat exchanger rejects about 300 W. Water is used as the cooling fluid. The amount of heat transferred away by water in the copper tubes due to convection is given by [70]:

$$q_{\text{water}} = \dot{m}_{\text{water}} C_{p,\text{water}} (T_{\text{exit}} - T_{\text{inlet}}) \quad [\text{A.1}]$$

Where $\dot{m}_{\text{water}}$ is the mass flow rate of water through the copper tubing, $C_{p,\text{water}}$ is the specific heat of water, T_{exit} is the temperature of the water at the exit and T_{inlet} is the inlet water temperature.

The pressure drop in the copper tubing is given by [70]:

$$\Delta p = \left(f_f \frac{L}{D} + \sum K_L \right) \frac{\rho_{\text{water}} U^2}{2} \quad [\text{A.2}]$$

Where f_f is the friction factor, $\frac{L}{D}$ is the ratio of overall length of tube to the diameter of the tube, ρ_{water} is the density of water, K_L is the minor loss coefficient, and U is the flow velocity. The length of the copper tubing through the heat exchanger is 0.64 m and the inner diameter is 0.0024 m.

The Reynolds number is given by:

$$\text{Re} = \frac{UD}{\nu_{\text{water}}} \quad [\text{A.3}]$$

Where ν_{water} is the kinematic viscosity of water.

The Reynolds number for the given conditions is 1750, which implies that the flow is laminar. The friction factor, f_f , is given by [70]:

$$f_f = \frac{64}{\text{Re}} = 0.037 \quad [\text{A.4}]$$

The overall pressure drop is calculated to be 38.3 kPa (Table A.1).

Parameter	Value
Length of the tubing (m)	0.64
Inner diameter of the tubing (m)	0.0024
L/D	267
Specific heat of water, $C_{p,\text{water}}$ (J/g.K)	4.18
Heat transferred by water, q_{water} (W)	300
Exit water temperature, T_{exit} (K)	313
Inlet water temperature, T_{inlet} (K)	293
Mass flow rate of water, $\dot{m}_{\text{water}}$ (kg/s)	2.39×10^{-3}
Kinematic viscosity of water, ν_{water} (m ² /s)	0.726×10^{-6}
Density of water, ρ_{water} (kg/m ³)	995
Velocity of water, U (m/s)	0.53
Reynolds number, Re	1750
Friction factor, f_f	0.037
Pressure drop, Δp (kPa)	38.3

Table A.1: Middle heat exchanger parameters for heat transfer

Appendix B - Optimization Results of Gas Mixture Ratio

Different gases can be used as working fluids in a thermoacoustic system (Section 4.7). As discussed earlier, lighter gases have higher sound speeds which tend to give a higher power output per unit volume. They also have a higher thermal conductivity which means that the thermal boundary layer is bigger, leading to easier fabrication of stacks. Also, it is desirable to have a low Prandtl number so that viscous dissipation losses are minimized. This is achieved by adding a small fraction of heavy gas to light gas. Due to this, pure gases such as Neon, Nitrogen, Argon and Carbon dioxide did not yield comparable results. Hydrogen is not used because of its flammable nature.

B.1 Optimization using Helium-Argon gas mixture as working fluid

Among the other gas mixtures tried, Helium-Argon mixture yielded comparable results to Helium-Xenon mixture. Initially the percentage of helium and argon in the mixture is varied and its effect on the COP is plotted. Highest COP is obtained (Figure B.1) for a mixture containing 75% Helium and 25% Argon.

Figure B.1: Variation of COP with change in the composition of He-Ar mixture

Similar to the results obtained with Helium-Xenon mixture (Section 4.13), it is seen that the higher COP is obtained at lower heat addition temperatures and the COP decreases at higher heat addition temperatures (Figure B.2). It is also observed that higher cooling power is obtained at higher heat addition temperatures (B.3). As there is an inverse correlation between the COP and cooling power with respect to heat addition temperature, as a compromise between the COP and cooling power, an intermediate value of heat addition temperature should be selected for the hot side heat exchanger. The highest efficiencies are obtained around 850K (Figure B.2) while the highest cooling power obtained is around 1100K (Figure B.3). These plots indicate that as temperature increases though the cooling power increases the COP drops. Therefore, as a compromise between the overall COP and cooling power, an intermediate value of should be selected for the hot side heat exchanger.

Figure B.2: Effect of heat addition temperature on COP

Figure B.3: Effect of heat addition temperature on cooling power

Optimization of middle heat exchanger temperature (Figure B. 4) is performed at different hot heat exchanger temperatures. This is the ambient heat exchanging temperature which rejects heat to the surroundings. It is seen that for a constant hot side heat exchanger temperature, COP decreases with increasing middle heat exchanger temperature. It is also observed that COP decreases with increasing hot heat exchanger temperature. This optimization is performed keeping the cold side heat exchanger temperature of the refrigerator section at 313K which is the required cooling temperature.

Figure B.4: Optimization of middle heat exchanger temperature

Another interesting plot is the effect of variation of duct length in between the middle heat exchanger on the COP (Figure B.5). It is seen that as the duct length is increased COP linearly decreases.

Figure B.5: Variation of duct length in between middle heat exchangers

Appendix C - DeltaEC Files

1. Prototype design of Thermoacoustic Engine-Refrigerator System

```

!Created with DeltaEC version 6.2b3 under win32, using Win 5.1.2600
(Service Pack 3) under Python DeltaEC.
!----- 0 -----
BEGIN
  1.0000E+06 a Mean P Pa
    100.78 b Freq Hz G
    717.82 c TBeg K G
  4.2610E+04 d |p| Pa G
    0.0000 e Ph(p) deg
    0.0000 f |U| m^3/s
    90.000 g Ph(U) deg
    0.8500 j nL
HeXe Gas type
!----- 1 -----
HARDEND
    0.0000 a R(1/z) 4.2610E+04 A |p| Pa
    0.0000 b I(1/z) 0.0000 B Ph(p) deg
    0.0000 c Htot W 0.0000 C |U| m^3/s
    0.0000 D Ph(U) deg
    0.0000 E Htot W
    0.0000 F Edot W
    0.0000 G R(1/z)
    0.0000 H I(1/z)
!----- 2 -----
DUCT
  1.0260E-02 a Area m^2 Mstr 4.2186E+04 A |p| Pa
    0.35903 b Perim m 2a 4.6927E-03 B Ph(p) deg
    0.14573 c Length m G 2.4237E-02 C |U| m^3/s
    -90.266 D Ph(U) deg
    0.0000 E Htot W
    -2.4159 F Edot W
stainless Solid type
!----- 3 -----
HX Hot HX
sameas 2a a Area m^2 4.2035E+04 A |p| Pa
    0.4000 b GasA/A 1.2866E-02 B Ph(p) deg
    1.0000E-02 c Length m 2.4923E-02 C |U| m^3/s
    3.0000E-03 d y0 m -90.326 D Ph(U) deg
    500.00 e HeatIn W 500.00 E Htot W
    800.00 f SolidT K =3H -3.099 F Edot W
    717.82 G GasT K
copper Solid type 800.00 H SolidT K
!----- 4 -----
STKRECT Engine Stack

```

sameas	2a	a	Area	m^2		4.0399E+04	A	p	Pa
	0.81768	b	GasA/A		4def	0.72078	B	Ph(p)	deg
	0.1000	c	Length	m		3.2492E-02	C	U	m^3/s
	4.2500E-04	d	aa	m	Mstr	-83.134	D	Ph(U)	deg
	4.5000E-05	e	Lplate	m	Mstr	500.00	E	Htot	W
	4.2500E-04	f	bb	m	Mstr	70.261	F	Edot	W
						717.82	G	TBeg	K
						356.26	H	TEnd	K
celcor			Solid type						
!----- 5 -----									
HX Ambient HX - Hot Side									
sameas	2a	a	Area	m^2		4.0270E+04	A	p	Pa
	0.7100	b	GasA/A			0.73019	B	Ph(p)	deg
	5.0800E-03	c	Length	m		3.3151E-02	C	U	m^3/s
	4.0000E-04	d	y0	m		-83.451	D	Ph(U)	deg
	-284.65	e	HeatIn	W	G	215.35	E	Htot	W
	350.00	f	SolidT	K	=5H	67.669	F	Edot	W
						356.26	G	GasT	K
						350.00	H	SolidT	K
copper			Solid type						
!----- 6 -----									
DUCT									
	1.0260E-02	a	Area	m^2	Mstr	3.7651E+04	A	p	Pa
	0.35915	b	Perim	m	6a	0.42852	B	Ph(p)	deg
	0.1270	c	Length	m		5.2467E-02	C	U	m^3/s
						-85.706	D	Ph(U)	deg
						215.35	E	Htot	W
						66.589	F	Edot	W
stainless			Solid type						
!----- 7 -----									
HX Ambient HX - Cold Side									
sameas	6a	a	Area	m^2		3.7444E+04	A	p	Pa
	0.7100	b	GasA/A			0.45697	B	Ph(p)	deg
	5.0800E-03	c	Length	m		5.3086E-02	C	U	m^3/s
	4.0000E-04	d	y0	m		-85.864	D	Ph(U)	deg
	-284.65	e	HeatIn	W	G	-69.298	E	Htot	W
sameas	5f	f	SolidT	K		63.777	F	Edot	W
						356.26	G	GasT	K
						350.00	H	SolidT	K
copper			Solid type						
!----- 8 -----									
STKSLAB Cooler Stack									
sameas	6a	a	Area	m^2		2.8653E+04	A	p	Pa
	0.80892	b	GasA/A		8de	1.1165	B	Ph(p)	deg
	0.2000	c	Length	m		7.5385E-02	C	U	m^3/s
	7.0000E-04	d	y0	m	Mstr	-87.842	D	Ph(U)	deg
	2.5400E-05	e	Lplate	m	Mstr	-69.298	E	Htot	W
						19.625	F	Edot	W
						356.26	G	TBeg	K
						319.38	H	TEnd	K
mylar			Solid type						
!----- 9 -----									
HX Cold HX									
sameas	6a	a	Area	m^2		2.8380E+04	A	p	Pa
	0.7100	b	GasA/A			1.3120	B	Ph(p)	deg
	4.0000E-03	c	Length	m		7.5851E-02	C	U	m^3/s
	1.9000E-04	d	y0	m		-87.911	D	Ph(U)	deg
	50.000	e	HeatIn	W		-19.298	E	Htot	W
	320.00	f	SolidT	K	=9H	14.604	F	Edot	W

				319.38	G	GasT	K
copper	Solid type			320.00	H	SolidT	K
!----- 10 -----							
DUCT							
sameas	6a a Area	m^2		2.5629E+04	A	p	Pa
sameas	6b b Perim	m		1.2436	B	Ph(p)	deg
6.4500E-02	c Length	m		8.2660E-02	C	U	m^3/s
				-87.988	D	Ph(U)	deg
				-19.298	E	Htot	W
stainless	Solid type			14.200	F	Edot	W
!----- 11 -----							
CONE							
sameas	6a a AreaI	m^2		2.1357E+04	A	p	Pa
sameas	6b b PerimI	m		1.1231	B	Ph(p)	deg
5.0000E-02	c Length	m		8.5543E-02	C	U	m^3/s
sameas	12a d AreaF	m^2		-88.021	D	Ph(U)	deg
sameas	12b e PerimF	m		-19.298	E	Htot	W
stainless	Solid type			13.645	F	Edot	W
!----- 12 -----							
DUCT							
2.9190E-03	a Area	m^2	Mstr	1.1472E+04	A	p	Pa
0.19154	b Perim	m	12a	-177.03	B	Ph(p)	deg
0.2000	c Length	m		8.6652E-02	C	U	m^3/s
				-88.063	D	Ph(U)	deg
				-19.298	E	Htot	W
stainless	Solid type			8.9554	F	Edot	W
!----- 13 -----							
CONE							
sameas	12a a AreaI	m^2		4.9452E+04	A	p	Pa
sameas	12b b PerimI	m		-177.95	B	Ph(p)	deg
0.5000	c Length	m		3.9763E-02	C	U	m^3/s
1.0260E-02	d AreaF	m^2	Mstr	-88.086	D	Ph(U)	deg
0.35906	e PerimF	m	13d	-19.298	E	Htot	W
stainless	Solid type			2.4213	F	Edot	W
!----- 14 -----							
DUCT							
sameas	13d a Area	m^2	Mstr	5.1605E+04	A	p	Pa
0.35907	b Perim	m	14a	-177.96	B	Ph(p)	deg
0.2000	c Length	m		4.1708E-17	C	U	m^3/s
5.0000E-04	d Srough			-93.428	D	Ph(U)	deg
				-19.298	E	Htot	W
ideal	Solid type			1.0263E-13	F	Edot	W
!----- 15 -----							
HARDEND							
0.0000	a R(1/z)		=15G	5.1605E+04	A	p	Pa
0.0000	b I(1/z)		=15H	-177.96	B	Ph(p)	deg
0.0000	c Htot	W		4.1708E-17	C	U	m^3/s
				-93.428	D	Ph(U)	deg
				-19.298	E	Htot	W
				1.0263E-13	F	Edot	W
				2.8602E-17	G	R(1/z)	
				2.9856E-16	H	I(1/z)	
!----- 16 -----							
RPN Temperature Difference in Mid-HX							


```

0.0000 a G or T          =16A          0.0000          A
7H 5H -
!-----
! The restart information below was generated by a previous run
! and will be used by DeltaEC the next time it opens this file.
guessz  0b  0c  0d  2c  5e  7e
targs   3f  5f  9f  15a 15b 16a
mstr-slave 7 2 -2 4 -1 6 -2 8 -1 12 -2 13 -3 14 -2

```

2. Designed Thermoacoustic Engine – Refrigerator System: Unit 1

```

!Created with DeltaEC version 6.2b3 under win32, using Win 5.1.2600
(Service Pack 3) under Python DeltaEC.
!----- 0 -----
BEGIN 0
6.5000E+06 a Mean P Pa
51.109 b Freq Hz G
1147.1 c TBeg K G
4.0988E+05 d |p| Pa G
0.0000 e Ph(p) deg
0.0000 f |U| m^3/s
90.000 g Ph(U) deg
0.7100 j nL
HeXe Gas type
!----- 1 -----
HARDEND
0.0000 a R(1/z) 4.0988E+05 A |p| Pa
0.0000 b I(1/z) 0.0000 B Ph(p) deg
0.0000 c Htot W 0.0000 C |U| m^3/s
0.0000 D Ph(U) deg
0.0000 E Htot W
0.0000 F Edot W
0.0000 G R(1/z)
0.0000 H I(1/z)
!----- 2 -----
DUCT
1.6000E-02 a Area m^2 Mstr 4.0979E+05 A |p| Pa
0.44835 b Perim m 2a 5.3641E-05 B Ph(p) deg
4.0305E-02 c Length m G 7.8544E-03 C |U| m^3/s
-90.144 D Ph(U) deg
0.0000 E Htot W
stainless Solid type -4.0372 F Edot W
!----- 3 -----
HX
sameas 2a a Area m^2 4.0967E+05 A |p| Pa
0.4000 b GasA/A 4.7976E-04 B Ph(p) deg
1.0000E-02 c Length m 8.6551E-03 C |U| m^3/s
3.0000E-03 d y0 m -90.285 D Ph(U) deg
800.00 e HeatIn W 800.00 E Htot W
1223.0 f SolidT K =3H -8.8313 F Edot W
1147.1 G GasT K
ideal Solid type 1223.0 H SolidT K
!----- 4 -----

```

STKRECT Engine Stack

sameas	2a	a	Area	m^2		4.0698E+05	A	p	Pa
	0.82645	b	GasA/A		4def	0.10995	B	Ph(p)	deg
	0.1000	c	Length	m		2.4063E-02	C	U	m^3/s
	2.5000E-04	d	aa	m	Mstr	-87.329	D	Ph(U)	deg
	2.5000E-05	e	Lplate	m	Mstr	800.00	E	Htot	W
	2.5000E-04	f	bb	m	Mstr	218.77	F	Edot	W
						1147.1	G	TBeg	K
celcor				Solid type		450.47	H	TEnd	K
!----- 5 -----									
HX Ambient HX - Hot Side									
sameas	2a	a	Area	m^2		4.0672E+05	A	p	Pa
	0.7100	b	GasA/A			0.11059	B	Ph(p)	deg
	5.0000E-03	c	Length	m		2.4801E-02	C	U	m^3/s
	5.0000E-04	d	y0	m		-87.533	D	Ph(U)	deg
	-596.24	e	HeatIn	W	G	203.76	E	Htot	W
	443.00	f	SolidT	K	=5H	207.32	F	Edot	W
						450.47	G	GasT	K
ideal				Solid type		443.00	H	SolidT	K
!----- 6 -----									
DUCT									
sameas	2a	a	Area	m^2	Mstr	4.0668E+05	A	p	Pa
	0.4484	b	Perim	m	6a	0.11039	B	Ph(p)	deg
	1.0000E-03	c	Length	m		2.4994E-02	C	U	m^3/s
	5.0000E-04	d	Srough			-87.552	D	Ph(U)	deg
						203.76	E	Htot	W
stainless				Solid type		207.28	F	Edot	W
!----- 7 -----									
CONE									
sameas	2a	a	AreaI	m^2		4.0339E+05	A	p	Pa
sameas	2b	b	PerimI	m		9.6450E-02	B	Ph(p)	deg
	8.0000E-02	c	Length	m		4.2718E-02	C	U	m^3/s
sameas	8a	d	AreaF	m^2		-88.55	D	Ph(U)	deg
sameas	8b	e	PerimF	m		203.76	E	Htot	W
stainless				Solid type		203.48	F	Edot	W
!----- 8 -----									
DUCT									
	2.1000E-02	a	Area	m^2	Mstr	4.0241E+05	A	p	Pa
	0.51383	b	Perim	m	8a	9.3434E-02	B	Ph(p)	deg
	2.0000E-02	c	Length	m		4.7737E-02	C	U	m^3/s
						-88.699	D	Ph(U)	deg
						203.76	E	Htot	W
stainless				Solid type		202.47	F	Edot	W
!----- 9 -----									
HX Ambient HX - Cold Side									
sameas	8a	a	Area	m^2		4.0203E+05	A	p	Pa
	0.7100	b	GasA/A			9.3909E-02	B	Ph(p)	deg
	5.0000E-03	c	Length	m		4.8661E-02	C	U	m^3/s
	1.0000E-03	d	y0	m		-88.764	D	Ph(U)	deg
	-391.28	e	HeatIn	W	G	-187.53	E	Htot	W
sameas	5f	f	SolidT	K		194.98	F	Edot	W
						450.47	G	GasT	K
ideal				Solid type		443.00	H	SolidT	K
!----- 10 -----									

```

STKSLAB Cooler Stack
sameas 8a a Area m^2 3.7848E+05 A |p| Pa
0.80592 b GasA/A 10de 0.18961 B Ph(p) deg
0.2000 c Length m 8.8751E-02 C |U| m^3/s
7.0000E-04 d y0 m Mstr -89.684 D Ph(U) deg
2.8100E-05 e Lplate m Mstr -187.53 E Htot W
36.916 F Edot W
450.47 G TBeg K
319.59 H TEnd K
celcor Solid type
!----- 11 -----
HX Cold HX
sameas 8a a Area m^2 3.7769E+05 A |p| Pa
0.7100 b GasA/A 0.19193 B Ph(p) deg
4.0000E-03 c Length m 8.9439E-02 C |U| m^3/s
1.0000E-03 d y0 m -89.698 D Ph(U) deg
150.00 e HeatIn W -37.529 E Htot W
323.00 f SolidT K =11H 32.366 F Edot W
319.59 G GasT K
323.00 H SolidT K
copper Solid type
!----- 12 -----
DUCT
sameas 8a a Area m^2 3.7756E+05 A |p| Pa
sameas 8b b Perim m 0.1919 B Ph(p) deg
1.0000E-03 c Length m 8.9675E-02 C |U| m^3/s
-89.699 D Ph(U) deg
-37.529 E Htot W
32.330 F Edot W
stainless Solid type
!----- 13 -----
CONE
sameas 8a a AreaI m^2 3.6099E+05 A |p| Pa
sameas 8b b PerimI m 0.18949 B Ph(p) deg
5.0000E-02 c Length m 9.5984E-02 C |U| m^3/s
sameas 14a d AreaF m^2 -89.71 D Ph(U) deg
sameas 14b e PerimF m -37.529 E Htot W
stainless Solid type 30.527 F Edot W
!----- 14 -----
DUCT
4.0000E-03 a Area m^2 Mstr 5.2197E+04 A |p| Pa
0.22421 b Perim m 14a -179.66 B Ph(p) deg
0.5000 c Length m 0.10534 C |U| m^3/s
-89.73 D Ph(U) deg
-37.529 E Htot W
3.4591 F Edot W
stainless Solid type
!----- 15 -----
CONE
sameas 14a a AreaI m^2 8.6344E+04 A |p| Pa
sameas 14b b PerimI m -179.7 B Ph(p) deg
0.1000 c Length m 0.10238 C |U| m^3/s
2.4328E-02 d AreaF m^2 -89.729 D Ph(U) deg
0.5529 e PerimF m -37.529 E Htot W
stainless Solid type 1.8666 F Edot W
!----- 16 -----
COMPLIANCE end bulb sphere
0.56561 a SurfAr m^2 16b 8.6344E+04 A |p| Pa

```

```

4.0000E-02 b Volume m^3      Mstr      -179.7      B Ph(p)  deg
                                7.3133E-13 C |U|      m^3/s
                                89.222      D Ph(U)  deg
                                -37.529      E Htot   W
stainless                      Solid type -5.9123E-10 F Edot   W
!----- 17 -----
HARDEND
    0.0000 a R(1/z)           =17G      8.6344E+04 A |p|      Pa
    0.0000 b I(1/z)           =17H      -179.7      B Ph(p)  deg
    0.0000 c Htot            W          7.3133E-13 C |U|      m^3/s
                                89.222      D Ph(U)  deg
                                -37.529      E Htot   W
                                -5.9123E-10 F Edot   W
                                -2.1468E-13 G R(1/z)
                                -1.1463E-11 H I(1/z)
!----- 18 -----
RPN Temperature Difference in Mid-Ex
    0.0000 a G or T           =18A      0.0000      A
9H 5H -
!-----
! The restart information below was generated by a previous run
! and will be used by DeltaEC the next time it opens this file.
guessz  0b  0c  0d  2c  5e  9e
targs    3f  5f  11f  17a  17b  18a
mstr-slave 7 2 -2 4 -1 6 -2 8 -2 10 -1 14 -2 16 -5

```

3. Thermoacoustic Engine – Refrigerator System: Unit 2

```

!Created with DeltaEC version 6.2b3 under win32, using Win 5.1.2600
(Service Pack 3) under Python DeltaEC.
!----- 0 -----
BEGIN
    6.0000E+06 a Mean P Pa
    54.432    b Freq  Hz      G
    1202.5    c TBeg  K        G
    2.9022E+05 d |p|    Pa      G
    0.0000    e Ph(p)  deg
    0.0000    f |U|    m^3/s
    90.000    g Ph(U)  deg
    0.6700    j nL
HeXe                      Gas type
!----- 1 -----
HARDEND
    0.0000 a R(1/z)           2.9022E+05 A |p|      Pa
    0.0000 b I(1/z)           0.0000 B Ph(p)  deg
    0.0000 c Htot            W          0.0000 C |U|      m^3/s
                                0.0000 D Ph(U)  deg
                                0.0000 E Htot   W
                                0.0000 F Edot   W
                                0.0000 G R(1/z)
                                0.0000 H I(1/z)
!----- 2 -----
DUCT

```

1.7000E-02	a	Area	m^2	Mstr	2.9001E+05	A	p	Pa
0.46214	b	Perim	m	2a	1.7743E-04	B	Ph(p)	deg
6.7630E-02	c	Length	m	G	1.1437E-02	C	U	m^3/s
					-90.139	D	Ph(U)	deg
					0.0000	E	Htot	W
					-4.0281	F	Edot	W
stainless			Solid type					
!-----					3	-----		
HX Hot HX								
sameas	2a	a	Area	m^2	2.8983E+05	A	p	Pa
0.4000	b	GasA/A			2.7085E-03	B	Ph(p)	deg
1.0000E-02	c	Length	m		1.2172E-02	C	U	m^3/s
1.0000E-03	d	y0	m		-90.415	D	Ph(U)	deg
800.00	e	HeatIn	W		800.00	E	Htot	W
1223.0	f	SolidT	K	=3H	-12.855	F	Edot	W
					1202.5	G	GasT	K
copper			Solid type		1223.0	H	SolidT	K
!-----					4	-----		
STKRECT Engine Stack								
sameas	2a	a	Area	m^2	2.8725E+05	A	p	Pa
0.82645	b	GasA/A		4def	0.18694	B	Ph(p)	deg
0.1000	c	Length	m		2.4817E-02	C	U	m^3/s
2.5000E-04	d	aa	m	Mstr	-86.996	D	Ph(U)	deg
2.5000E-05	e	Lplate	m	Mstr	800.00	E	Htot	W
2.5000E-04	f	bb	m	Mstr	175.20	F	Edot	W
					1202.5	G	TBeg	K
celcor			Solid type		585.97	H	TEnd	K
!-----					5	-----		
HX Ambient HX - Hot Side								
sameas	2a	a	Area	m^2	2.8703E+05	A	p	Pa
0.7100	b	GasA/A			0.18895	B	Ph(p)	deg
5.0000E-03	c	Length	m		2.5477E-02	C	U	m^3/s
4.0000E-04	d	y0	m		-87.23	D	Ph(U)	deg
-573.73	e	HeatIn	W	G	226.27	E	Htot	W
580.00	f	SolidT	K	=5H	164.65	F	Edot	W
					585.97	G	GasT	K
copper			Solid type		580.00	H	SolidT	K
!-----					6	-----		
DUCT								
sameas	2a	a	Area	m^2	2.8700E+05	A	p	Pa
0.46221	b	Perim	m	6a	0.18869	B	Ph(p)	deg
1.0000E-03	c	Length	m		2.5644E-02	C	U	m^3/s
5.0000E-04	d	Srough			-87.247	D	Ph(U)	deg
					226.27	E	Htot	W
stainless			Solid type		164.62	F	Edot	W
!-----					7	-----		
CONE								
sameas	2a	a	AreaI	m^2	2.8433E+05	A	p	Pa
sameas	2b	b	PerimI	m	0.17034	B	Ph(p)	deg
8.0000E-02	c	Length	m		4.0475E-02	C	U	m^3/s
sameas	8a	d	AreaF	m^2	-88.217	D	Ph(U)	deg
sameas	8b	e	PerimF	m	226.27	E	Htot	W
stainless			Solid type		161.95	F	Edot	W
!-----					8	-----		
DUCT								

2.1000E-02	a	Area	m^2	Mstr	2.8313E+05	A	p	Pa
0.51383	b	Perim	m	8a	0.16417	B	Ph(p)	deg
3.0000E-02	c	Length	m		4.6594E-02	C	U	m^3/s
					-88.438	D	Ph(U)	deg
					226.27	E	Htot	W
					160.91	F	Edot	W
stainless			Solid type					
!-----					9	-----		
HX Ambient HX - Cold Side								
sameas	8a	a	Area	m^2	2.8280E+05	A	p	Pa
0.7100	b	GasA/A			0.16595	B	Ph(p)	deg
5.0000E-03	c	Length	m		4.7364E-02	C	U	m^3/s
7.0000E-04	d	y0	m		-88.521	D	Ph(U)	deg
-404.98	e	HeatIn	W	G	-178.71	E	Htot	W
sameas	5f	f	SolidT	K	153.46	F	Edot	W
					585.97	G	GasT	K
copper			Solid type		580.00	H	SolidT	K
!-----					10	-----		
STKSLAB Cooler Stack								
sameas	8a	a	Area	m^2	2.6482E+05	A	p	Pa
0.80592	b	GasA/A		10de	0.29563	B	Ph(p)	deg
0.2000	c	Length	m		7.9999E-02	C	U	m^3/s
7.0000E-04	d	y0	m	Mstr	-89.536	D	Ph(U)	deg
2.8100E-05	e	Lplate	m	Mstr	-178.71	E	Htot	W
					31.160	F	Edot	W
					585.97	G	TBeg	K
celcor			Solid type		440.72	H	TEnd	K
!-----					11	-----		
HX Cold HX								
sameas	8a	a	Area	m^2	2.6424E+05	A	p	Pa
0.7100	b	GasA/A			0.3011	B	Ph(p)	deg
4.0000E-03	c	Length	m		8.0572E-02	C	U	m^3/s
6.0000E-04	d	y0	m		-89.561	D	Ph(U)	deg
150.00	e	HeatIn	W		-28.708	E	Htot	W
443.00	f	SolidT	K	=11H	25.576	F	Edot	W
					440.72	G	GasT	K
copper			Solid type		443.00	H	SolidT	K
!-----					12	-----		
DUCT								
sameas	8a	a	Area	m^2	2.6414E+05	A	p	Pa
sameas	8b	b	Perim	m	0.30106	B	Ph(p)	deg
1.0000E-03	c	Length	m		8.0762E-02	C	U	m^3/s
					-89.562	D	Ph(U)	deg
					-28.708	E	Htot	W
stainless			Solid type		25.550	F	Edot	W
!-----					13	-----		
CONE								
sameas	8a	a	AreaI	m^2	2.5224E+05	A	p	Pa
sameas	8b	b	PerimI	m	0.29792	B	Ph(p)	deg
5.0000E-02	c	Length	m		8.5855E-02	C	U	m^3/s
sameas	14a	d	AreaF	m^2	-89.574	D	Ph(U)	deg
sameas	14b	e	PerimF	m	-28.708	E	Htot	W
stainless			Solid type		24.200	F	Edot	W
!-----					14	-----		
DUCT								

```

4.0000E-03 a Area m^2 Mstr 4.1975E+04 A |p| Pa
0.22421 b Perim m 14a -179.51 B Ph(p) deg
0.5000 c Length m 9.3197E-02 C |U| m^3/s
-89.597 D Ph(U) deg
-28.708 E Htot W
2.8640 F Edot W
stainless Solid type
!----- 15 -----
CONE
sameas 14a a AreaI m^2 6.6195E+04 A |p| Pa
sameas 14b b PerimI m -179.57 B Ph(p) deg
0.1000 c Length m 9.0557E-02 C |U| m^3/s
2.4328E-02 d AreaF m^2 -89.596 D Ph(U) deg
0.5529 e PerimF m -28.708 E Htot W
stainless Solid type 1.5987 F Edot W
!----- 16 -----
COMPLIANCE
0.56561 a SurfAr m^2 16b 6.6195E+04 A |p| Pa
4.0000E-02 b Volume m^3 Mstr -179.57 B Ph(p) deg
1.3073E-12 C |U| m^3/s
-86.714 D Ph(U) deg
-28.708 E Htot W
stainless Solid type -2.1529E-09 F Edot W
!----- 17 -----
HARDEND
0.0000 a R(1/z) =17G 6.6195E+04 A |p| Pa
0.0000 b I(1/z) =17H -179.57 B Ph(p) deg
0.0000 c Htot W 1.3073E-12 C |U| m^3/s
-86.714 D Ph(U) deg
-28.708 E Htot W
-2.1529E-09 F Edot W
-1.1087E-12 G R(1/z)
2.2254E-11 H I(1/z)
!----- 18 -----
RPN Temperature Difference in Mid-Ex
0.0000 a G or T =18A 0.0000 A
9H 5H -
!-----
! The restart information below was generated by a previous run
! and will be used by DeltaEC the next time it opens this file.
guessz 0b 0c 0d 2c 5e 9e
targs 3f 5f 11f 17a 17b 18a
mstr-slave 7 2 -2 4 -1 6 -2 8 -2 10 -1 14 -2 16 -5

```

4. Thermoacoustic Engine – Refrigerator System: Unit 3

!Created with DeltaEC version 6.2b3 under win32,using Win 5.1.2600
(Service Pack 3) under Python DeltaEC.

```

!----- 0 -----
BEGIN
6.0000E+06 a Mean P Pa
65.579 b Freq Hz G
1195.2 c TBeg K G
2.4207E+05 d |p| Pa G
0.0000 e Ph(p) deg

```

```

0.0000 f |U|      m^3/s
90.000  g Ph(U)   deg
0.7200  j nL
HeXe      Gas type
!----- 1 -----
HARDEND
0.0000 a R(1/z)      2.4207E+05 A |p|      Pa
0.0000 b I(1/z)      0.0000 B Ph(p)   deg
0.0000 c Htot      W
0.0000      0.0000 C |U|      m^3/s
0.0000      0.0000 D Ph(U)   deg
0.0000      0.0000 E Htot      W
0.0000      0.0000 F Edot      W
0.0000      0.0000 G R(1/z)
0.0000      0.0000 H I(1/z)
!----- 2 -----
DUCT Hot Duct 6 (*changeable Hot duct*)      4
1.6000E-02 a Area      m^2      Mstr      2.4132E+05 A |p|      Pa
0.44834 b Perim      m      2a      7.4477E-04 B Ph(p)   deg
0.12346 c Length      m      G      1.9730E-02 C |U|      m^3/s
-90.138      D Ph(U)   deg
0.0000      E Htot      W
stainless      Solid type      -5.7829 F Edot      W
!----- 3 -----
HX Hot HX
sameas      2a a Area      m^2      2.4099E+05 A |p|      Pa
0.4000      b GasA/A      6.2322E-03 B Ph(p)   deg
1.0000E-02 c Length      m      2.0422E-02 C |U|      m^3/s
1.0000E-03 d y0      m      -90.288      D Ph(U)   deg
1200.0      e HeatIn      W      1200.0      E Htot      W
1223.0      f SolidT      K      =3H      -12.654      F Edot      W
1195.2      G GasT      K
copper      Solid type      1223.0      H SolidT      K
!----- 4 -----
STKRECT Engine Stack
sameas      2a a Area      m^2      2.3741E+05 A |p|      Pa
0.75614 b GasA/A      4def      0.30552 B Ph(p)   deg
0.1000      c Length      m      3.1003E-02 C |U|      m^3/s
3.0000E-04 d aa      m      Mstr      -87.026      D Ph(U)   deg
4.5000E-05 e Lplate      m      Mstr      1200.0      E Htot      W
3.0000E-04 f bb      m      Mstr      171.33      F Edot      W
1195.2      G TBeg      K
celcor      Solid type      752.42      H TEnd      K
!----- 5 -----
HX Ambient HX - Hot Side
sameas      2a a Area      m^2      2.3717E+05 A |p|      Pa
0.7100      b GasA/A      0.30809 B Ph(p)   deg
5.0000E-03 c Length      m      3.1620E-02 C |U|      m^3/s
5.0000E-04 d y0      m      -87.196      D Ph(U)   deg
-771.16      e HeatIn      W      G      428.84      E Htot      W
743.00      f SolidT      K      =5H      163.32      F Edot      W
752.42      G GasT      K
copper      Solid type      743.00      H SolidT      K
!----- 6 -----
DUCT

```



```

sameas 2a a Area m^2 Mstr 2.3714E+05 A |p| Pa
0.44841 b Perim m 6a 0.30777 B Ph(p) deg
1.0000E-03 c Length m 3.1776E-02 C |U| m^3/s
5.0000E-04 d Srough -87.208 D Ph(U) deg
428.84 E Htot W
163.29 F Edot W
stainless Solid type
!----- 7 -----
CONE
sameas 2a a AreaI m^2 2.3452E+05 A |p| Pa
sameas 2b b PerimI m 0.28562 B Ph(p) deg
8.0000E-02 c Length m 4.6123E-02 C |U| m^3/s
sameas 8a d AreaF m^2 -88.012 D Ph(U) deg
sameas 8b e PerimF m 428.84 E Htot W
stainless Solid type 160.67 F Edot W
!----- 8 -----
DUCT
2.1000E-02 a Area m^2 Mstr 2.3342E+05 A |p| Pa
0.51383 b Perim m 8a 0.27839 B Ph(p) deg
3.0000E-02 c Length m 5.2203E-02 C |U| m^3/s
-88.22 D Ph(U) deg
428.84 E Htot W
159.63 F Edot W
stainless Solid type
!----- 9 -----
HX Ambient HX - Cold Side
sameas 8a a Area m^2 2.3324E+05 A |p| Pa
0.7100 b GasA/A 0.28124 B Ph(p) deg
3.0000E-03 c Length m 5.2681E-02 C |U| m^3/s
5.0000E-04 d y0 m -88.288 D Ph(U) deg
-607.29 e HeatIn W G -178.44 E Htot W
sameas 5f f SolidT K 153.36 F Edot W
752.42 G GasT K
743.00 H SolidT K
copper Solid type
!----- 10 -----
STKSLAB Cooler Stack
sameas 8a a Area m^2 2.1739E+05 A |p| Pa
0.7950 b GasA/A 10de 0.45798 B Ph(p) deg
0.2000 c Length m 8.4735E-02 C |U| m^3/s
7.0000E-04 d y0 m Mstr -89.339 D Ph(U) deg
3.8100E-05 e Lplate m Mstr -178.44 E Htot W
32.713 F Edot W
752.42 G TBeg K
577.76 H TEnd K
celcor Solid type
!----- 11 -----
HX Cold HX
sameas 8a a Area m^2 2.1702E+05 A |p| Pa
0.7100 b GasA/A 0.46458 B Ph(p) deg
3.0000E-03 c Length m 8.5173E-02 C |U| m^3/s
5.0000E-04 d y0 m -89.365 D Ph(U) deg
150.00 e HeatIn W -28.445 E Htot W
580.00 f SolidT K =11H 27.562 F Edot W
577.76 G GasT K
580.00 H SolidT K
copper Solid type
!----- 12 -----
DUCT

```

```

sameas      8a a Area      m^2                2.1694E+05 A |p|      Pa
sameas      8b b Perim     m                   0.46453 B Ph(p)    deg
1.0000E-03 c Length      m                   8.5361E-02 C |U|      m^3/s
                                           -89.365 D Ph(U)    deg
                                           -28.445 E Htot     W
                                           27.536 F Edot      W
stainless                               Solid type
!----- 13 -----
CONE
sameas      8a a AreaI     m^2                2.0700E+05 A |p|      Pa
sameas      8b b PerimI    m                   0.46053 B Ph(p)    deg
5.0000E-02 c Length      m                   9.0400E-02 C |U|      m^3/s
sameas     14a d AreaF     m^2                -89.379 D Ph(U)    deg
sameas     14b e PerimF    m                  -28.445 E Htot     W
stainless                               Solid type
!----- 14 -----
DUCT
4.0000E-03 a Area      m^2      Mstr          3.7416E+04 A |p|      Pa
0.22421 b Perim      m      14a             -179.31 B Ph(p)    deg
0.5000 c Length      m                   9.7532E-02 C |U|      m^3/s
                                           -89.406 D Ph(U)    deg
                                           -28.445 E Htot     W
                                           3.1539 F Edot      W
stainless                               Solid type
!----- 15 -----
CONE
sameas     14a a AreaI     m^2                5.7492E+04 A |p|      Pa
sameas     14b b PerimI    m                  -179.37 B Ph(p)    deg
0.1000 c Length      m                   9.4757E-02 C |U|      m^3/s
2.4328E-02 d AreaF     m^2                -89.405 D Ph(U)    deg
0.5529 e PerimF    m                  -28.445 E Htot     W
stainless                               Solid type
!----- 16 -----
COMPLIANCE
0.56561 a SurfAr      m^2      16b          5.7492E+04 A |p|      Pa
4.0000E-02 b Volume    m^3      Mstr          -179.37 B Ph(p)    deg
                                           4.3022E-16 C |U|      m^3/s
                                           89.740 D Ph(U)    deg
                                           -28.445 E Htot     W
                                           -1.9260E-13 F Edot      W
stainless                               Solid type
!----- 17 -----
HARDEND
0.0000 a R(1/z)      =17G          5.7492E+04 A |p|      Pa
0.0000 b I(1/z)      =17H          -179.37 B Ph(p)    deg
0.0000 c Htot      W          4.3022E-16 C |U|      m^3/s
                                           89.740 D Ph(U)    deg
                                           -28.445 E Htot     W
                                           -1.9260E-13 F Edot      W
                                           -1.0660E-16 G R(1/z)
                                           -6.8440E-15 H I(1/z)
!----- 18 -----
RPN Temperature Difference in Mid-HX
0.0000 a G or T      =18A          -1.1369E-13 A
9H 5H -
!-----
! The restart information below was generated by a previous run

```

```
! and will be used by DeltaEC the next time it opens this file.  
guessz  0b  0c  0d  2c  5e  9e  
targs   3f  5f  11f 17a 17b 18a  
mstr-slave 7 2 -2 4 -1 6 -2 8 -2 10 -1 14 -2 16 -5
```

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