INNOVATIVE TECHNOLOGICAL PATHWAY FOR NEW COMMERCIAL APPLICATIONS OF STIRLING CYCLE- BASED SYSTEMS

Doctoral Thesis

by

DMITRY SMIRNOV

DOCTORAL PROGRAM IN ENGINEERING SYSTEMS

Supervisor

Dr. Henni Ouerdane
Assistant Professor, Center for Energy Science and Technology

Moscow

ABSTRACT

The importance of developing competitive commercial energy systems for the economy cannot be overstated. Existing research in design literature indicates that the question of how to carry out technical development of commercial science-intensive energy conversion technologies is covered inadequately. To cover this gap, the present thesis proposes a novel design methodology, which was validated by a real commercial development process of an energy conversion system based on the thermodynamic Stirling cycle. The design methodology contains five new design methods, each concerned with how to choose among alternative design configurations at different design stages. The first method offers an approach to a literature review at the conceptual design stage for Stirling systems using big data technologies. The second method describes an approach to perform trade studies at the system design stage for commercial Stirling machine configurations. The third method applies game theory to improve interactions between designers during the detail design stage and reduce negative influence of designers' collaborative decision-making on the design process outcomes. The fourth method shows how to improve the critical performance characteristics of the Stirling machines experimentally during testing stage, based on the example of the piston-cylinder sealing mechanism in a Stirling refrigerator. And the fifth method focuses on the problem of scaling the Stirling machine at the stage of scale-up by developing an experimentally validated numerical model using the example of the Stirling refrigerator. The application of the proposed design methods provides several insightful results. For example, the literature review shows that many existing designs of commercial Stirling machines for on-Earth civil applications are disadvantageously based on the design configuration that was initially developed for space applications. The results of trade studies indicate that alpha-type engines are better suited for high-power applications and beta-type – for high-efficiency applications. Game theory analysis shows that inappropriate designers' decision-making can lead to design outcomes with key perforce metrics 20% worse than the global optimum design. Experimental optimization shows that the choice of the sealing mechanism can significantly limit the commercial applications of the Stirling refrigerator due to its high-stress operation and complex behavior. Finally, the scale modeling, in combination with experimental data for the Stirling refrigerator, shows that its cooling performance demonstrates an exponential dependence from the cooling temperature. This relation is critical for accurate technology scaling. The results of this work show the possibility to apply the proposed design methodology to the development process of other energy conversion technologies. The work formulates a simple visual model that integrates key aspects of developed design methods and is named the Pentagon Model. This model can be used in the commercial development process of other science-intensive energy conversion systems, apart from Stirling machines.

List of publications

Smirnov, D., Dvortsov, V., Saichenko, A., Tkachenko, M., Kukolev, M., Bischi, A., and Ouerdane, H., 2019. Experimental study of a high-tolerance piston-cylinder pair in the alpha Ross-yoke Stirling refrigerator. *International Journal of Refrigeration*, *100*, pp.235-245.

> **Author's contribution:** design, assembling and testing the experimental Stirling refrigerator; development of the test methodology; carrying out experimental investigation; writing the paper.

Smirnov, D. and Golkar, A., 2019. Design Optimization Using Game Theory. *IEEE Transactions on Systems, Man, and Cybernetics: Systems*. Early access.

> **Author's contribution:** development of the mathematical framework using game theory; carrying out numerical calculations for the case study of the Stirling system; writing the paper.

Smirnov, D. and Golkar, A., 2015. Stirling engine systems tradespace exploration framework. *Procedia Computer Science*, *44*, pp.558-567.

> **Author's contribution:** development of the mathematical framework using the concept of trade studies for Stirling engines; carrying out numerical calculations for the case study of the Stirling system; writing the paper.

ACKNOWLEDGMENTS

This work would be impossible without strong and systematic support, trust, and instructive guidance of my research supervisor Dr. Henni Ouerdane. I am honored and thankful that I had the privilege to work under his supervision.

My co-advisor Dr. Kelvin Willoughby inspired by his example to think critically and independently. He helped with his valuable advice in the area of intellectual property and technology-based entrepreneurship.

My co-advisor Dr. Aldo Bischi helped me to be realistic and supported with technical analysis of the experimental system.

I thank Dr. Alessandro Golkar for support and fruitful collaboration in the area of systems engineering and game theory.

The experimental work in this thesis would be impossible without instrumental involvement of several individuals that I would like to thank here: Dr. Aleksandr Saichenko, Vladimir Dvorcov, Dr.Sc. Maxim Kukolev and Dr. Mikhail Tkachenko from Thermal Motors LLC; Dr. Keith Stevenson, Polina Rumyantseva, and Albina Ostanina from Skoltech.

I want to thank my parents and family for their strong support throughout this journey. I am profoundly grateful to my better half Anna for spending weekends with me at the lab collecting the data and being an endless joy and a supportive friend through the years of my work on this dissertation.

Thank you.

TABLE OF CONTENTS

LIST OF FIGURES

CHAPTER I. INTRODUCTION

CHAPTER II. BIBLIOMETRIC STUDY OF STIRLING ENGINES

Chapter III. Game theory in design optimization

Chapter IV. Bibliometric study of Stirling refrigerators

Chapter V. Piston sealing in Stirling refrigerators

Chapter VI. Modeling heating load

Chapter VII. Concluding discussion

LIST OF TABLES

CHAPTER I. INTRODUCTION

CHAPTER III. GAME THEORY IN DESIGN OPTIMIZATION

CHAPTER V. PISTON SEALING IN STIRLING REFRIGERATORS

CHAPTER VI. MODELING HEATING LOAD

CHAPTER VII. CONCLUDING DISCUSSION

CHAPTER I
INTRODUCTION

This work aims to formulate a standard approach that enables to select any science-intensive energy conversion technology (ECT) and develop it into a commercial product. The term "technological pathway" in the title of this dissertation loosely means "design methodology." The latter seems to be static, relating to initial design stages, whereas the present work describes a design technique that addresses several design stages in one comprehensive manner. These stages follow consequently from the conceptual design stage, through detail design, to manufacturing and scaling. For the multi-stage consideration, the term "technological pathway" is more appropriate. One can think about a multi-stage design process as an "assembly line" of different design methods applied sequentially to technologically treat the object of design as it moves from the conceptual stage to the product stage. The performance of this "assembly line" directly affects the quality and cost of design process outputs. The innovative technological pathway includes a collection of novel design methods focused on commercial development. It is essential to select a particular type of science-intensive ECT and use it as an example to formulate key principles of the technological pathway. This work uses the energy conversion system based on the thermodynamic Stirling cycle as an example of ECT and offers a collection of novel design methods that address each stage of the design process from concept to scale-up and are adapted for new commercial applications of Stirling machines. However, the general conclusions from this work have important implications for the development of any science-intensive ECT.

Since this work offers a technological pathway that simplifies the commercialization of science-intensive energy conversion systems, I ground this study in the framework of the "technology push" concept [1]. Scientists typically carry out the commercial development of science-intensive energy conversion technologies, such as gas turbines, nuclear power plants, and fusion power systems. There is a differentiation between scientific and commercial development activities in R&D intensive organizations [2]. However, the academic literature often overlooks or assumes implicitly that the same people – scientists – would carry out the commercial development because of the high scientific intensity of energy conversion systems. Scientific approach can be characterized as curious, rigorous, reflective, systematically establishing relations for natural phenomena. The problem arises when scientific perspective engages with the commercial development that is market-oriented, limited in time and resources [3], and characterized by the bounded rationality of decision-making [4], which stems from finite nature of commercial development. This is a paradox of developing science-intensive products: it needs scientists, but its driving factors inherently conflict with scientific thinking. The result of such a contradiction could be a low success rate of development, poor product functionality, high product cost, and low reliability. Therefore, there is a need to develop a set of design technics that take into account scientific uncertainty and forces of commercial development, systematize and standardize the development process, and allow to churn out (despite some negative connotation to this idiom) commercial systems based on energy conversion technologies. This work attempts to build an "assembly line" of novel design methods to churn out commercial systems, using the example of the machines based on the thermodynamic Stirling cycle. My aspiration for the future is to apply the same technological pathway to develop other energy conversion technologies into commercial products.

As a general note, in the course of the chapters, when referring to designers, I use the masculine pronouns generally and sometimes the feminine form when appropriate. Each chapter has its own list of references. For more than two consequent references, for example [1], [2], [3], I omit the references in the middle and write [1]-[3] for simplicity.

I.1 Motivation

An attentive reader of the dissertation title may immediately ponder over four questions. Why was it essential for the author to focus on energy conversion technologies and in particular on Stirling cycle systems? Why was the author interested in commercial applications rather than fundamental ones? Why did the author felt the need for an innovative technological pathway for new commercial applications? And finally, was it so vital to write a whole Ph.D. thesis about it? One could use a plethora of academic literature to argue the importance of Stirling cycle systems or the commercial potential of Stirling machines. But to give these four questions justice, I would like first to discuss my intrinsic motivations for this research.

I believe two factors encouraged me to focus on energy conversion technologies and specifically on the Stirling cycle systems. One is a deep family tradition in the development of energy conversion technologies, and two is technical elegance, which I see in the Stirling cycle. Three generations of my family before me worked in the design and construction of energy conversion systems. My great grandfather German Kiselev was awarded the Order of Red Star in 1937 by the head of the Soviet State Mikhail Kalinin for the development of a new aircraft motor. Forty years later in 1977 my grandfather German Gusev developed an advanced jet engine and was awarded the Medal for Labor Valor by the Presidium of USSR. Family exemplary achievements and multi-generation legacy forged my interest in the development of energy conversion technologies. It crystallized first through family stories and library, and later – through family professional activities and knowledge in the practical development of energy systems. While being enrolled in a BSc program in power machine building at St.Petersburg Polytechnic University, I learned about the Stirling engine in the beginning of 2010. I was interested because the motor could be applied to harvest waste heat of gas turbines and generate useful energy, reducing the energy cost. Upon closer look, I was amazed by the elegance of the Stirling cycle. A quiet and closed thermodynamic system, capable of operating on any external

– and thus optimized and most preferable – heat source. The cycle has the highest ideal theoretical efficiency equal to the Carnot efficiency [5]. My immediate reaction was to develop a project with my like-minded colleagues to utilize, using the Stirling engine, the waste heat of gas turbines generated by natural gas pumping stations. The project received a winning prize at the conference of Gazprom young specialists [6], which encouraged me to work on this project further. This technology captured my engineering imagination. The idea of building my own Stirling engine incepted, and in 2011, during my free hours, I started devising a plan to build a real Stirling machine. At the end of my bachelor studies, the amplifying interaction between the influence of my family legacy and my subjective preference towards the Stirling machine motivated me to continue studying this ECT scientifically during MSc studies in Skoltech. My research interest inclined towards the commercial development of the Stirling machines, and here is why.

I believe the commercialization of scientific results is the most efficient way to make my research useful. In the first part of 2012, after working as a research assistant in a power machine building lab and before starting my MSc studies, I became discontent how applied research and related know-how in power machine building rarely found its way to markets. And even if it happened, then it is most certainly had to be large Russian companies who could succeed in this assiduous task. At least, this was a believe that they could. The reasons why it had to be industrial companies were clear to me: financial resources, administrative advantage, qualified scientists and engineers, and the understanding of markets. Yet, my experience in research projects for two large machine building companies showed that big companies are extremely selective, rationalizing, and frequently too slow in adapting the results of applied studies. Similar tardiness was apparent at the side of scientific institutions that conducted this research. Prior studies of commercialization practices in Russian universities confirm this observation [7]. As a consequence, many experimentally tested and prospective technical solutions were sunk into the oblivion of dusty unread reports or were lost in vain somewhere in scrap equipment of research labs. I did not want to be a part of this slow and unproductive culture. I felt the urge to prove wrong the prejudice that only big companies can succeed in energy conversion technologies. I believed a small group of committed individuals could find resolve and resources to build a commercially successful energy system fast and efficiently. And these believe reformed the original discontent into a latent (at that time) intention to start a commercial venture in energy conversion technologies. In 2012, I enrolled in the MSc program of Skoltech and embraced its bold idea of commercializing science-intensive research. The same year, I wrote an essay "Why the Skoltech idea matters to me." It has been viewed more than 400 times on my LinkedIn profile since then. During two years of Master studies, I was learning how to find a way for science-intensive engineering systems to market. In my MSc thesis, I focused on studying the Stirling engine to understand better the scientific principles behind it and how to make it competitive. In my free hours, I kept planning and in 2014 indeed started a company Thermal Motors LLC with a long-term objective to develop commercially energy conversion technologies, where the Stirling engine would become the first example. The existing preconceptions and methods of commercializing energy conversion technologies through research institutions and large companies proved in my eye inefficient, and I was convinced in the need to find new technological pathways to market energy conversion technologies.

The preliminary studies during the MSc work showed the scientific complexity and multidisciplinary nature of developing a commercial Stirling machine. Therefore, it was necessary to combine scientific and business activities to develop a successful Stirling system. I decided to engage in Ph.D. work to complete the scientific part of this development. A distinctive qualification of this thesis is that it chronologically replicates the development of the commercial Stirling machine. This process required to address several critical research questions related to the Stirling cycle technologies, which were worthy of a Ph.D. thesis. The commercial development started with the focus on the Stirling engine, and over time, the focus pivoted towards the Stirling refrigerator. In 2018, the project resulted in an operating experimental Stirling refrigerator with possible applications in several commercial segments. A reader may ask how the specific research questions that arose during the development of the Stirling machine are related to the design methodology to commercially develop other energy conversion technologies apart from those based on the Stirling cycle? There are two important comments here. One is that although the results from investigating those questions are specifically related to the Stirling machines, the methods that were used during the process of solving them are general enough and are concerned with "how" and not "what". The second comment, it is the main assumption of this work that other energy-conversion technologies would have similar design problems during the commercial development due to high degree of scientific work embedded in them, the material nature of hardware devices required to take control over energy conversion processes, and common laws of energy conversion and heat transfer.

A reader may further inquire "Why would not we explore the existing development methods employed in industry to find an appropriate technological pathway? Lauff *et al.* [8], p. 1, argued that there is currently a small empirical base about current design practices in companies. This account may seem unfair since the design literature is abundant with novel design methods. However, the focus of the work of Lauff *et al.* was rather on the methods actually employed by industry engineers, and these methods may be oftentimes different from the proposed in the academic literature. During the creation of the operating Stirling machine, several design methods were developed. In most instances, elements of these methods were not known a priori and instead were crystallized by the trial and error tinkering at times of important engineering challenges or were imposed by project constraints. The full formulation of these methods was completed a posteriori under the assumption that similar problems may arise in future and it would be prudent to manage them more formally rather than in the original ad-hoc fashion. Therefore, this dissertation in part offers a solution to the problem coined by Lauff and her colleagues by systematizing in a comprehensive manner a set of design methods implemented during the commercial development of an ECT with a strong scientific component.

Unlike one may expect from a typical doctoral dissertation, my motivations for the selection of the present topic are not extrinsic and do not stem from a methodological selection based on academic literature with careful evaluation of novelty and relevance. Instead, the choice of my topic was motivated by intrinsic factors: personal experience and subjective judgment. I choose ECT over, for example, a computer science because of my family legacy calling. I chose Stirling cycle systems and not, for example, solar cells or wind turbines because I am fascinated by the technical elegance of this conversion technology. I selected the commercialization focus for the technology choice because I strongly believe that the society cannot solve energy problems without bringing perspective energy-conversion technologies to the market, while the development of the commercial system is the most efficient way to do it. And I am convinced that there is a need for a more efficient pathway to develop energy-conversion technologies because I am dissatisfied with the way it is carried out presently through large market players and research institutions. My highest hope is that by writing this thesis not only I contribute with original results into the field, but I would also demonstrate my qualification for organizing development work of market-relevant energy-conversion technologies, building and optimizing experimental systems and scale it for different commercial applications.

I.2 Description of the energy conversion technology case study

Before discussing the methodology of this thesis, it is important to define an ECT that was used as an example or a case study to validate the proposed technological pathway.

I.2.1 Short historic reference of Stirling engines and refrigerators

At the beginning of the 19th century, the industrial zeal for coal resulted in the excavation of deep mines that, as a result, were flooded by underground waters. The steam engine pumps dried the mines, but frequently exploded due to weak iron-based construction materials. The high-pressure steam would burst out and violently attack operational personnel, causing casualties and even deaths. To prevent pains of his parish, the Scottish reverend Robert Stirling designed a pumping engine that would be safer by operating air as a working gas instead of vapor. After the invention of Bessemer steel in 1856, the material strength and power of steam engines grew, outperforming and obliterating the Stirling engine. In 1861, Alexander Krik developed the first Stirling refrigerator by reversing the cycle with the application of the external work. Instead of power on the shaft, the machine produced cooling capacity. In the middle of the 21st century, the principles of the Stirling engine saw the lights of the day once again and were implemented in new thermal machines by a small number of industrial companies. From that time, the devices based on the Stirling cycle have gained the interest of the scientific and industrial community.

Why is it essential for society to conduct research activities related to Stirling engines and refrigerators? In theory, this technology possesses several advantages and engineering challenges that are reported commonly in the literature. Due to the unique process of heat regeneration, the theoretical energy efficiency of Stirling machines is the highest among alternative heat engine systems [9]. However, the real-world mechanical realization of the Stirling cycle in a way that can compete with conventional alternatives is a challenge. Also, the studies that systematically compare the performance of Stirling machines under controlled conditions are scares. Most claims about the high potential of Stirling cycle technology are subjective because they were made without reference to rigorous experimental comparison studies. Some limited references to this analysis include Teruyuki *et al.* [10]. The comparison of the Stirling engine with other alternative energy conversion technologies, including gas turbines, gasoline engine and diesel engine, shows that this system has the highest heat efficiency up to 40% for powers ranging from 1 kW to 20 kW, which is more than 50% compared with the efficiency of 20% for gasoline engine in the same range. Studies for Stirling refrigerators include the comparison with conventional vapor-compression systems. Examples include the work of Hermes and Barbosa [11] for the cooling capacity approximately 20 W at a cooling temperature around 250 K. An important parameter reported in this work is the ratio of cooling capacity to input energy (also known as the coefficient of performance). For the vapor compressor refrigerators based on the Rankine cycle, it was 0.79 for the reciprocal compressor and 0.62 for the linear compressor, while the same parameter for the Stirling refrigerator was 0.98, with at least 24% efficiency increase. These studies show significant efficiency improvements for engines and refrigerators and warrant further research studies in the field of Stirling machines.

I.2.2 STIRLING CYCLE FOR ENGINES

The Stirling cycle for engines is a physical concept that describes a thermodynamic cycle, which produces useful work. Fig. 1 exhibits an ideal Stirling cycle [12] for engines, which is also known as the *direct* Stirling cycle. It consists of two isothermal (1-2, 3-4) and two isochoric (2-3, 4-1) processes. During a single-phase isothermal compression (1-2) the working gas (for example, helium) is compressed and at the same time cooled with rejection of heat Q_c. During the isochoric process (2-3) the gas is pre-heated with the regenerated heat Q_r. During isothermal single-phase expansion of (3-4) the heat Q_h is supplied to the cycle from an external sources and at the same time gas produces work. After this, a part of gas heat

Q_r is conserved in a regenerator (heat-storage) in order to pre-heat gas in the next cycle during process (3-4).

I.2.3 STIRLING CYCLE FOR REFRIGERATORS

The Stirling cycle for refrigerators is a physical concept that describes a thermodynamic cycle, which produces cooling capacity. Fig. 2 exhibits an ideal Stirling cycle [13] for refrigerators, which is also known as the *reversed* Stirling cycle. It consists of two isothermal (1-2, 3-4) and two isochoric (2-3, 4-1) processes. The heat lifting capacity Q_c of the Stirling refrigerator is created during isothermal single-phase expansion (3-4) of refrigerant gas (for example, helium, nitrogen or hydrogen). The rejection of accepted heat Q_h is realized during single-phase isothermal compression (1-2). The process of the refrigerant temperature reduction is conducted during the isochoric regenerative heat rejection Q_r (2-3) and isochoric regenerative cold charge Q_r (4-1). The next section explains how the mechanical system of designed Stirling refrigerator realizes the reversed cycle; however, it is worth making several comments here.

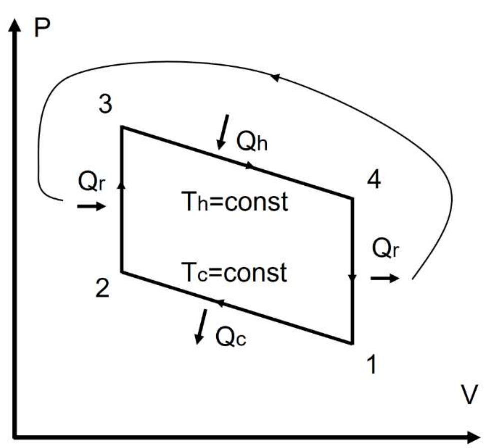


Fig. 1. Ideal Stirling engine cycle.

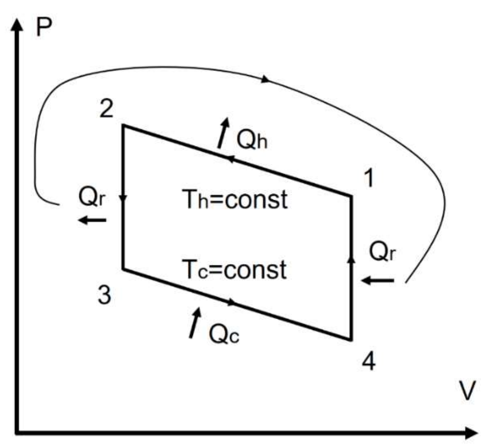


Fig. 2. Ideal Stirling refrigerator cycle.

A similar thermodynamic cycle at the level of first principles can be operated in two different ways, which results in the creation of two separate industrial segments – Stirling engines and Stirling refrigerators. From the engineering point of view, these two segments would most likely require two different designs of Stirling machines. The same design of a Stirling machine would not be optimal at the same time for the engine application and the refrigerator application. This conclusion can be deduced from the operation conditions. The operating temperature for the engine is around 800K and for the refrigerator is around 100K. This significant difference affects the choice of materials for the heat exchangers, the regenerator, and the piston-cylinder sealing mechanism. Besides, the heat transfer for engines and refrigerators has a different regime and thermal resistances. This difference affects the design of heat exchangers. Another aspect is the objective performance metric for the system. For the Stirling engine this could be power and for the Stirling refrigerator – efficiency. The arrangement of gas channels inside the machine and as a result, the choice of the mechanical drive will be different and depend on whether the objective is to maximize power or efficiency. These design aspects demonstrate that the optimal design for the Stirling engine and the Stirling refrigerator will be different. However, the critical questions are how much different and could the same conclusion be made about the necessity of different designs from a commercial point of view? These questions extend the concept of technical optimality that is typically used in scientific and engineering literature and involves a commercial consideration. Perhaps, the benefit of design cost savings from using the same design configuration both for engine and refrigerator applications could balance the loss of corresponding performance metric. The systematic study of this problem is not known to the author. This work represents an example of the design process that kept the option of choosing the final application (engine or refrigerator) open as long as it was possible to account for real-world factors in developing an optimal design of a Stirling machine. This approach relates to the notion of set-based thinking [14]. The next paragraph discusses a design configuration that was developed for the engine application but later was reconfigured for the refrigerator use.

I.2.4 Stirling refrigeration machine

This technical specification section describes in detail the structure, technical parameters, and operational characteristics of the studied Stirling machine with its test rig.

a) The structure of the Stirling refrigerator

Figure 3 presents a schematic diagram of the Stirling refrigerator elements. The kinematic mechanism 1 is driven through the shaft by the electrical motor (not shown). The mechanism converts rotational movement of the shaft into reciprocal movement of the compression piston 2 that increases helium pressure in the compression space 3 (process 1-2 in Fig. 2). Compressed helium concurrently moves into the refrigerator radiator (heat sink) 4 where it rejects heat to the externally supplied cooling water. Thus, the compression process can be considered isothermal. After the isothermal compression, helium enters the regenerator 5 and rejects heat to the material of the regenerator, which works as fast charge-discharge heat storage (process 2-3 in Fig. 2). Precooled compressed helium enters the expansion volume 7 through cold head pipes 6 and expands (process 3-4 in Fig. 2). The expansion causes further temperature reduction of helium. In parallel, the heat exchanger pipes and the external surface of the expansion cylinder accept heat from the external environment, thus providing the cooling capability. After the displacer 8 reaches the lowest expansion point, it is returned back by the expansion piston 9. The piston is driven by the kinematic mechanism 1. The expanded cold helium enters the regenerator and accepts previously rejected heat while cooling the regenerator to even lower temperature (process 4-1 in Fig. 2). After the regenerator, helium moves through the cooler and enters the compression space. The cycle is repeated again. For each new cycle, the curve 3-4 in Fig. 2 moves down, demonstrating lower cooling temperature. With each new cycle, the regenerator precools the refrigerant to even lower temperature. This would happen until the point where heat losses of the regenerator equilibrate the cold charge. The achieved cooling temperature can be considered as the lowest steady state cooling temperature of the Stirling refrigerator. This description shows that the regenerator plays a key role in providing cooling capability for the Stirling refrigerator. Not surprisingly, the results of bibliometric studies for engines and refrigerators will show that the regenerator receives a strong attention of the international scientific community.

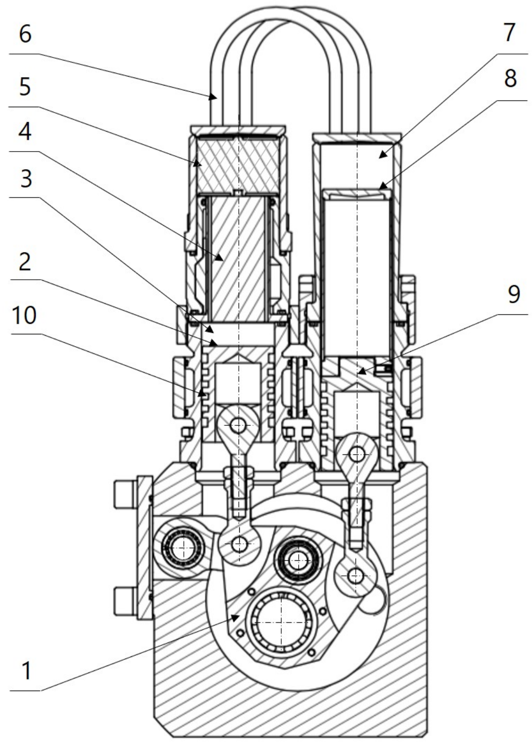


Fig. 3. Internal structure of the Stirling refrigerator TM1.

b) Technical parameters of the Stirling refrigerator and the thermal chamber

Table 1 describes key technical characteristics of the Stirling refrigerator TM1 (Fig. 3) and the thermal chamber (Fig. 4). Subsections in the Table 1 "Cold head", "Piston sealing in cylinders", "Mechanical drive", and "Thermal chamber" have references to supporting figures below the table.

Table 1. Technical parameters of the Stirling refrigerator TM1 and the thermal chamber.

Parameter	Unit	Value
General parameters of the refrigerator		
Working fluid		Helium
Charge pressure range	MPa	0.1÷2.5
Shaft frequency range	Hz	1.8÷9.1
Cooling capacity at 2.5 MPa and 9.1 Hz	W	75
Cold head temperature at 2.5 MPa and 9.1 Hz	K	108
Power consumption at 2.5 MPa and 9.1 Hz	kW	1.51
Width × height × depth	mm	262 × 474 × 205
Mass	kg	35
Compressor		
Bore diameter	mm	48
Stroke	mm	30
Compression volume	cm^3	54.3
Compressor volume gap	mm	2
Dead compressor volume	cm^3	3.6

Expander		
Bore diameter	mm	48
Stroke	mm	23
Expansion volume	cm^3	41.6
Expander volume gap	mm	2
Dead expander volume	cm^3	3.6
Radiator		
Channel length	mm	83
Channel diameter	mm	2
Number of channels		30
Volume of channels	cm^3	7.8
Material of radiator body		Copper
Type of cooling		Colling jacket with tap cold water
Temperature of cooling water	K	287÷295
Cooling water flow rate	$l \cdot min^{-1}$	5.1
Regenerator		
Regenerator diameter	mm	51
Regenerator length	mm	35
Volumetric porosity	%	80
Regenerator void volume	cm^3	57.2
Material		Layers of stainless steel wire mesh
Cold head, Fig. 5		
Pipe length	mm	220
Pipe internal diameter	mm	6
Pipe external diameter	mm	8
Number of pipes		3
Volume of cold head pipes	cm^3	18.7
Pipes material		Stainless steel
Diameter of the bases, D_b	mm	60
Height of the base, H_b	mm	10
Parameter s_1	mm	15
Parameter s_2	mm	20
Parameter s_3	mm	10
Total area of cold head	m^2	0.024
Intermediary gaps		
Gap between radiator and regenerator	mm	1.5
Gap between regenerator and cold head	mm	1.5
Diameter in the gap areas	mm	53
Volume of intermediary gaps	cm^3	6.6
Dead volume		
Dead compressor volume	cm^3	3.6
Dead expander volume	cm^3	3.6
Volume of radiator channels	cm^3	7.8
Regenerator void volume	cm^3	57.2
Volume of cold head pipes	cm^3	18.7
Volume of intermediary gaps	cm^3	6.6
Total dead volume	cm^3	97.6
Piston sealing in cylinders, Fig. 6		
Cylinder material		Stainless steel
Cylinder wall arithmetic mean roughness	μm	0.32
Piston ring material		Ecoflon4: PTFE 75% + Coke 25%
Type of cooling		Colling jacket with tap cold water
Mechanical drive, Fig. 7		
Type of the drive		Ross yoke
Kinematic type		Alpha configuration
Pistons phase angle	deg.	90

Shaft sealing		Two radial lip PTFE seals with oil
Friction loss of shaft sealing at 550 rpm and internal pressure 2.5 MPa		20 W
Thermal chamber, Fig. 4 and 8 (7)		
Width	mm	135
Length	mm	225
Height	mm	275
Wall thickness	mm	62
Material of the wall		Foamed polystyrene
Thermal conductance of the wall	$W \cdot m^{-1} \cdot K^{-1}$	0.0397
Internal surface area	m^2	0.259
External surface area	m^2	0.724
Internal volume	m^3	0.0084
Air fan heat production	W	2
Speed of the air approach flow to pipes of cold head	$m \cdot s^{-1}$	1
Speed of air free-stream flow near chamber walls	$m \cdot s^{-1}$	0.7

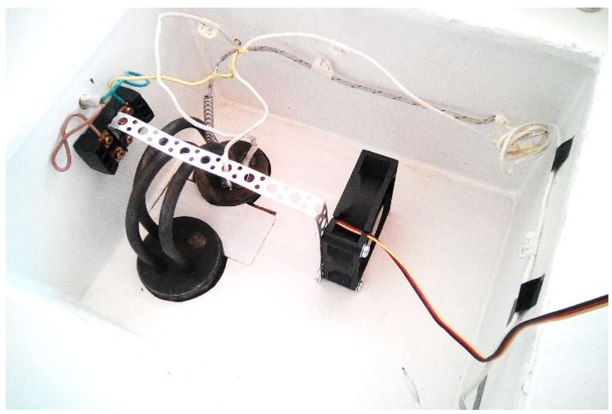


Fig. 4. Internal view of the thermal chamber with the cold head, air fan and thermocouples. External view is in Fig. 8.

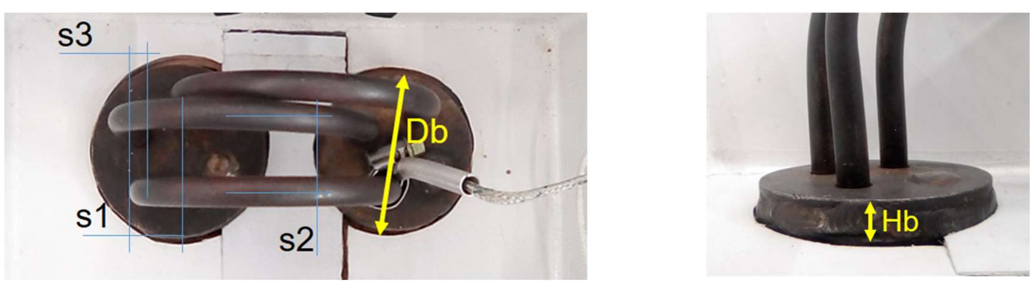


Fig. 5. External views of the cold head inside the thermal chamber with key geometrical parameters: the top view (left) and the approach flow view (right).

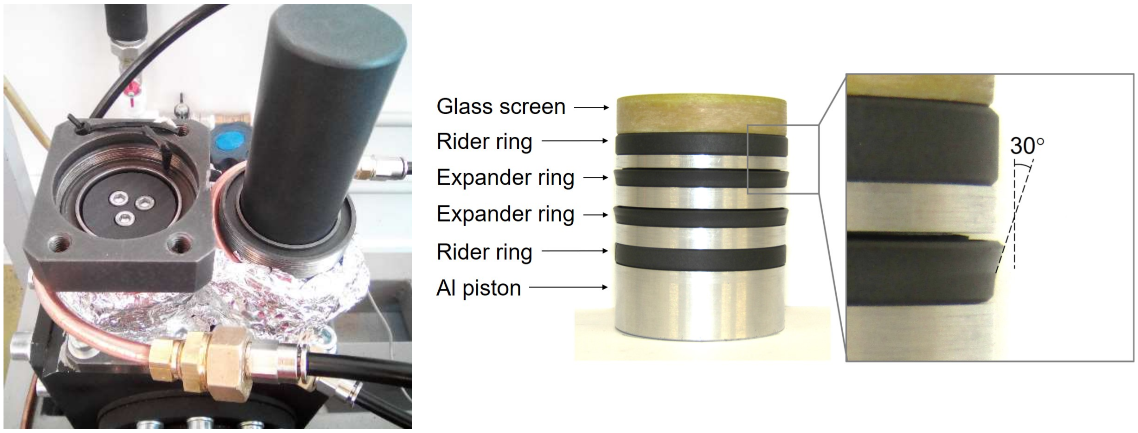
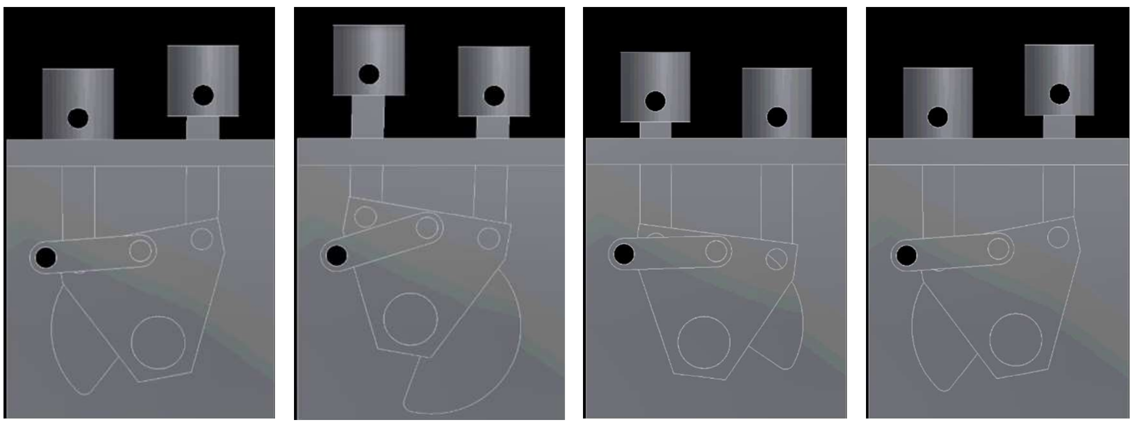


Fig. 6. Piston-cylinder pairs. On the left: Installed piston pairs for compressor and expander with dismounted cold head. On the right: the external view of the aluminum compressor piston with PTFE piston rings.



(a) (b) (c) (d)

Fig. 7. Schematic depiction of the Ross-yoke drive with instantaneous cycle pictures. The compressor piston is on the left, the expander piston is on the right: a) the beginning of compression, b) the end of compression and beginning of expansion, c) the end of expansion d) the beginning of compression. Credit for the dynamic model of the mechanical drive: Vladimir Dvorzov.

c) The structure and technical parameters of the test rig

The principal diagram and the external view of the SR test rig is depicted on Fig. 8. Key sub-systems included: the helium supply system, positions (9)–(13), the cooling water supply system, positions (16)-(22), the electric motor system, positions (23) and (24), the temperature measurement system, positions (4) and (5), and the electrical heating system, position (1)–(3), (7) and (8). Table 2 summarizes measured in the test rig parameters with information about the measurements accuracy.

d) Operational characteristics of the Stirling refrigerator

Fig. 9 depicts the relation between the efficiency of SR, charge pressure and shaft frequency for a better characterization of the objective ECT. The efficiency is characterized by the coefficient of performance (COP), which is the ratio between measured cooling capacity and the consumed grid power including losses in the electrical motor. The pumping power for cooling water and the electrical consumption of measuring devices were excluded from the characteristic in Fig. 9 because the cooling water was taken from the municipal system and the measuring devices were not a part of the original SR design. It is evident from Fig. 9 that the charge pressures for the maximum cooling capacity and the maximum efficiency are not the

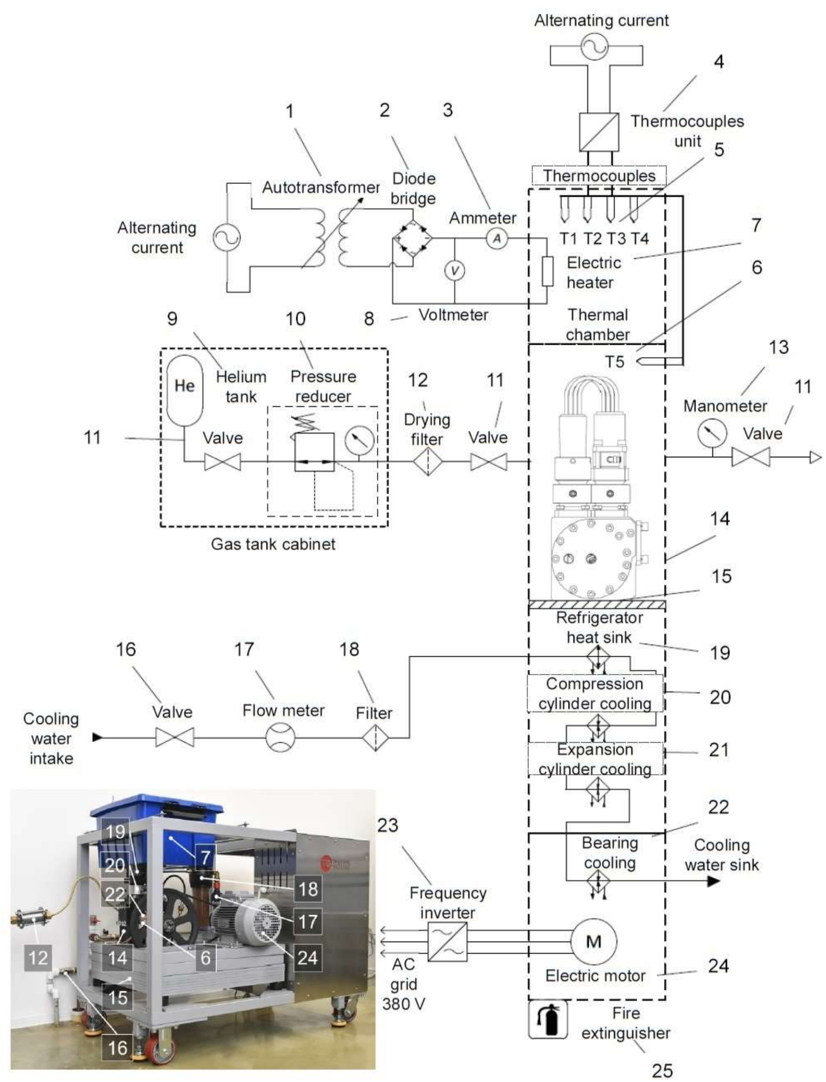


Fig. 8. Principal diagram and external view of the in-house Stirling refrigerator test rig.

Table 2. Measured parameters and instruments in the in-house Stirling refrigerator test rig.

Parameter	Unit	Accuracy	Fig. 8	Instrument
Current of heating load, I_{dc}	A	±2%	3	Digital multimeter Robiton Master DMM-200
Voltage of heating load, U_{dc}	V	±0.8%	8	Digital multimeter Robiton Master DMM-200
Cooling temperature, T_c	°C	±0.25%	5 (T1)	Thermocouple Chromel Alumel (K), –270°C…1372°C
Ambience T_a, electrical heater plate T_{eh}, cooling water T_h, bearing system T_b temperatures	°C	±0.25%	5 (T2, T3, T4), 6 (T5)	Thermocouple Chromel Alumel (K), –200°C…780°C
Charge pressure p,	MPa	±2.5%	13	Manometer YAFU 4.0 MPa
Shaft frequency f	rpm	±0.05%	Not shown	Digital tachometer Sinometer DT2234B
Cooling water flow rate, $\dot{m}_c$	$l \cdot s^{-1}$	±5%	17	Digital flow meter Gardena 8188

same. This effect is present at different shaft frequencies. The percentage difference for COP values at maximum cooling capacity and maximum efficiency for frequency 550 rpm is significant and is equal to 40%; therefore, the question of operational strategy "maximum cooling power VS maximum efficiency" is relevant for future studies and the search of market applications. In the present work, a general strategy was to increase the cooling power and reduce the cooling temperature. These two characteristics are good indicators of the system refrigeration capability and helped to track the progress during the experimental optimization of the system with different piston-cylinder sealing (Chapter V). In addition, this strategy helps to extend the range of possible market applications at an earlier stage and perform early experimental prototyping to demonstrate the proof of concept for different applications.

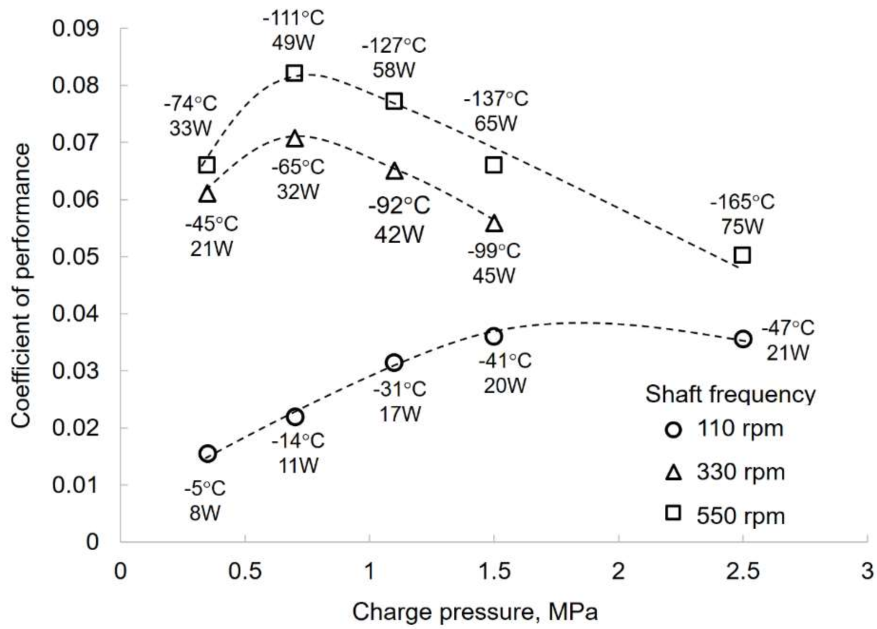


Fig. 9. The coefficient of performance from the refrigerator charge pressure and the shaft frequency. Each point indicates the steady-state cold head temperature and the cooling capacity.

I.2.5 System level definitions

The existing literature rarely differentiates with resounding clarity the difference between the design-independent thermodynamic Stirling cycle (e.g., Fig. 1), the design-dependent Stirling machine (Fig. 3), principle-oriented Stirling technologies, and more complex systems that include a Stirling machine, either engine or refrigeration, as a component. These differences become extremely important in the process of commercial development of Stirling machines. NASA reported on the study to evaluate the technology readiness of the 55W free-piston Stirling engine as “an alternative power technology for NASA's future deep space science missions” [15]. Does this mean that the free-piston Stirling engine is the only type of alternative power technology that can be build based on the Stirling cycle? Another work argued the need to “advance automotive engine technologies such as the Stirling cycle” [16]. Is this cycle a technology or rather a fundamental law? There are accounts of using the terms “Stirling technology” and “Stirling machines” interchangeably [17]. Does this mean that there could not be technologies related to the components of Stirling machines? One can even find anecdotal references to the Stirling refrigerator as an engine. It is essential to differentiate the terminology because at each level of component design in Stirling machines, there are possibilities for innovative and even breakthrough designs configurations and technologies. By calling the Stirling technology a cycle or a machine, we limit those possibilities. In this dissertation, the following definitions were adapted:

The Stirling cycle is a theoretical concept that describes a physical process, in which a working medium (a working gas) consequently undergoes through four thermodynamic processes to produce work or cooling capacity as in Fig. 1 and 2. These four processes include compression with constant temperature, expansion with constant volume, expansion with constant temperature, and compression with constant volume. The states are characterized by a change of macroscopic parameters, such as gas temperature, pressure, and volume. The Stirling cycle is a solution neutral concept, which does not specify how exactly and using what equipment, the cycle is realized in the real world. For simplicity, the cycle can be called a fundamental law for Stirling machines.

The Stirling machine is a solution-dependent mechanical device that realizes the Stirling cycle and has a specific form like in Figure 3. It defines the arrangement of internal channels, the selection of materials, and the exact geometry of its elements. Typically, the design of a Stirling machine is optimized for a particular application to maximize efficiency or power output, or to minimize the cost. The Stirling machine is a component of a more extensive Stirling system.

The Stirling system is a combination of different mechanical, electrical, and software subsystems, including the Stirling machine. These components work together to serve a particular market application.

The Stirling technology in the broadest sense is the method of converting energy using the thermodynamic Stirling cycle. This conversion method may take many different forms represented by various configurations of Stirling machines. Therefore it is always important to specify whether it is the Stirling refrigeration technology or the Stirling engine technology and clarify key design features. Within the domain of Stirling technology there could be many technologies at sub-system level, for examples for the mechanical drive – free-piston drive or crack-shaft drive; or for the level of the regenerator – mesh or porous medium; or for the level of the piston sealing – rings or high-tolerance gap. This variety of possible design configurations calls for a more critical approach to evaluate alternative design decisions at a sub-system level.

These definitions help to describe better the energy-conversion system that was selected as an example to validate a technological pathway in developing a commercial product from different science-intensive ECT.

I.3 OBJECT OF STUDY

This dissertation focuses on the design process of energy-conversion systems as an object of its study in the framework of the design research field [18]. Although the specific technology based on the Stirling cycle is an essential aspect of this work, nevertheless it only informs the results of this dissertation. This work is not seeking to identify market requirements on the demand side of ECT. It assumes that the requirements are known and offers general tools to choose among alternative design decisions when developing a commercial Stirling machines in accordance with some assumed market requirements. The thesis primarily objective is to develop an innovative design methodology, and the intended thesis contribution is to the science of design [19] with a particular focus on the field of science-intensive energy conversion technologies. Strictly speaking, the results of proposed design methods are tested only on the design process of a Stirling machine. The applicability of the proposed design methodology for other technical domains in energy conversion, such as electrochemical systems or solar cells, should be verified first. However, due to a high degree of scientific work embedded in other energy-conversion systems, the material nature of hardware devices required to take control over energy conversion processes, and universal laws of energy conversion and heat transfer, it is assumed that the proposed methodology applies to other types of energy-conversion technologies. Therefore, this dissertation offers a comprehensive design methodology to develop commercial energy conversion systems and unifies design methods at different development stages validated by the experimental process of the development. The results of applying the proposed design methods are reported for Stirling machines in various scientific publications, but the overarching connection of the methods that were used to obtain those results represents a single technological pathway and is the main contribution of the present work.

I.4 RESEARCH QUESTION

According to the seminal work of Eppinger and Ulrich [20], p. 9, a typical product development encompasses six phases: 1) Planning, 2) Concept development, 3) System-level design, 4) Detail design, 5) Testing and refinement, 6) Production ramp-up. Moving from one phase to another requires making design decisions, choosing among alternative design configurations. Phases 1 and 2 typically include extensive literature reviews and require the selection of the conceptual design based on benchmarking. System-level design involves trade-studies to select the design architecture and identify system-level requirements. Detail design is the stage where a group of multi-disciplinary experts is involved to make sub-system design decisions. Phase 5 is concerned with experimental system improvement and with design decisions about the most efficient design alternatives. And the production rump-up would require to scale up the system depending on applications. A closer look would reveal that at each mentioned phase there is a need to make design decisions among available alternatives.

Obviously, these alternative design decisions and the circumstances for their choice would vary depending on the phase and the nature of the project (scientific or commercial). But the need to have a formalized design decision methods based on practical experience still remains at each phase. In Section I.1, I mentioned that the purpose of this dissertation is to propose novel design methods for each mentioned development stage. In the light of the need for formal design-decision methods, the proposed design technics and subsequently this dissertation answer to the question *how to choose among alternative design decisions when developing a commercial Stirling machines*?

I.5 Dissertation structure

The content of this dissertation spins around and is informed by my practical experience of developing a commercial Stirling engine and later a Stirling refrigerator. The nature of this development carried several specific characteristics: constrained resources [21], stemming from this finite nature of project bounded rationality of decision-making [4], focus on increasing commercial value by fulfilling market requirements rather than following scientific objectives [22], and the need for a novel design because of the required patent protection and generally competitive organizational landscape [23]. At each development stage (Fig. 10, the framework of Eppinger and Ulrich discussed in the previous section), there was a specific design challenge that required scientific research to find a solution. Therefore, the objective of every chapter was modulated by a particular design challenge faced at a related stage. The following paragraph will discuss the design challenges and connect to them relevant thesis chapters. But before turning to this, it is essential to mention that the nature of design challenges changed along the process of technology maturity, which I characterized in Figure 10 by the TRL (Technology Readiness Level). I intentionally placed chapters in a chronological order along the process of maturing technology to demonstrate what design and organizational challenges one would face at each step of development, and what design methods would be necessary to organize the process of efficient selection of alternative design configurations for the commercial energy-conversion systems.

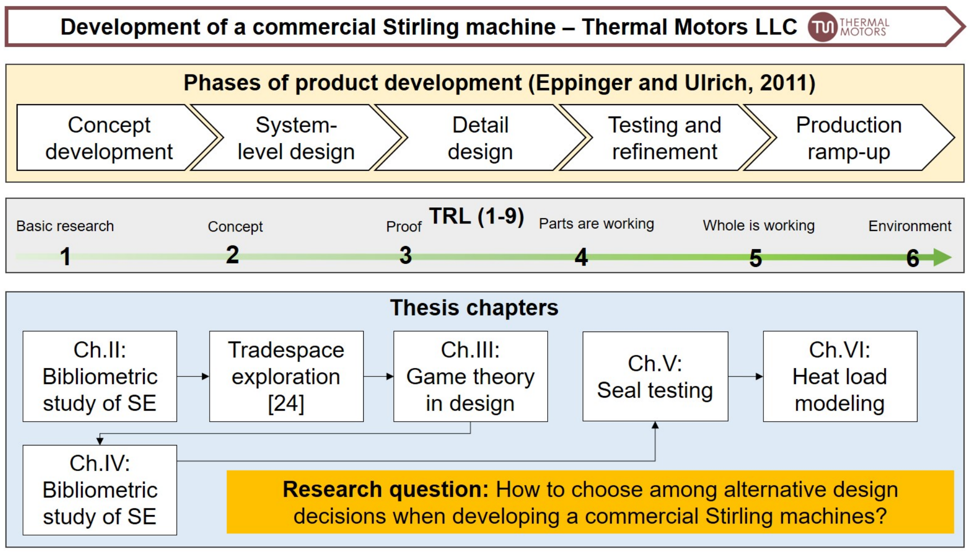


Fig. 10. The relation between chronological development of a commercial Stirling machine and the chapters of the thesis.

As mentioned before, the concept stage required a novel and competitive concept design. The original focus of the development was on Stirling engines. The main design problem was how to identify existing design configurations of Stirling engines and analyze their performance. Chapter II proposes a method for the literature review to identify alternative design configurations of Stirling engines developed both in the academia and in the commercial sector. Unlike a conventional selective literature review, this chapter proposes an alternative approach to study literature based on big data technologies. It represents a substantial amount of work atypical for results of traditional literature reviews and therefore deserves a separate chapter in this thesis. Before discussing the approach further, it is worth mentioning that besides technical literature, there is another valuable source of innovation, which is the input from potential end-users during market analysis. The objective of the concept stage is to generate novel design alternatives, and investigation of what customers can suggest consciously or unconsciously is important. This input is different from system-level requirements derived from the initial market analysis and assumed to be known for the analyzed ECT in this work. Different design configurations suggested by end-users can satisfy the same system-level requirements. Therefore, the analysis of the market can recommend innovative design configurations that can be embedded in the list of concepts for design. In other words, literature review offers a tool that reviews “supply side” alternative design solutions. Market analysis is a tool that can generate “demand side” design alternatives. Ideally, both tools should be employed at the concept stage. However, given the assumed earlier in this chapter “technology push” character of ECTs, there are some methodological limitations to integrating market analysis into this dissertation, mainly limited information in the market for previously unknown technologies and resourcefulness to obtain reliable market information.

There is a very limited information about novel energy-conversion systems that can be derived from the market. Novel energy conversion technologies classified in the category of “technology push” solutions include such problems as: “Information with regard to customer needs, potential applications, competitors and suppliers is missing” [1]. The author interviewed at least ten companies in engine and refrigeration sector each as a part of the commercial effort. Technical experts from companies do not know Stirling cycle systems or have no experience in operating them. Therefore, they could not formulate a list of requirements, were unwilling to offer one because they did not want to discuss technical systems that would imply additional significant investments, or did not consider potential problems in operation of conventional ECT systems as problems, but rather as normal operational conditions. Therefore, it is unlikely that they would recognize technical problems, and as a result offer alternative design solutions or create the demand for the application of new technologies. As an example, the growing LNG sector has a problem with the generation of explosive boil-off methane in fuel tanks, which – if not used in the combustion during the driving mode – is emitted in the atmosphere through the blow-off valve every 5 days. The Stirling refrigerator could be used as a small and efficient ancillary system to condensate the gas back to the liquid form and extend the drainage free operation to 365 days a year. Interviews with three technical experts from different Russian companies that sell LNG fuel tanks indicated that they do not perceive boil-off gas drainage as a problem, but as a normal operating condition of equipment. And yet, they recognized that this condition requires additional measures to optimize how often the vehicles should be driven, specific ventilation equipment and receiving safety licenses for the building. This indicates, at least inherently, that there is a problem of increased costs for LNG fuel tanks. Should the experts knew about a solution to the boil-off problem, such as Stirling refrigerators, they could call the generation of gas “a problem”. The only industrial field with mature understanding of Stirling cycle systems is aerospace and military that are outside of the author’s interest. Technologies based on the Stirling cycle are known to limited groups of scientists and engineers from different institutions and technology companies. Getting access to them for reliable input is expensive. This conclusion leads to the second methodological problem.

Reliable market analysis for the thesis is a resourceful task in working hours and monetary value. It is difficult to find a significant number of potential users of the system that would be a reliable representative group due to the problem discussed above. In addition, there are more than five possible application segments for Stirling refrigerators. Collecting high quality and statistically significant market information for each of these segments is difficult in the absence of market information. An alternative validation sources could be market studies conducted by analytical agencies. But the licenses for these documents are at least $3000 per user, there is a limited number of such agencies that can produce a reliable results for Stirling machines, and the results would be most likely rather generic. Other sources are not reliable.

As an alternative to a direct market analysis, the analysis of patent literature can be considered as a substitute. The patent literature is not only the product of researchers. It also reflects the situation at the demand side. For example, the results of patent analysis that will be discussed in details in Chapters II and IV supports this assumption. The study of patent literature for Stirling engines (Fig. 6, Chapter II) before 2006 showed that all ten top-filing organizations (46% of all patent applications) were companies. After 2006, five out of ten top-filing organizations (8% of all patent applications) were companies. Similar situation was observed for the Stirling refrigerators (Fig. 6, Chapter IV). Companies are economically rational agencies that reflect the requests at the demand side. Patent literature can be used with caution to gain an insight in customer preferences.

Given significant limitations to conduct a reliable direct market analysis, the author decided to conduct a high quality and in-depth analysis of at least knowledge sources at the supply side (technical literature) with a claim that patent literature is a good indicator of the end-user expectations. Naturally, this approach does not capture some valuable insights from the market analysis and this drawback could be considered as a limitation of this work. On the other hand, the complexities associated with obtaining reliable market information encouraged the author to put this analysis outside the scope of the thesis.

The next step of the commercial project was to identify what engine architectures to choose among many available alternatives for predetermined market applications. Our key application idea at the time was waste heat recovery. For this reason, during my MSc work, I developed a novel method with the application of tradespace exploration to numerically model the performance of different Stirling engine architectures and to organize an easy and visual process to select most desirable architectures for different commercial applications [24]. Although the developed method as such and results based on this method are not included in this thesis, its integration into the novel technological pathway is relevant and explained in the discussion section of Chapter II. The next development step was to initiate detail design and construction of the selected engine architecture. This project stage involved a group of disciplinary designers, engine component developers, and suppliers working under several constraints, including conflicting objectives, communication barriers, fixed allocated time with a limited budget and bounded rationality. The main design problem at the time was the management of the interplay between designers and the influence of this interplay on the final design outcomes. To that effect, Chapter III offers a novel mathematical design optimization method based on game theory. This method accounts for the influence of designers on the outcome of the design process carried out under information uncertainty, conflicting objectives, and bounded rationality. From the engineering point of view, two main questions had to be answered for the expected operation of the Stirling engine: the design of high-temperature sealing mechanism and the efficient heat transfer from the external source to the engine. Before the engine tests, we conducted a typical routine of break-in tests at the 500 rpm shaft speed for the mechanical drive to wear freshly manufactured components. During the break-in tests, we observed a strong refrigeration effect. Be it an engineering or scientific project; the focus would continue on the engineering or fundamental activities related to the Stirling engine. However, the main factors that drove this project were commercial: profitable application, limited resources, and bounded rationality of the decision-making. The observation of the refrigeration effect initiated the evaluation of the refrigerator market applications, which was carried in parallel with engineering development. The result of these two parallel processes was the shift in the commercial and design focus from the engine to refrigerator applications. As discussed in Section I.2.3, the shift from the engine to the refrigerator using the same design configuration would most likely limit the performance of the refrigerator. However, the question is by how much? If the operation of the refrigerator can demonstrate satisfactory characteristics for some market applications, then the design can still be implemented at a reduced development cost. Also, the operation of the refrigerator generates real indispensable data that can be used to optimize the design and reach the highest performance. The change of engineering and application focus, which is also often called a "pivot" [25], led to a similar bibliometric analysis of scientific and patent literature reported in Chapter IV for Stirling refrigerators. The next stage of the project was the process of testing and improving key performance characteristic of the refrigerator to serve several market segments. The key design challenge at this stage was to reach a high performance of the system, that would warrant the possibility to target as many market segments as possible and thus to maximize the return on the R&D development. The main technical obstacle was the sealing mechanism between the piston and cylinder. Chapter V focuses on the experimental optimization of this technical element. After achieving satisfactory key performance characteristics of the experimental Stirling refrigerator, the next question was how to scale the developed system for different commercial applications? Chapter VI discusses the process of developing a numerical model validated by experimental results to predict the performance of the Stirling refrigerator for different operational and design parameters. Chapter VIII offers concluding remarks about the limitations of this work and its prospects. With some clarity about the case-study technology, the structure of the thesis and initial comments about the proposed design methods, the next section announces and discusses the thesis statement.

I.6 Thesis statement

The process of selection between alternative design decisions at different design stages during the commercial development of energy conversion systems requires a unique set of design methods – the bibliometric big data analysis of scientific and patent literature, tradespace exploration, application of game theory, experimental optimization of critical components, and technology scaling based on experimentally validated model – to respond to commercial factors of the development process. It is challenging to formulate a more specific statement for the work that focuses on several design stages, each having a specific design and commercial characteristics. However, the statement connects different elements of this dissertation into an overarching treaty. When developing commercial ECTs, the process of choosing design alternatives is altered by the presence of commercial requirements. This work argues that several novel design methods shall be used to respond to the pressing forces of commercial development effectively. Specific commercial factors influence each development stage, and it is essential to discuss now how commercial requirements affect the choice of design methods.

I.6.1 CONCEPT DEVELOPMENT (CHAPTER II AND IV)

At the concept stage of the commercial development, it is crucial to obtain *competitive and novel concept* alternatives that would satisfy general market requirements supplied as an input to the concept stage. Design concepts need to be competitive with existing direct competition and novel for intellectual property (IP) protection. Therefore, there is a need for a method that helps to find high number of alternative design configurations in scientific and patent literature for the comparison and the development of competitive design concepts. To respond to this design challenge, I developed a bibliometric approach to literature review using big data technologies, which is described in Chapters II and IV.

I.6.2 SYSTEM DESIGN (CHAPTER II)

At the system design stage of commercial development there is a need to select *designs for market requirements*. On the one hand, there is a specific set of system-level market requirements, and on the other hand, there is a significant number of alternative design configurations that could be produced, but that do not necessarily satisfy those requirements. There is a need in a design method that helps to compare and downsize the set of alternative system designs. I developed the objective method during my MSc thesis by adapting the "tradespace exploration" methodology from systems engineering literature. However, the application of this method in this dissertation is essential, because the method links the concept stage and the detail design stage, which are both studied in this work, in a coherent and novel life-cycle design methodology for the development of commercial energy conversion systems. The relevant discussion of this method is given in the discussion section of Chapter II.

I.6.3 Detail design (Chapter III)

The development resources during this important design stage are finite. There are limited time and funding for the development, and there is the pressure of commercial competition to place the product in the market before the competitors. A group of disciplinary designers begins a detailed development of the system. One of the most challenging design problems at this stage is reaching an optimal design outcome for *collaborating disciplinary designers under finite resources*. This problem calls for a design optimization method that accounts for the designers' interaction during the design process. Such method was developed with the application of game theory.

I.6.4 Testing and refinement (Chapter V)

Many technical aspects in the operation of a real system are difficult to predict during the detail design. This limitation poses a question. Will the experimental system reach *market-level performance*? At this stage, it is important to reach a satisfactory level of performance to meet market expectations under the complications caused by the operation of components that are both critical for performance parameters and work under highly stressful conditions and difficult to model. In Stirling machines, one of these components is the piston-cylinder sealing. The optimization of such elements cannot be performed effectively only by numerical modeling. There is a need for experimental optimization. This design approach is demonstrated on the example of optimizing the piston-cylinder pair in Chapter V.

I.6.5 PRODUCTION AND RUMP-UP (CHAPTER VI)

The main design challenge at the production stage is how to *scale technology for different applications*. The main objective is to size the experimental system for different commercial applications. It is critical to be able to scale the system for a different set of requirements, because this set of requirements changes with new information about customers and because more market applications should be covered to maximize the return on the R&D investment. Therefore, there is a design challenge to develop an accurate physical model of the technology to be able to size it. To solve this problem for Stirling machines, in Chapter VI, I offer a model for a Stirling machine verified by experimental data.

This discussion vividly shows that different development stages and related commercial factors create the need for a set of unique design methods for Stirling machines. Similar design challenges may be relevant for the development of other science-intensive energy-conversion systems. The interplay of these design methods is depicted in Fig. 11, where on the left, there is a development stage and on the right – the defining commercial factor that both play a decisive role in rendering the required design method. Chapter II and IV focus on the bibliometric study. Chapter III discusses the tradespace exploration and studies in details the application of game theory in design. Chapter V demonstrates the method of experimental optimization of critical components in Stirling machines, and Chapter VI reflects on the process of scaling the Stirling machine.

The combination of design methods in Fig. 11 defines the innovative technological pathway to design commercial energy systems, and in particular, Stirling machines. A reader may ask, "Was it necessary to focus on both the engine and the refrigerator? The author could develop the pathway only for the Stirling refrigerator." I agree that since I conducted the same bibliometric study for the Stirling refrigerator, the thesis could have had a more straightforward structure by informing the proposed design pathway only using the results for the Stirling

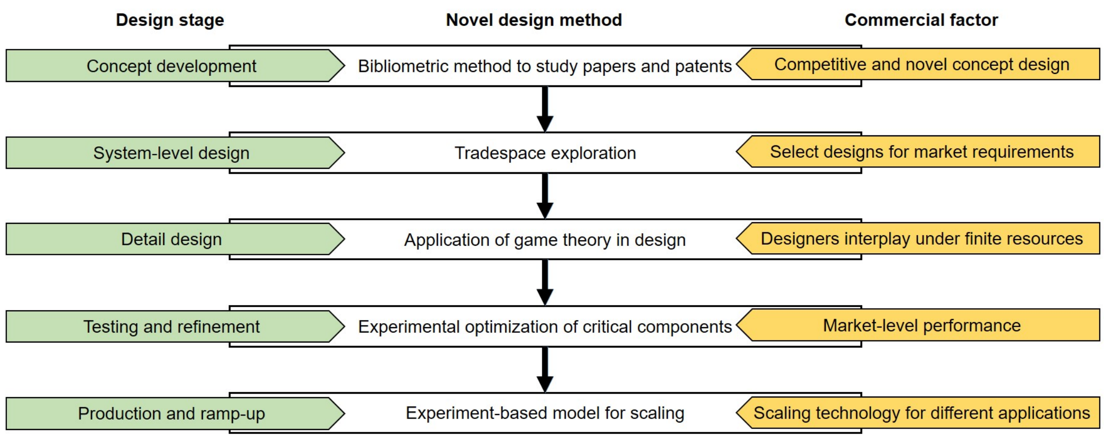


Fig. 11. The visual representation of the main thesis of this work.

refrigerator. However, I believe it is vital to use the results for both the engine and the refrigerator due to several reasons. I intended to use the real development process with all challenges and the pivot as they were as a core storyline for the thesis. This process started with the engine and later focused on the refrigerator. The change of commercial focus from the engine to the refrigerator finds some precedents in history [26], and technology developers would most likely observe the engine and refrigeration capability in the systems that they develop regardless of the initially intended application, which makes the proposed thesis structure relevant. I also believe that a similar pivot may occur in the development process for other energy-conversion technologies with an active scientific component. The technological pathway follows through several stages, including the system-level design stage. At this stage, the results were obtained for the Stirling engine; therefore, it was essential to include the Stirling engine in the thesis content. Besides, the object of this thesis is the design methodology, and it is of the secondary importance whether the proposed methodological recommendations were tested on the Stirling engine or the refrigerator. Last, but not least, both types of energy conversion systems operate the same thermodynamic principle; however, only a small number of studies attempted to analyze both technologies within the same methodological framework to compare their characteristics. These reasons encouraged me to focus on both the Stirling engine and Stirling refrigerator in this thesis. It would be difficult to exclude the Stirling engine without breaking the continuity of the proposed technological pathway.

I.7 Novelty and contribution

I.7.1 Chapter II

- Developed a method of analysis for scientific and patent literature with the use of big data technologies and bibliometric information;
- Identified historically accurate account of scientific and patenting activities for Stirling engine over the last 58 years;
- Identified statistically significant distribution of research topics about Stirling engines;
- Developed a theoretical framework that defines factors – apart from technical – that need to be taken into account when making design choices at the concept stage for commercial development of Stirling engines and other energy conversion technologies.

I.7.2 Chapter III

- Developed a formal approach to apply game theory in the process of engineering design to account for the influence of disciplinary designers on results of detail design outcomes. Proposed practical recommendations of the design proces;
- Developed the notion of Nash front as an alternative concept to the Pareto front, which accounts for the influence of design authority reallocation in a multidisciplinary design process;
- Calculated the difference between Pareto-efficient solutions and Nash-efficient solutions for a two-team design process for the Stirling machine;
- Proposed algorithm to apply game theoretic framework to other energy conversion systems apart from Stirling machines.

I.7.3 Chapter IV

- Identified historically accurate account of scientific and patenting activities for Stirling refrigerators over the last 58 years;
- Identified statistically significant distribution of research topics about Stirling refrigerators;
- Developed a theoretical framework that defines factors that need to be taken into account when making design choices at the concept stage for commercial development of Stirling refrigerators and other energy conversion technologies.

I.7.4 Chapter V

- Systematized the knowledge about existing designs of piston-cylinder seals with advantages and disadvantages;
- Conducted experimental optimization of the piston-cylinder sealing in a real Stirling refrigerator with the alpha configuration of the Ross-yoke mechanical drive with the improvement of the minimum refrigeration temperature from 173 K to 108 K;
- Obtained the performance of the high-tolerance piston cylinder seal with the 10 µm gap;
- Demonstrated how the performance of the Stirling refrigerator can be improved through the experimental optimization of the sealing mechanism.

I.7.5 Chapter VI

- Developed a physical model validated experimentally of the system "Stirling-refrigerator – thermal chamber" by unifying the thermo-electrical analogy model of the thermal chamber and the model for the operation of the Stirling refrigerator;
- Discovered the effect of declining value for the refrigeration performance parameter, Otaka number, previously believed to be a constant value;

- Offered a relatively simple, yet effective approach to scale ECTs using mathematical modeling and experimental validations.

References

[1] Herstatt, C. and Lettl, C., 2004. Management of" technology push" development projects. *International Journal of Technology Management*, *27*(2-3), pp.155-175.

[2] Banal-Estañol, A. and Macho-Stadler, I., 2010. Scientific and commercial incentives in R&D: research versus development? *Journal of Economics & management strategy*, *19*(1), pp.185-221.

[3] Barge-Gil, A. and López, A., 2014. R versus D: estimating the differentiated effect of research and development on innovation results. *Industrial and Corporate Change*, *24*(1), pp.93-129.

[4] Simon, H.A., 1990. Bounded rationality. In *Utility and probability* (pp. 15-18). Palgrave Macmillan, London.

[5] Saychenko, A., Smirnov, D. Application of Stirling engines coupled with Gas Compression Units for automation and thermal efficiency increase. *Open Applied Science Conference of Yong Professionals in Gazprom: proceedings*, 2010, in Russian, pp. 238, p. 38.

[6] Lambrecht, J., 2005. Multigenerational transition in family businesses: A new explanatory model. *Family business review*, 18(4), pp.267-282.

[7] Carayannis, E.G., Cherepovitsyn, A.Y. and Ilinova, A.A., 2016. Technology commercialization in entrepreneurial universities: the US and Russian experience. *The Journal of Technology Transfer*, 41(5), pp.1135-1147.

[8] Lauff, C.A., Kotys-Schwartz, D. and Rentschler, M.E., 2018, August. Design Methods Used During Early Stages of Product Development: Three Company Cases. In *ASME 2018 International Design Engineering Technical Conferences and Computers and Information in Engineering Conference*. American Society of Mechanical Engineers.

[9] Formosa, F. and Despesse, G., 2010. Analytical model for Stirling cycle machine design. *Energy Conversion and Management*, 51(10), pp.1855-1863.

[10] Teruyuki, A., Koichi, H., Takeshi, H. and Kazuhito, F., 2014. The performance of Stirling engine of the free piston type enhanced with SiC ceramics heater. *Modern Machinery Science Journal*, p. 537.

[11] Hermes, C.J. and Barbosa Jr, J.R., 2012. Thermodynamic comparison of Peltier, Stirling, and vapor compression portable coolers. *Applied Energy*, *91*(1), pp.51-58.

[12] Martini, W.R., 1978. *Stirling engine design manual*. US Department of Energy, Office of Conservation and Solar Applications, Division of Transportation Energy Conservation.

[13] Kostionk, V. V. (2003). *A Text Book of Cryogenics*. Discovery Publishing House, p.131.

[14] Ghosh, S. and Seering, W., 2014, August. Set-based thinking in the engineering design community and beyond. In ASME 2014 *International Design Engineering Technical Conferences and Computers and Information in Engineering Conference* (pp. V007T07A040-V007T07A040). American Society of Mechanical Engineers.

[15] Furlong, R. and Shaltens, R., 2000. Technology assessment of DoE's 55-We Stirling technology demonstrator convertor (TDC). In *Collection of Technical Papers. 35th Intersociety Energy Conversion Engineering Conference and Exhibit (IECEC), (Cat. No. 00CH37022)*, Vol. 2, pp. 1213-1221. IEEE.

[16] Nightingale, N.P., 1987. The Development Status of an Automotive Stirling Engine. In *Automotive Engine Alternatives*(pp. 125-142). Springer, Boston, MA.

[17] Ross, B., 1995. Status of the emerging technology of Stirling machines. *IEEE Aerospace and Electronic Systems Magazine*, *10*(6), pp.34-39.

[18] Bayazit, N., 2004. Investigating design: A review of forty years of design research. *Design issues*, *20*(1), pp.16-29.

[19] Simon, H.A., 1988. The science of design: Creating the artificial. Design Issues, pp.67-82.

[20] S. Eppinger and K. Ulrich, Product Design and Development. McGraw-Hill Education, 2011

[21] Cooper, R.G., 2006. Managing technology development projects. *Research-Technology Management*, *49*(6), pp.23-31.

[22] Banal-Estañol, A. and Macho-Stadler, I., 2010. Scientific and commercial incentives in R&D: research versus development?. *Journal of Economics & management strategy*, *19*(1), pp.185-221.

[23] Pennetier, C., Girotra, K. and Mihm, J., 2019. R&D Spending: Dynamic or Persistent?. *Manufacturing & Service Operations Management*.

[24] Smirnov, D. and Golkar, A., 2015. Stirling engine systems tradespace exploration framework. *Procedia Computer Science*, 44, pp.558-567.

[25] Crilly, N., 2018. 'Fixation'and 'the pivot': balancing persistence with flexibility in design and entrepreneurship. *International Journal of Design Creativity and Innovation*, *6*(1-2), pp.52-65.

[26] Walker, G., 1986. Miniature Stirling cryocoolers: trends in development. *Cryogenics*, 26(7), pp.387-391.

CHAPTER II
BIBLIOMETRIC STUDY OF STIRLING ENGINES

The conceptual stage of design process requires the generation of alternative design concepts. The present chapter offers a method to perform this task and tests this method in the conceptual analysis of Stirling engines as an example of energy conversion technologies (ECT). Chronologically, the concept generation stage for the development project of the Stirling machine occurred before the period of my Ph.D. studies; however, the method itself was developed in 2017 because of frequent and non-systematic inquiries to the academic and patent literature. The purpose of these inquiries was the identification of alternative design configurations, possible market applications, and design configuration of competitors. Apart from that, I was interested in making sense of historical technology development, leading geographical regions and sources of published knowledge. As was mentioned in Section I.6.1 in Chapter I, design concepts need to be competitive with existing direct competition and novel for intellectual property (IP) protection while satisfying the market requirements identified prior to the concept stage during marketing research. To address this design challenge and systematize knowledge inquiries, I developed a method that helps to research existing scientific and engineering literature using big data technologies. A similar analysis would likely be relevant for other ECT, apart from Stirling machines. Therefore, this method can also be applied to the analysis of alternative ECT and is recommended for the implementation at the conceptual design stage.

I INTRODUCTION

The design process of Stirling machines for commercialization inherently contains the problem of choice. What design alternatives are available and how to choose among them? This design problem can easily be illustrated by the extraordinary variety of design configurations for Stirling machines developed by commercial companies like Toyota [1], Hyundai [2], Yanmar [3], and Isuzu [4]. The most common approach to choose among design alternatives is to perform design optimization. This procedure includes the maximization of conflicting figures of merit –quantitative parameters of technical performance – attributed to objective design alternatives. The design configuration with the highest figure of merit is then selected. The term "design configuration" here means any alternative specific design of a system element with the same functionality. Such a machine element could be a piston sealing or the mechanical drive. The evaluation process of alternative design configurations very often starts with a numerical modeling and sometimes proceeds with actual experiment. In most instances, a literature review precedes technical design and experiment to evaluate the state of art, existing approaches for alternative designs, and their modeling and experimentation methods. It is easy to see that a selected approach for literature review strongly influences the final selection of design alternatives and the success of a commercial machine.

One of key criteria for literature review according to Boote and Beile [5] is *coverage*, which justifies for inclusion or exclusion of research. However, authors offer a cursory discussion on techniques to obtain the appropriate coverage. Randolph [6] lists three coverage scenarios, including *exhaustive review* with considering every available research relevant to the topic, *representative sample* of documents with inferences about the entire population of research from the sample and *purposive sample* with the examination of transformative articles in the field. In addition, the author offers some practical recommendations for the exhaustive search of relevant research papers, suggesting to search electronic databases, perform the analysis of paper references, and ask experts for help. Several studies, for example Daim *et al*. [7], extended these techniques by using *bibliometrics* and *patent analysis* to derive insightful conclusions about state of art in the field.

Most academic studies about Stirling machines employ exhaustive reviews of specific topics in scientific literature. In theory, this approach helps to identify the most relevant papers and study specific research questions; however, it is a demanding task to devise a truly exhaustive review [6]. Therefore, typical reviews have many gaps [5] and at their best are more selective rather than exhaustive. In addition, unlike for scientific research where the focus is on a very specific problem, during technology development process it is important to have a list of design alternatives to choose from; therefore, the development of Stirling machines requires information about different design alternatives that often goes beyond specific scientific questions. Selective scientific literature reviews have three key limitations that negatively affect the quality of the design selection process and, as a result, the competitiveness of the commercial Stirling machines: the wider picture of the existing knowledge is often missing; the evolution of research focus for key countries, research organizations, and publication sources is difficult to trace because some vital research studies could be missed by a researcher; and the choice of sources is often limited to academic literature and does not include patent documents.

Missing the wider picture of research activities has several consequences for the quality of a selected design configurations. It is difficult to select relevant and reliable design alternatives without the knowledge of what countries, research organizations, and publication sources contributed the most into the knowledgebase of a specific technical domain. Selective review may be misleading about what technical choices yield high operational performance or how mature a design alternative is. In addition, it is difficult to understand application areas without the knowledge of active research organizations and their motivations.

The lack of understanding how key countries, research organizations, publication sources, and their focus changed over time also may have important implications on the selected design configurations. The evolution of activities could be an indicator, for example, of relevance, complexity and maturity of research in a specific domain. Without analysis of these indicators, the design choice may be poor. The evolution of published documents clearly shows when and how much knowledge about a technical domain was generated over time. The evolution of research outcomes by countries or organizations may also represent the degree of knowledge concentration and opinion influence, which may or may not be a good design influence and may require to be balanced by alternative design choices before making final selection. The change of publication sources can demonstrate, where the community of the qualified experts in the field publish their findings.

The last limitation I mention is the very scarce amount of patent documents in selective reviews, which has a negative effect on the choice of design configurations. Patent documents are a rich source of different design configurations. A similar bibliometric analysis helps to identify design alternatives and allows discuss of their performance.

These shortcomings show that the sole process of reviewing literature for a relevant research area using only bottom-up selective reviews is flawed and has a negative impact on the quality of selected design configurations. In order to solve these shortcomings, an additional top-down approach is needed. In this chapter we develop and discuss a bibliometric approach to literature analysis in the domain of Stirling machines based on big data from academic literature and patent databases to give a historical account of how research and patenting changed over time in this technical domain. The results of this analysis provide valuable insights into the state of the art of Stirling machines and into the principles of selecting their design configuration alternatives.

II METHODOLOGY

II.1 DATABASES

The data used in the analysis of academic literature was derived from two leading indexing databases – Scopus and Web of Science. Although there is a significant overlap in some of the data, this complementary approach serves two purposes. First, for some periods it allows validating common trends and results – such as top-ten publishing countries or top-ten publishing organizations – by comparing data from two independent and reliable sources. Second, it allows covering documents that are unique to each database.

This approach has limitations. During some periods, there can be a significant difference in the number of published documents in the two databases. The data itself pose several limitations for the analysis. For example, when categorizing publications by a publishing source, some documents in databases have unidentifiable sources. Countries where English is not the first language have publications in native languages and are not referenced in selected databases. There also exists classified research, which is not indexed. These shortcomings may not allow reflecting adequately the state of art in the domains. Although these limitations do affect the accuracy of results, their impact is considered insignificant to undermine the outcomes of the research. A different number of published documents and trends in two databases illustrates a more complete situation in the technology development. In most instances, unidentifiable parameters of data are related to conference publications that have lower quality standards as compared to peer-reviewed journal papers and thus do not affect the results significantly. Here, I do not attempt to provide an exclusive referencing of papers; therefore, we consider Scopus-referenced documents to be representative of all country activities regardless of the language of papers. Although, classified research may exhibit a more advanced state of art, it is not in the public domain and cannot effectively be used for commercialization purposes.

The data for the analysis of patent activity were drawn from two databases – Scopus and Cipher. Scopus contains data about publications that are primarily related to scientific work. Cipher, on the other hand, is more related to engineering and technological literature by citing only patent documents.

The patent data from Scopus database have a number of limitations. First, it does not provide the categorization of patent documents by country or organization, similar to the analysis of scientific literature. Second, the data do not differentiate between patent applications and granted patents. Furthermore, the unit of Scopus database analysis is any patent document registered in patent offices, and not a patent family, which accounts for the same inventions filed in different countries. This approach may cause double-counting of patents from the same family. Therefore the data from Scopus were used only for the analysis of patent activities over time.

The Cipher database enables required categorization of patent documents. However, the data for patent family applications and for granted patents are available only beginning from 1980 and 2000 respectively. Nevertheless, in case of granted patent families, the data indicate the total number of granted patent families to date, which allows counting granted patent families before 2000, beginning from 1980s. The data for patent family applications are avaliable from 1980.

I believe there is a considerable amount of scientific and engineering activities that are not published due to reasons not mentioned above. This knowledge exisits in the tacit form. As a result, the conclusions derived based on data from Scopus and Cipher only, may not represent the actual status of Stirling engine systems. Interviewing domain experts and visiting specialized labs would be necessary to draw a better picture and could be addressed in future work. Presently we assume that Scopus and Cipher provide sufficiently full account of global research about Stirling machines for the purpose of the current research.

II.2 Technical domains

Two technical domains in Stirling technologies were explored in the databases: the engines and refrigerators. For the former domain I used the search term *"Stirling engine"*; for the latter, a combination of terms that reflects varying terminology in the field: *"Stirling AND (refrigerator OR cooler OR cryocooler OR "heat pump")"*.

II.3 Data analysis

For the scientific literature the data were categorized by the number of documents published each year between 1960 and 2018. It was further categorized by countries, organizations and publications sources. To identify the most common research topics in leading publishing countries, we analyzed a number of leading publishing organizations within each country. The exact number of leading countries and research organizations in each of two analyzed domains varied depending on the variety and the amount of published work, but was typically ten. For Stirling engines, we analyzed three countries – United States, Japan and China. And for the Stirling refrigerators five countries – United States, Japan, China, France and Netherlands. The analysis of research topics was conducted based on the information provided in paper title and abstract. In cases, where the information was not sufficient, I analyzed the main text of the paper. The results section shows only the analysis of Scopus data because it referenced significantly more documents than Web of Science and to avoid unnecessary volume of data presentation.

In order to track the evolution of patent activity, the following approach was used. We normalized the number of patent documents by the maximum value per year of the corresponding group of patent documents because absolute numbers for patent documents per year in Scopus are significantly higher than for Cipher due to the lack of differentiation between patent applications and granted patents, and a patent and a patent family in Scopus. The categorization of granted patent families was made by territory and territory expenditure in US dollars. The categorization by territory expenditure allows assessing commercial commitment of organizations to commercialize patents on the indicated territory. Additionally, we made the categorization of patent family applications and granted patent families by organization.

III RESULTS

According to Scopus, during the period between 1960 and 2006, there were 1942 scientific documents about Stirling engines published worldwide. In the period between 2007 and 2018, the number was 1455 documents. This means that the last twelve years generated 43% of scientific knowledge about Stirling engines referenced in Scopus during the past 58 years. Here we present 6 figures that help to understand the scientific and patent worldwide publishing. Figure 1 depicts the publishing trends of scientific and patent documents over that time. Figure 2 depicts the geographic affiliation of published research before and after 2007. Figure 3 shows the scientific sources where research about Stirling engines was published before and after 2007. Figure 4 illustrates the geographic distribution for the cumulative number of granted patents and patent expenditure since 2000. Figure 5 helps to look into the structure of organizations affiliated with 1942 scientific documents published before 2007 and 1455 documents for the period between 2007 and 2018. Figure 6 helps to compare the results in Figure 5 with the record for patent family applications for the period between 1980 and 2006 and the period from 2007 to 2018.

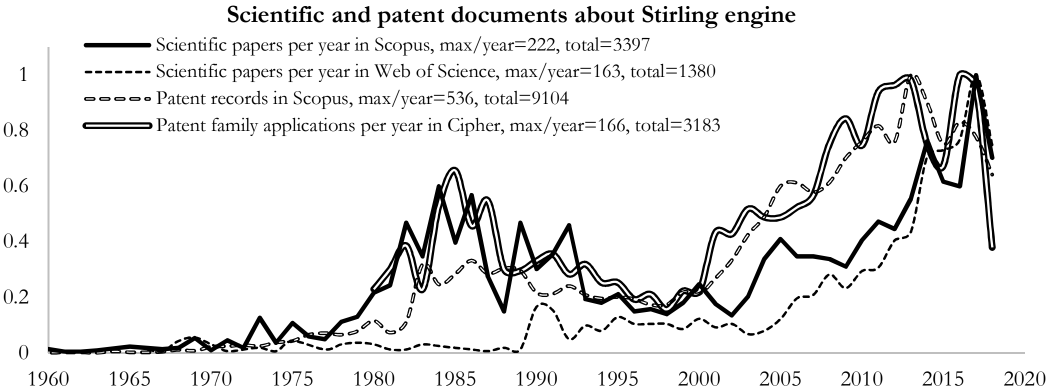


Fig. 1. Publishing activity for scientific papers and patent documents for the search term “Stirling engine” based on data from Scopus, Web of Science and Cipher for a period between 1960 and 2018. Absolute numbers are normalized by the maximum number of documents per year.

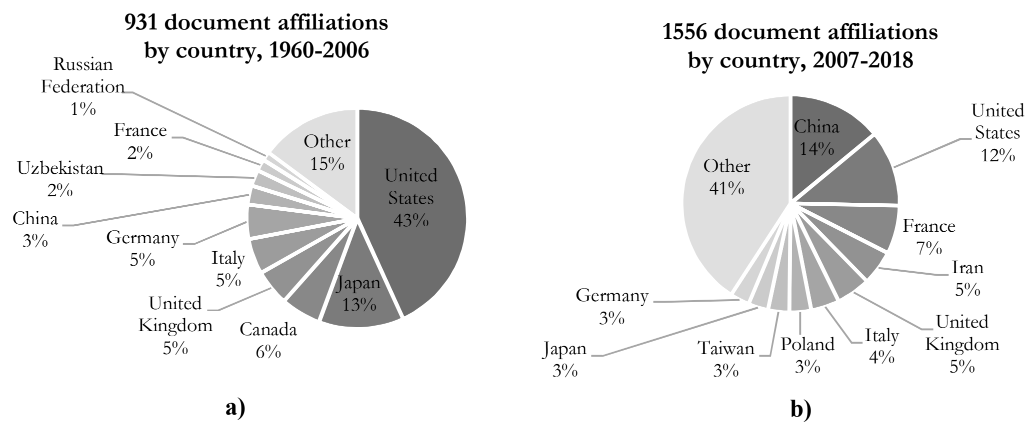


Fig. 2. Document affiliations by country in Scopus with identifiable countries of origin: a) 1960-2006, total 931 document affiliations; b) 2007-2018, total 1556 document affiliations.

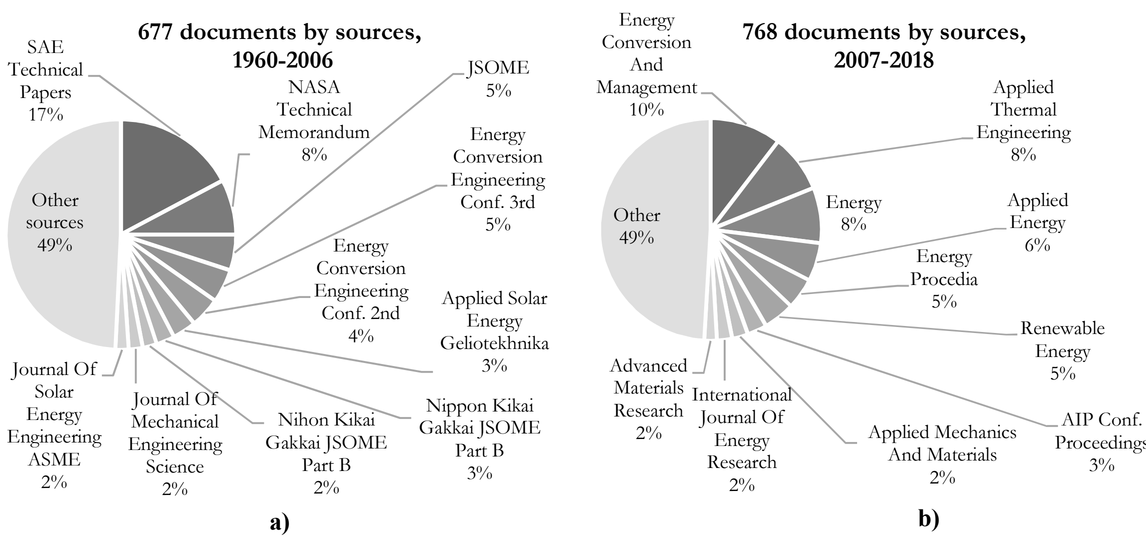
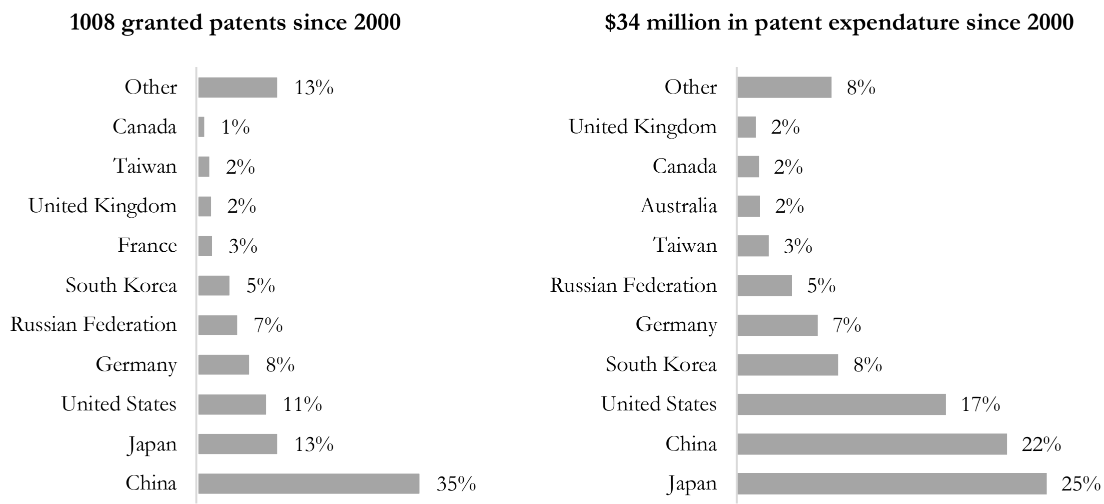
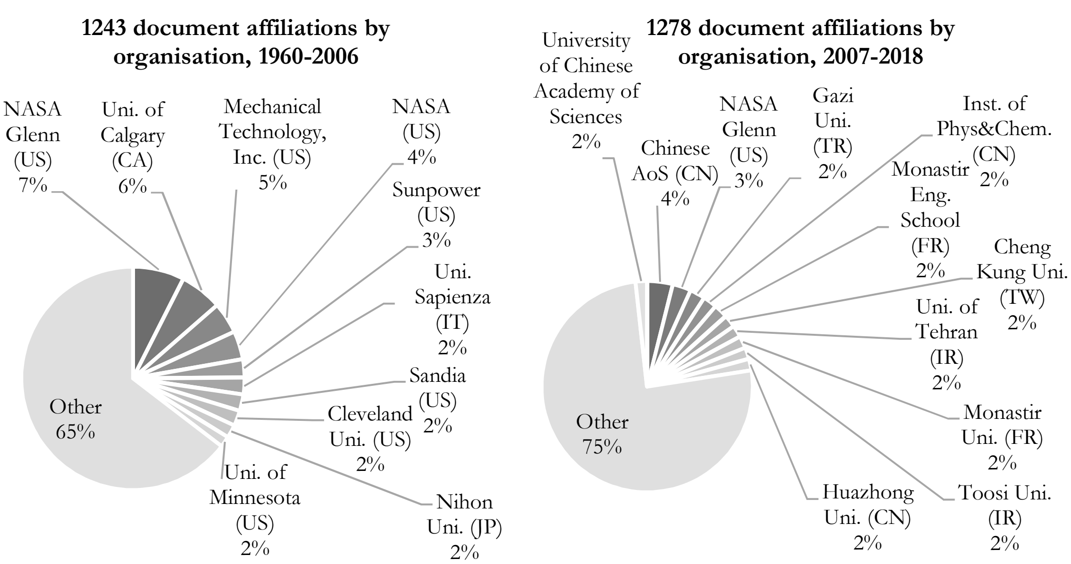


Fig. 3. Documents by source in Scopus with identifiable sources: a) 1960-2006, total 677 documents; b) 2007-2018 total 768 documents.



a) **b)**

Fig. 4. Geographic distribution data in Cipher in 2018, cummulative since 2000 for: a) 1008 granted patents and b) $34 million in patent expendature.



a) **b)**

Fig. 5. Document affiliations by organization in Scopus: a) 1960-2006, total 1243 document affiliations; b) 2007-2018, total 1278 document affiliations.

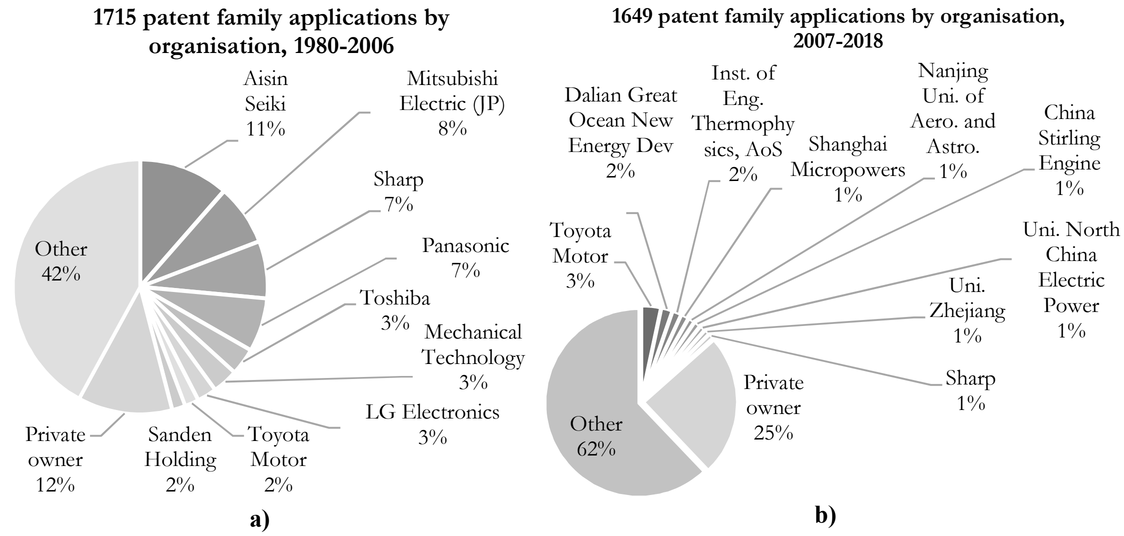


Fig. 6. Patent family applications by organization in Cipher: a) total 1715 families, 1980-2006; b) 2007-2018, total 1649 families.

In next three sections, I specifically analyze the scientific and patent activities in three top-publishing countries: United States, Japan and China.

III.1 United States

According to the Scopus database, during the period between 1960 and 2006, there were 402 scientific documents with identifiable country of origin about Stirling engines published in the United States. In the period between 2007 and 2018 the same number was 176 documents. Figure 7 shows the documents affiliations for the two periods. Between 1996 and 2006, the NASA Glenn Research Center specialized in various fundamental and applied engineering questions related to free-piston Stirling engines, including high-power (SPRE 12.5 kW engine) [9] and low-power (SRG110 55W engine) [10] free-piston Stirling engines. Modeling of Stirling engines was an important research activity [11]. The operation of actual engines allowed NASA Glenn to obtain experimental data and validate models for the feasibility studies of the engine under various space launch and outer space conditions. The recorded research activities between 2007 and 2016 indicate that the main theme of research studies was the development of a nuclear power plant based on the SE for deep-space

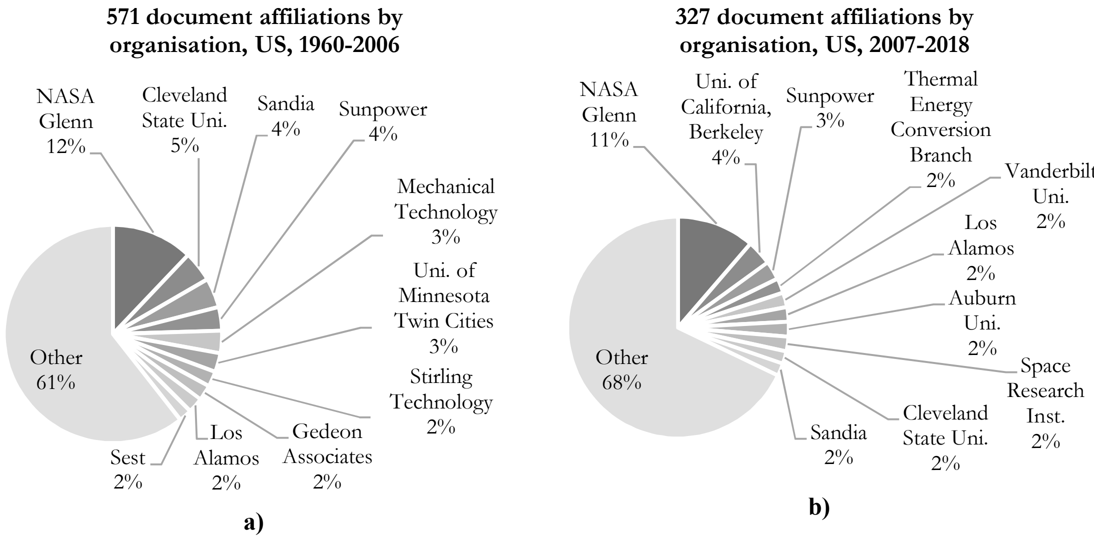


Fig. 7 Document affiliations in the United States by organization in Scopus: a) 1960-2006, total 571 document affiliations; b) 2007-2018, total 327 document affiliations.

exploration [12]. The research activities of the Cleveland State University conducted in collaboration with NASA Glenn Research Center covered the study of fundamental questions about regeneration matrix and the oscillating flow in SE [13]. The regeneration matrix is a specially arranged material that constitutes the regenerator. Oscillating flow refers to the operational condition of engines' working medium, which reciprocates during the operation. The university did not develop its own Stirling engine.

Research activities in Sandia National Labs show a gradual change from fundamental research to practical applications of SE-based solar systems over the period between 1989 and 2005 [14]. This was a result of the critical mass accumulation of the design knowledge and commercial development agreements with the US Department of Energy and Cummins Company. Although, the research did not focus on the improvements of the SE technology as such, it has developed innovative technologies in the solar dish-Stirling systems.

Before 2007, the activities of Sunpower Company (hereafter Sunpower) show the shift of the focus from relatively high power free-piston Stirling engines (3 kW) [15] to low-power Stirling engines (35W and 80W) [16] for military and space applications. After 2007, the recorded activities show that the organization has developed its technical capabilities to design free-piston SE engine technologies for a wide number of applications, including military and space. Sunpower has gone through the organizational modifications that would allow manufacturing the engines in larger series with a higher quality.

The activities of Mechanical Technology Inc. are recorded only until 1992 It reveals arguably one of the most complicated SE technology developments that resulted in the automotive Stirling engines of three consecutive generations – Mod I (peak power 53.9 kW, peak net efficiency 37.4%), Mod II (peak power 58 kW, peak net efficiency 38%) [17], and Mod III (completed design project for 115 kW power) [18] engines. The research activities were carried out under the funding of the US Department of Energy and NASA. The results of this development were not applied in the transportation sector due to cut funding.

Beginning from 2012, recorded research indicates that Los Alamos National Lab (LANL) has focused on the design and proved the operational feasibility of the heat-pipe cooled fast reactor together with the free-piston Stirling engine [19]. The recorded scientific investigation at LANL demonstrate the availability of operational Stirling engine nuclear reactor.

Auburn University was conducting research related to the SE technology in collaboration with Foster-Miller Inc. under the funding of the NASA Vision for Exploration program [20]. The program aimed to provide a nuclear reactor with a free-piston Stirling convertor at a power level of 30-40 kW to generate power on the Moon. It was a large governmental program introduced in 2004 by the President Bush that, apart from the Stirling engine, included other topics. The program was discontinued by the political decision of the Obama Administration in 2010.

Figure 8 shows the distribution of research topics in the sample of documents published by leading organizations in each corresponding time period. Figure 9 analyzes the ownership structure of active granted patent families in 2018 and depicts organizations that spurred the growth in granted patent families over the last decade. Toyota Motor has obtained a fully developed alpha-type crank-shaft engine, integrated into the thermal system of a transportation vehicle, with a sophisticated electronic control [1]. The recorded patents indicate that the research focus of Sunpower was on the control systems for free-piston engines [21] and on various inventions related to the linear alternator, sensor system, free-piston engine and the mechanism for minimizing

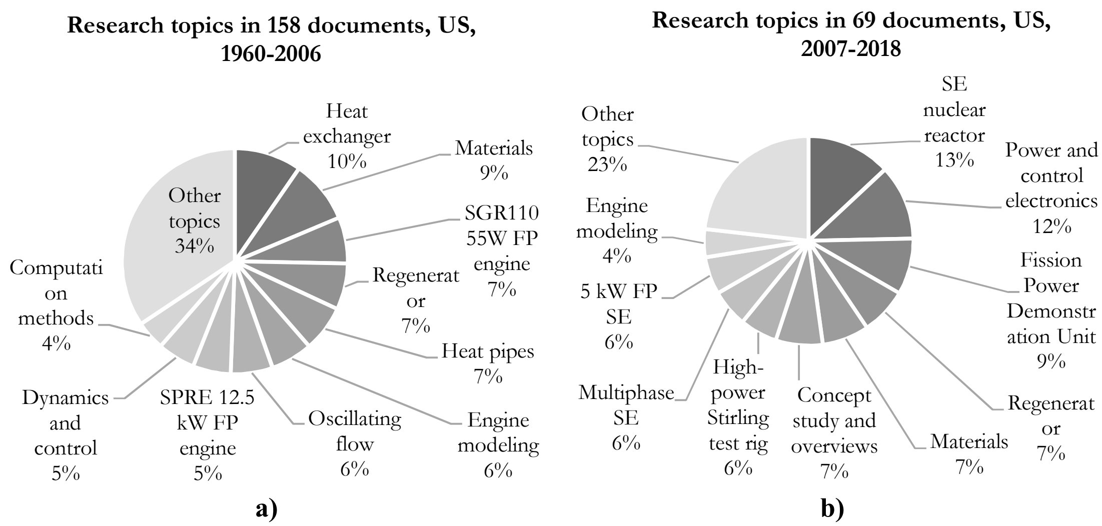


Fig. 8. Distribution of research topics in scientific documents by ten top-publishing organizations in the United States: a) 1966-2006, 158 analyzed documents from 402 total b) 2007-2018, 69 analyzed documents from 176 total.

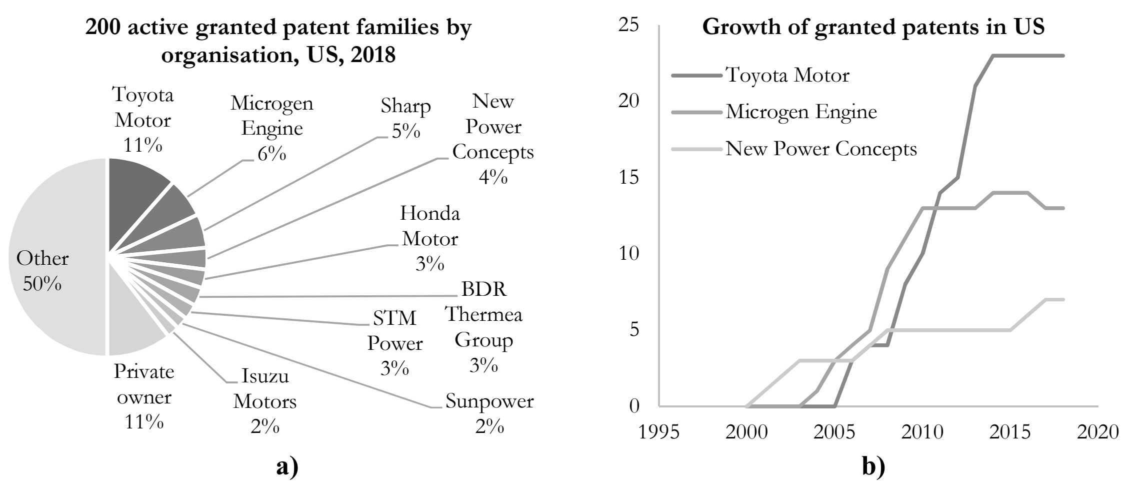


Fig. 9. Granted patent families in the United States: a) distribution of 200 active granted patent families by organization, active in 2018 b) Organizations that defined growth in granted patent families in the US over the last decade.

vibrations [22]. The analysis of patents by Microgen Engine Corp. indicates that most of the patents are not related to the design of engine as such, but rather to the integral design and operation of the gas-fueled combined heat and power units based on SE [23].

The research activities in the US over the last thirty years represent a coordinated by NASA and governmentally funded effort to design and develop a power system based on the free-piston Stirling engine for deep-space exploration.

III.2 JAPAN

According to Scopus, during the period between 1960 and 2006 there were 116 scientific documents with identifiable country of origin about Stirling engines published in Japan. It is 3.5 times less than in the United States over the same period. In the period between 2007 and 2018, there were 46 documents, 3.8 times less than in the US. Figure 10 shows the document affiliation for the two time periods.

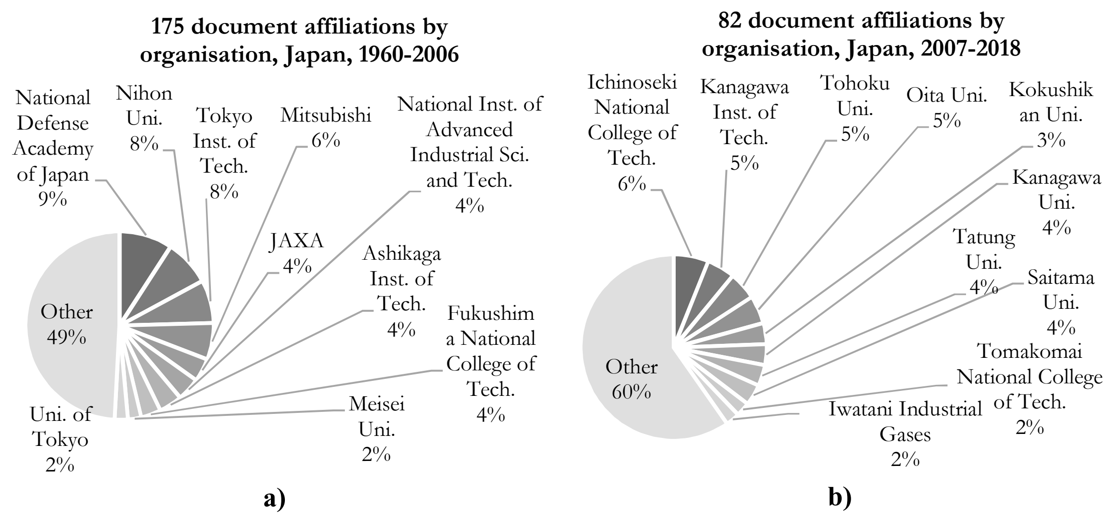


Fig. 10. Document affiliations in Japan by organization in Scopus: a) 1960-2006, total 175 document affiliations; b) 2007-2018, total 82 document affiliations.

The research activities at the National Defense Academy of Japan investigated different aspects of the engine operation, including regenerator configurations, noise and vibrations, combined work with vapor compression system for heat pumps, the modeling of thermodynamic and mechanical operation and of the cooling system [24]. The research activities of Nihon University, Tokyo Institute of Technology and Ashikaga Institute of Technology have included experimental evaluation of a number of technological options for Stirling engine with built and tested prototypes, such as solar-dish systems, regenerators, steam cycle Stirling engines, and mechanical drive configurations [25, 26]. Research activities at Japan Aerospace Exploration Agency (JAXA) related to the development of Stirling engines started in 1999 and have focused primarily on the solar concentrator free-piston SE for space applications [27]. Joint research activities at Nagoya University and Aichi University recorded in the Web of Science database are related to low-power output thermo-acoustic engines [28]. Research activities in Ichinoseki National College of Technology beginning in 2000s resulted in the design and development of a beta-type SE 3 kW in 2005 [29]. Further research activities focused on an integrated approach of investigating engine operation within a cogeneration system for different types of biofuels. Activities at Kanagawa University have mainly been related to the investigation of new configurations of SE regenerators [30]. Figure 11 shows the distribution of research topics in the sample of documents published by leading organizations in each corresponding time period.

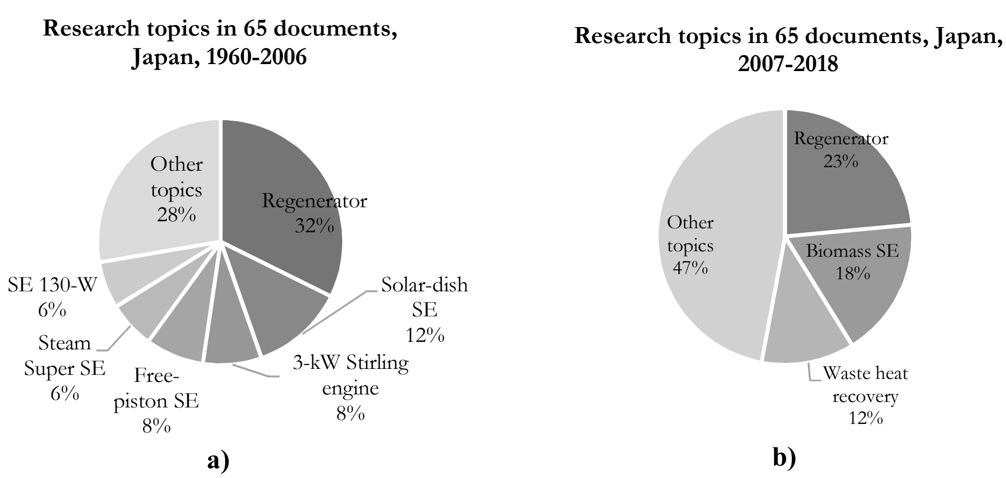


Fig. 11. Distribution of research topics in scientific documents by top publishing organizations in Japan: a) 1966-2006, 65 analyzed documents from 116 total b) 2007-2018, 17 analyzed documents from 46 total.

Figure 12 illustrates the ownership structure of active granted patent families in 2018 and indicates organizations that defined the growth in granted patent families over the last decade.

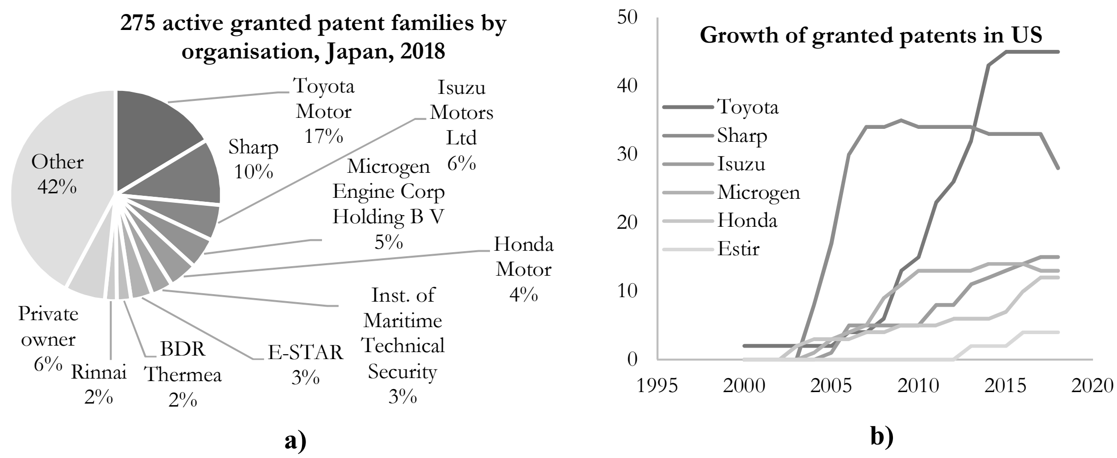


Fig. 12. Granted patent families in Japan: a) distribution of 275 active granted patent families by organization, active in 2018 b) Organizations that defined growth in granted patent families in Japan over the last decade.

Five commercial organizations control 42% of granted patent families in Japan with Toyota accumulating the biggest patent portfolio over the last ten years (Fig. 12). A considerable share of granted patent families (27%) is owned by companies that manufacture equipment for the transportation sector. This observation implies that, apart from space applications, another promising applications for the SE technology is the transportation sector.

Toyota Motor and Microgen Engine have a similar patent portfolio as in the United States. Sharp has obtained a series of patent families protecting different structural and thermal aspects of a proprietary free-piston Stirling engine [31]. Patent activity of Isuzu Motor indicates the development of different engine configurations before 2008, such as SE with the magnetic drive mechanism, the kinematic drive mechanism, and the miniature SE for computer circuits cooling. After 2008, the organization has selected a free-piston engine configuration with the application to vehicles for the waste heat utilization [4]. Honda quickly obtained the expertise in the Stirling engine domain by introducing beta-type kinematic designs of the engine [32] for different applications, including a power unit for prosthetics drive [33] and a waste heat recovery for vehicles [34]. The patent activity of Estir Co. indicates a consistent development of the beta-type crank-shaft Stirling engine by Estir Co. Ltd. and the subsequent development of the waste heat utilization system for ships [3].

Although the total amount of research publications is approximately 3.4 times less than in the US over the same period, the variety of topics and applications related to Stirling engines is more complex and less coordinated by the space exploration agency. Also, Japanese companies show much stronger patenting activity than in the US. This observation indicates that Japan has a rich variety of design configurations for different application fields.

III.3 China

Between 1960 and 2006, there were 26 scientific documents with identifiable country of origin about Stirling engines published in, which is 15.5 times less than in the United States and 4.5 – than in Japan. In the period between 2007 and 2018, there were 218 published documents, or 1.2 times more than in the US and 4.7 – than in Japan, indicating an incredible growth in scientific activities on Stirling engines in China over the last 12 years. Figure 13 depicts the publishing trends of scientific and patent documents over time in China. Figure 14 shows the document affiliation for the two analyzed periods. Research activities at Naval Academy of Engineering focused on the theoretical analysis of quantum Stirling engines and their application for the refrigeration systems [35]. This direction was interesting because traditional models, even those that accounted for finite-time constraints, had not studied the working fluid as a quantum fluid, like He, Ne, D_2, with the operation of the engine abided by laws of quantum mechanics. Chinese Academy of Sciences performed design modeling and did early experimental work for thermoacoustic Stirling engines [36] that it continued to develop after 2006 [37] along with solar heat receivers and heat exchangers [38].

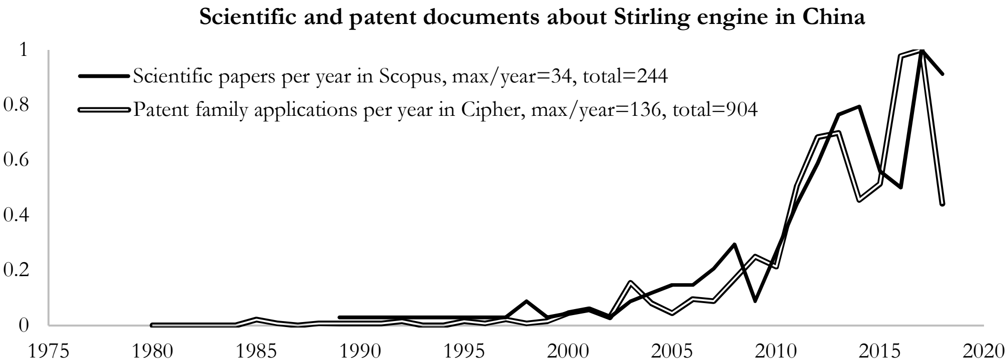
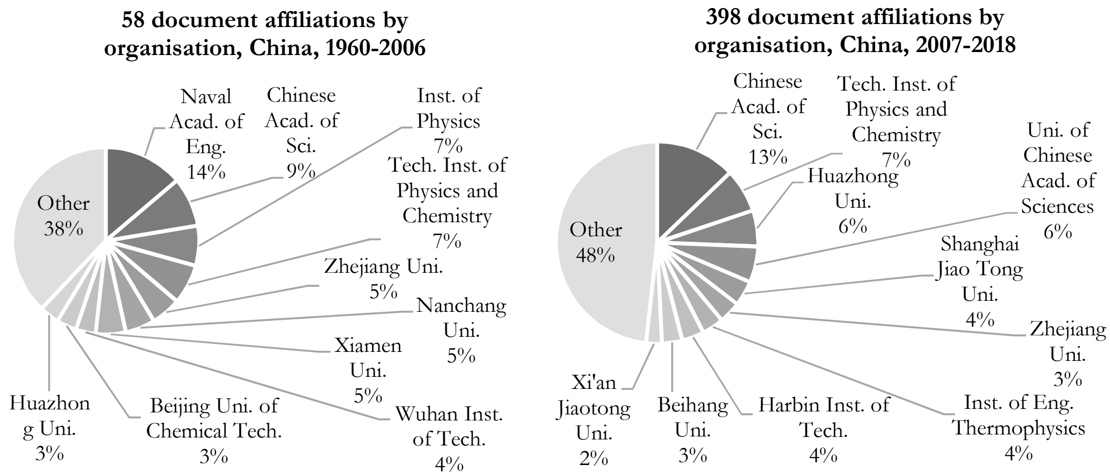


Fig. 13. Publishing activity for scientific papers and patent documents for the search term “Stirling engine” based on data from Scopus and Cipher for a period between 1980 and 2018 in China. Absolute numbers are normalized by the maximum number of documents per year.



a) **b)**

Fig. 14. Document affiliations in China by organization in Scopus: a) 1960-2006, total 58 document affiliations; b) 2007-2018, total 398 document affiliations.

The research activities at the Shanghai Jiaotong University were primarily oriented towards experimental studies of combustion processes for Stirling engine combustion chambers [39]. Huazhong Institute of Science and Technology covered various subject areas, including thermoacoustic engines, moving magnet compressors, MEMS Stirling engines, thermodynamic optimization, and combustion chamber, and developed a 0.3 kW beta-type rhombic drive Stirling engine [40]. Figure 15 shows the distribution of research topics in the sample of documents published by leading organizations in each corresponding time period.

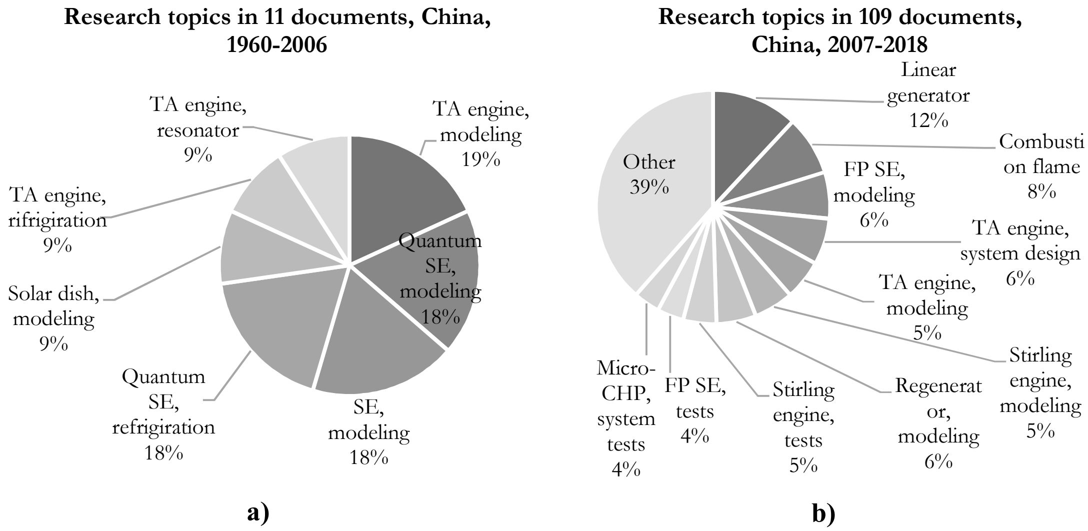


Fig. 15. Distribution of research topics in scientific documents by top publishing organizations in China: a) 1966-2006, 11 analyzed documents from 26 total b) 2007-2018, 109 analyzed documents from 218 total.

Figure 15a shows that before 2007 Chinese research activities focused on different aspects of thermoacoustic (37%) and quantum (36%) Stirling engines. The research was conducted by a small number of organizations and was fledgling in nature. After 2006 (Fig. 15b), the research records demonstrates an incredible increase of research works and research organizations. The focus on the research of thermoacoustic engines remained (11%), however a significant number of publications addressed the linear generator (12%), combustion flame (8%), free-piston Stirling engine tests and modeling (10%) and similar work related to other engine configurations (10%). Figure 16 analyzes the ownership structure of active granted patent families in 2018 and indicates organizations that defined the growth in granted patent families in China over the last decade.

The patenting activity of Dalian Honghai New Energy Dev. Co. Ltd was related to the off-grid solar-dish stirling engine energy unit and covers different structural, design, and control aspects of the system [41]. Institute of Engineering Thermophysics was focused on various elements and portable applications of a free-piston Stirling engine [42]. Shanghai Micropowers focused on improvements of a Stirling dish concentrating power unit with patents

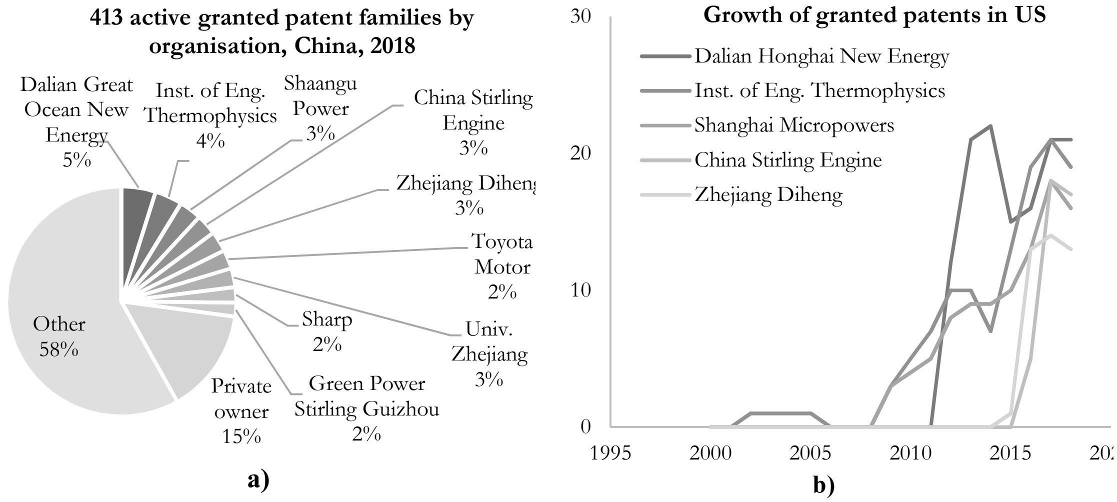


Fig. 16. Granted patent families in Japan: a) distribution of 413 active granted patent families by organization, active in 2018 b) Organizations that defined growth in granted patent families in Japan over the last decade.

ranging from local improvements of heat exchangers, to oil systems, to control algorithms and systems [43]. China Stirling Engine focus primarily on patenting inventions about the applications of a free-piston Stirling engine as an engine and as a refrigeration device, for example solar concentrator [44] of refrigeration box [45]. The analyzed patents indicate that these organizations do not own a proprietary design of a Stirling engine and seem to license it from other organizations, and most of the documents are utility models and.

China has shown incredible growth in the number of scientific research in the last 12 years. This growth has not yet resulted in diversified research activities. The focus was made on fundamental analysis of thermoacoustic engines and on patenting activities about the integration of the free-piston Stirling engine in solar systems.

IV Discussion

Figure 1 shows two distinct peaks in scientific publications for Scopus data. The first peak was during the 1980s. The second peak had started growing around 2002 and was at the top in 2018. The trend for patent applications was peaking during the 1980s similarly to that year research work; however, the second wave started in 2000 and was five years ahead the scientific publishing until around 2015 reducing to one year gap in 2018. These two peaks both in scientific and patent literature must represent two noticeable phases in the development of knowledge for Stirling engines. It is interesting to understand their reasons and implications for the choice of alternative design configurations.

IV.1 Phase I, 1980-1990

IV.1.1 Scientific literature

Figure 1 shows that Web of Science does not track the publishing peak during Phase I, indicating that the peak consisted from documents from unknown publishing sources that were not referenced among reputable journals. In addition, Scopus references 895 documents for Phase I, out of which 794 documents have undefined country origin. This provides a similar indication about the quality of published research. Only 259 (29%) of those documents were published in peer-reviewed journals, books or editorials. The rest were conference papers. Examples of these conference proceedings could be seen in Fig. 3a. Therefore, Phase I represent a period of conference discussions, generating ideas, and perhaps finding collaborators. This period may be described, informally, as being characterized by “exploratory technological chatter”. This is a fledgling phase of knowledge about Stirling engines mostly without systematic studies of the technology.

A reader may ask, why Phase I occurred during the 1980s and not some other time? A likely explanation could be found below in Figure 17. Increased scientific interest could result from crude oil price surge in early 1980s. These observations indicate that the published results during 1980s were strongly influenced by oil prices and the search for alternative prime mover technology. This hypothesis is also confirmed by the research affiliated with NASA Glenn Research Center during the 1980s related to Stirling engine for automotive application [46], [47] and Mechanical Technologies that developed several configurations of the automotive Stirling engine in collaboration with the US Department of Energy (Section III.1.1). Along with the oil crisis, there could be environmental concerns that contributed to the increase of scientific inquiry. A four year lag in scientific publications could be explained by the time needed to solicit research funding, conduct studies, and undergo through a publication review process. However, given that the majority of published scientific work at the time was conference papers, a four year lag seems a long time to react on surging oil prices.

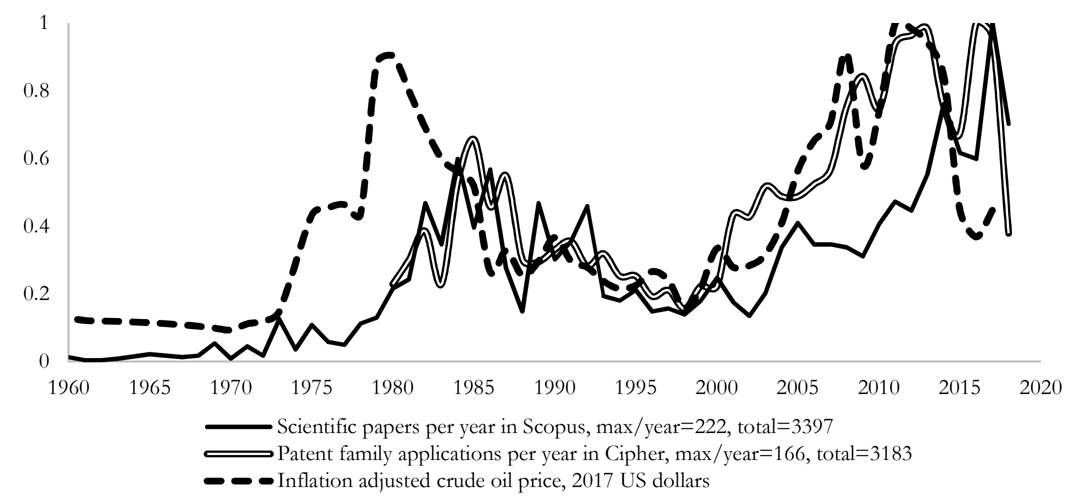


Fig. 17. Scientific (Scopus) and patent (Cipher) publishing and the crude oil price [48], normalized by the corresponding maximum annual value.

IV.1.2 PATENT APPLICATIONS

Patent applications grew and declined in parallel with scientific work during Phase I (Fig. 1). This could mean that research activities during Phase I fed inventing work with ideas and were mutually beneficial to each other. However, the obtained results do not affirm this conclusion. Figure 5a and 6a show that the organizations that conducted research and filed patent applications were different during the period between 1960 and 2006. In addition, these organizations were located in different geographical regions – the United States for scientific activities and Japan for patenting. Therefore a mutually beneficial influence between science and engineering was unlikely to exist at the time and the domains were developing independently from each other.

To better understand who conducted inventing and patenting activities during Phase I, one can look at the structure of organizations that filed patents during this phase (Fig. 18). Approximately 60% of inventing activities were performed by three companies: Panasonic, Mitsubishi Electronics and Aisin Seiki. All three organizations are Japanese. This shows that the engineering knowledge about Stirling engines was centralized geographically among three companies. This is also confirmed by a relatively small portion of other organizations that filed patents during Phase I and a very small portion of private owners that understandably did not have access to public knowledge at that time to make inventions.

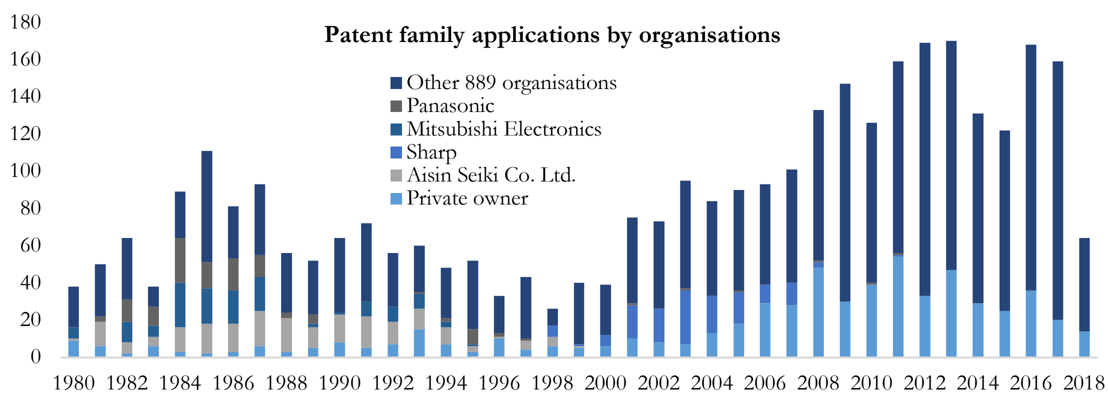


Fig. 18. Patent family applications by organizations in Cipher

IV.2 PHASE II, 2000-2018

IV.2.1 SCIENTIFIC LITERATURE

Phase II is a systematic verification of principles embedded in Stirling engines consisting of 1867 scientific documents. Scopus data shows that 1069 (57.3%) documents during Phase II were published in peer-reviewed journals, books or editorials – a striking difference from Phase I with 29% or 259 peer-reviewed papers. Figure 3b confirms this conclusion and depicts examples of publishing journals. In addition, the content of scientific research for examples of the United States (Fig. 8a) and Japan (Fig. 11a) indicate that Phase II was primarily concerned with application studies of engine indicating deep and systematic understanding of the core technology.

Phase II indicates that more organizations from different countries participated in research activities. For example, Fig. 2a shows that before 2007, the share of other countries in the document affiliation was 15% and during the last twelve years this portion grew to 41% (Fig. 2b). The share of other organizations in document affiliation grew from 65% to 75% (Fig. 5). These findings show that during Phase II more countries and organizations engaged in scientific activities about Stirling engines that also confirms the maturing knowledge about the technology. Otherwise, other countries would not have access to publications and would not be able to quickly achieve a noticeable level of research progress. China represents a good example of quick development of research activities in this domain from 26 documents before 2007 to 218 documents in 2018 (Section III.1.3) with the content focus on fundamental and to a much lesser extent on applications questions.

IV.2.2 Patent applications

Phase II represents a broad scope and large volume of inventing activities by a significant number of organizations and private owners in different geographies with now available access to public knowledge about Stirling engines. Figure 6 shows that the share of other organizations in filed patent family applications grew from 42% to 62% and the share of private owners – from 12% to 25%. This is also noticeable in Figure 18. China, as a prominent example of growth of patent applications (Fig. 14) and granted patents (Fig. 4a) during Phase II, may give a clue about the technology content of inventions. Most analyzed patent documents are utility models and do not protect a proprietary design of Stirling engines, but rather focus on the design of application systems where the engine is a core integral component. This observation indicates that the growth of patenting activities during Phase II is mostly concerned with Stirling engine applications. This is also confirmed by the patenting of Microgen Engine (see III.1.1) and later patents of Sunpower (III.1.1) and Estir Co. Ltd. (see III.1.2). Most companies licensed proprietary engine designs from other companies with proprietary designs, for example from Sunpower. On the other hand, Japanese companies including Toyota, Sharp, Isuzu and Honda have obtained patents for proprietary designs of Stirling engines even during Phase II, showing that this period did not only include inventions for applications, but also original engine designs. It is important to notice here, that Japan historically was the most advanced region in Stirling engine development, including Phase I. We can conclude that although Phase II was mainly about application inventions, knowledge intensive geographical regions kept designing original Stirling engines.

Another interesting observation is that patent applications are approximately 5 years ahead of scientific publishing during the major part of Phase II (Fig. 1). This is a fascinating result, because it means that patenting was developing independently from scientific research and could even be a cause for research activities about Stirling engines and not *vice versa* as it is commonly thought.

IV.3 Phase I and II for selecting design alternative configurations

What can the results of Phase I and II teach us about selecting design alternative configurations for Stirling engines? A first conclusion that has solid proofs in both phases is that Japan is a number one geographic region in terms of proprietary designs for Stirling engines. The country defined Phase I in terms of invention activities and was the main contributor of original engine designs during Phase II. A second conclusion is that Phase I is a good source of knowledge about fundamental questions, especially research conducted in the US, and Phase II – about application designs of Stirling engines. This is confirmed by the content of both scientific and patent documents. However, countries that started growing the knowledge capacity about Stirling engines during Phase II, like China, have a significant number of fundamental results obtained during Phase II that may be compared with results of earlier research during Phase I.

IV. 4 APPLICABILITY OF SCIENTIFIC RESEARCH AND INVENTIONS

Although 39% of all patents to date have been granted in China (Fig. 5a), only 22% of patent expenses were made in this Asian country (Fig. 5a), which in addition has had a relatively young contribution to the scientific knowledge accumulated in academic publications (Fig. 13) noticeable only during the last twelve years. As mentioned previously, in-depth analysis of scientific research and patent publications shows that the country is only gaining capacity in systematic understanding of Stirling engines (Fig. 15) and is protecting inventions primarily in the design of application systems based on Stirling engines using utility model patents. These facts show that China is a territory with a very competitive patent landscape, where patents rather have a symbolic value with a fledgling scientific background than are well-researched high-quality protection. The opposite situation can be observed in Japan, which has a share of 25% in patent expenditure, approximately the same amount as China, but has had 2.7 times less granted patents. Japan demonstrates a more mature research background with patents considered as commercial investments. The United States, on the other hand, has a very strong research background (Fig 2). Yet, for such extensive research work, the country obtained only 11% share in granted patents. This implies that the United States organizations are more focused on publicly funded research for specific applications of Stirling engines than their commercialization. This is also confirmed by the US research activities during the last twelve years about system integration of nuclear fission power systems for space exploration. The difference between Japan and United States is that the former focuses on commercializing Stirling engines primarily for the transportation sector and latter focuses on governmentally funded long-term aerospace or military initiatives. Therefore, the design configurations selected by Japanese organizations should better reflect the requirements for commercial integration of Stirling engines, while the design configurations selected by the United States organizations are better for the specific requirements of space explorations and military activities.

IV. 5 Influence of publicly available knowledge

We have established that the US has a very prominent role in the generation of public knowledge about Stirling engines. One consequence of this is high influence of the US research activities on other countries and organizations. These countries and organizations have significantly increased their share in scientific and inventive activities (Fig.5 and 6). The question is what knowledge base were these organizations using as reference for their activities? Most likely it was research published by the US organizations. The research on free-piston engines is a landmark of the US space oriented organizations (free-piston 12.5 kW, 55 W Stirling engines, dynamics and control). This research was primarily focused on Stirling engines designed for space and military applications. This is confirmed by Figure 5a, which shows that before 2007, at least 24% of global research in Stirling engines was performed by organizations related to space exploration (NASA and Sunpower) or in partnerships with them (University of Calgary and Cleveland University). The hypothesis that this knowledge could significantly influence other countries is confirmed by the fact that the share of free-piston related research (including linear generators) by Chinese organizations increased from 0% to 22% over the last twelve years. In addition, a detailed analysis of Chinese patent literature indicates that many patent documents use free-piston Stirling engine as an integral component for inventions. In contrast, Japanese paten activities show that free-piston engine configuration is not a leading engine design, and apart from Isuzu, Japanese companies selected engine designs with a kinematic drive, indicating that this design is better suited for on-Earth commercial applications. Therefore, without careful selection, organizations that implemented the available knowledge as a reference for their activities may have integrated design choices and used research data that is less applicable for on-Earth commercial requirements.

IV.6 Scientific and engineering domains for Stirling engines

It is natural to expect that some results of scientific activities would be transferred to commercial organizations and implemented in commercial systems. There are different channels of this transfer. The scientists may be hired by companies. Technical employees in the companies may read the scientific publications. The universities may license the research to companies in the form of patents. It is difficult to use the available data to evaluate the first two channels of knowledge transfer, but it is possible to comment on the technology licensing. From Figures 5 and 6 we can see that apart from one organization (Sunpower), the list of top patenting organizations does not include organizations from the list of leading scientific organization. This observation shows that universities and research laboratories do not transfer its knowledge to companies through patents, otherwise we would see some of scientific organizations in the list of top patent owners. The existence of this technology transfer channel is nevertheless confirmed by Sunpower that is present in both scientific and patenting organizations and was known for licensing its technologies to organizations in US, Europe, Japan and China. However, this is only one example; and therefore, is more an exception than a rule. These findings also indicate that companies develop and patent original technology, otherwise they would not be able to obtain patents on knowledge that had been previously disclosed in scientific publishing. In addition, as discussed previously, Figure 1 shows that patenting was 5 years ahead of scientific research during the major part of the last twelve years showing its independence. These findings show that knowledge bases of scientific publishing and patents are two distinct domains of technical knowledge about Stirling engines. When analyzing design alternatives for Stirling engines, one should explore both domains to find original design solutions.

IV.7 THE EXAMPLE OF CHINA: SHOULD WE CARE, OR IT IS EASIER TO DISREGARD?

The rapid growth of scientific activities in China (Fig. 13) could indicate a new discovery in the field. It could also be attributed to the growth of unoriginal research and utility model patents, which are not examined until litigated. It could be also the combination of both, in which case there is a need to pay attention because there could be important discovery hidden beyond excessive generation of papers and petty patents. The available data suggest the third kind of situation. The strong publication growth started not until 2010, the time where a significant body of knowledge has been already generated in papers and patents by United States and Japan (Fig. 2). It would be easier to use this research and patents to produce research and utility models in China, indicating towards the first scenario. This is confirmed by the structure of the main publication topics after 2007 (Fig. 15b) with somewhat less original research on free-piston engines and linear generators that most likely were borrowed from United States and Japan. In depth analysis of the patent records of leading patenting Chinese organizations also suggests a dominating presences of utility model patents. On the other hand, the structure of the main publication topics during the period before 2007 (Fig. 15a) shows original research topics comparing to research focus in United States and Japan, and some of this research topics, for example themoacousting engines, were carried after 2007 (Fig. 15b) indicating original results and confirming the first scenario. These observations show that research and patent activities in China has had original work but also mimicked to some degree knowledge generated in United States and Japan. This differentiation of original and mimicked research and patents helps to identify unique design configurations.

IV.8 Final remarks on selecting design alternatives

From this discussion, it is obvious that in order to identify and compare different design configurations for the development of commercial Stirling engines, the selective literature review may not be sufficient. To identify original and well tested design alternatives, it is important to: know the development phases of technology to differentiate between fundamental and applied results; understand the intended applications of different organizations; minimize the influence of design alternatives that are common in literature; be aware in what technical domain – scientific or engineering – the original design alternatives could be found; and be capable to disentangle the plethora of scientific and patent documents produced by different countries.

One interesting technical observation that can be derived from analysis of scientific and patent literature example is the proliferation of design alternatives for the kinematic configurations of the designed engines. These configurations can be broadly classified in three types – alpha, beta, gamma. These names are given to different arrangements of working spaces inside the engine that significantly affect the mechanical realization of the prime mover (Fig. 19).

For the alpha-type engine, the compression and expansion spaces are located in two different cylinders, which are connected by a passage with the regenerator. For the beta-type engine, the compression and expansion spaces are located in the same cylinder at the opposite ends, which are separated by a displacer piston. The compression space at one end of the cylinder is connected with the expansion space at the other end of the cylinder through an

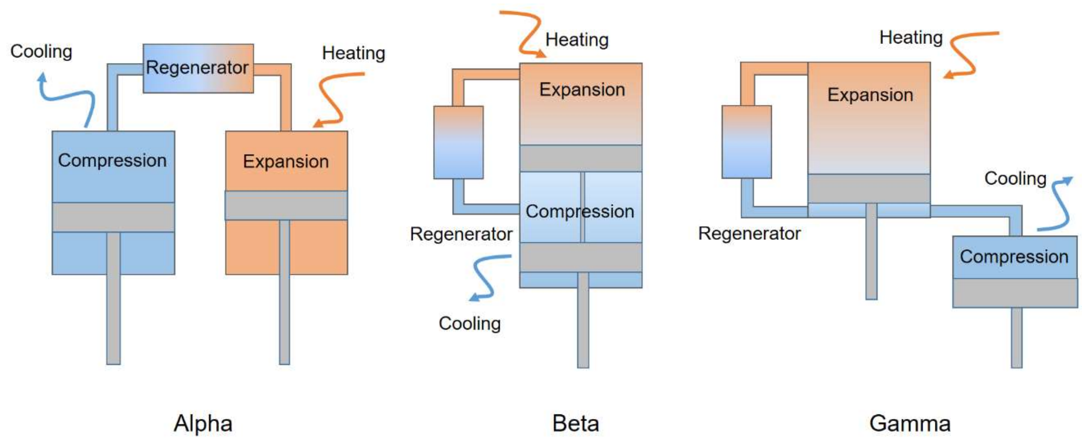


Fig. 19. Schematic views of different kinematic configurations of Stirling engines: alpha, beta and gamma types.

external passage with regenerator material. For the gamma-type engine, the compression is organized in a separate cylinder. The expansion cylinder is connected with the compression cylinder by an external passage. The regeneration is organized similar to the beta-type configuration. Apart from these architectural configurations, each engine design may have different volume of working spaces defined by bore and stroke, and varying temperatures of heating and cooling.

Research and commercial organizations employed different configurations depending on requirements to power, efficiency, available space, vibration tolerance and end-user application. However, surprisingly enough, literature does not offer a convincing performance comparison for these configurations. Therefore, the system level analysis can be performed to mathematically model and compare the performance of these three configurations. Typical system-level performance characteristics identified in literature for Stirling engines is power, efficiency and cost. Basic design variables include hot and cold temperatures, and the engine volume, which can be defined by bore and stroke. These four design parameters can be used for controlled comparison of the objective configurations. The next section discuss in details the design method developed to compare alternative configurations for ECT and offers additional insights in a preferable engine configuration.

IV.9 DOWNSIZING THE SPACE OF DESIGN ALTERNATIVES

This chapter was concerned with the concept design stage. It is now essential to discuss how to evaluate and downsize design alternatives identified and selected during the concept stage. For the demonstration example, I selected alternatives for kinematic configurations – alpha, beta and gamma-type discussed in the previous section. This procedure is carried out typically at the system-level design stage. The proposed method to solve this design problem is called tradespace exploration for Stirling engines [49], which was developed during my MSc studies. Therefore, this design method is not presented in a separate chapter as a novel contribution to this Ph.D. thesis. However, the application of this method as an integral part of the novel technological pathway is new. As such, I believe it is vital to comment in more detail on the manner of this integration into the proposed design methodology.

The process of generating concept designs is essential as it provides several design configurations of the energy conversion technology (ECT). For the example of the Stirling engine, these alternative configurations include three engine architectures: alpha, beta, and gamma. One of critical notions of this design method is the subscription to a philosophical idea that it is crucial to enumerate all possible design solutions from the onset, before focusing on single-point design. This idea is different from a typical design process by expert designers [50], who would typically narrow the design space from the beginning of the design process because of their knowledge and design experience. The enumeration of all possible concepts from the onset helps to explore different design alternatives, yet it requires a design tool that allows downsizing these alternatives based on selected system performance metrics, figures of system merit. The proposed design method was developed for the Stirling engine by adapting the tradespace concept from the field of systems engineering [51]. This method includes the design of the system model that integrates several sub-models. First sub-model describes engine internal technical relations between the design parameters (engine bore, stroke, heating, and cooling temperature) and the system metrics (power and heat efficiency). These relations are based on analytical equations derived from experimental results in literature; they are different for an alpha, beta, and gamma configurations. The second sub-model describes the interaction of the engine with external heat baths (hot and cold) with the help of general heat transfer relations. The third sub-model is the economic model that builds a relationship between input design parameters and engine equipment cost. The integration of these three sub-models with an input a set of design parameters helps to evaluate power, efficiency, and cost of the engine for an alpha, beta, and gamma configurations. Figure 20 shows the system model, and Figures 21, 22 and 23 show the results of this method.

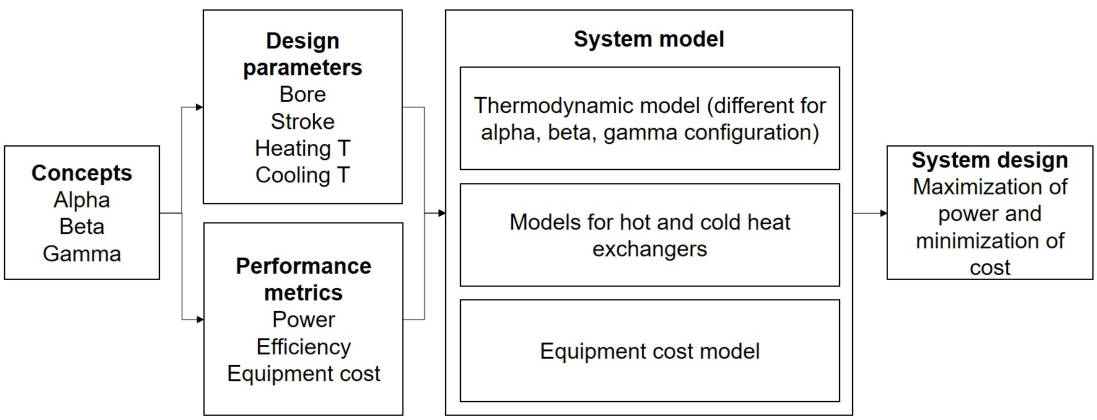


Fig. 20. The sequence of steps to evaluate conceptual designs using tradespace exploration. Adapted from [49].

In Fig. 21, each color represents a particular type of engine architecture (alpha, beta, and gamma, Fig. 19), and each point of similar color is a different design based on varying input design parameters.

The range of design parameters was the same for three architecture types; therefore, we consider that the comparison of design architectures was carried out under similar conditions. The objective of the design process could be to maximize power, efficiency, or minimize the engine cost. For the Cost-Power trade-off (Fig. 21, a), for the same cost, the alpha-configuration is preferable because it maximizes power output. For the Cost-Efficiency trade-off (Fig. 21, b),

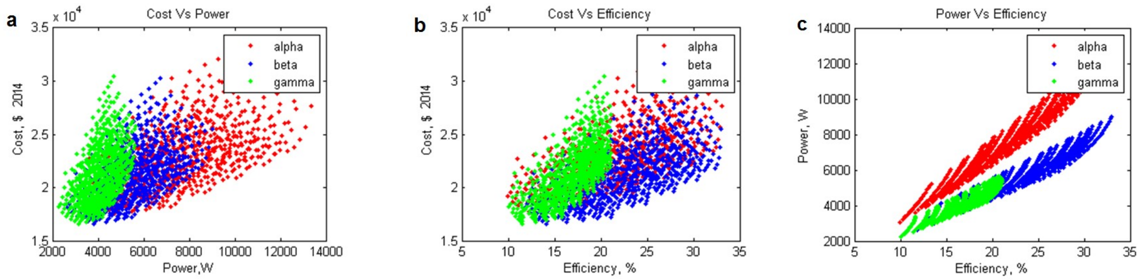


Fig. 21. Engine architecture tradespace, (a) Cost-Power trade-off; (b) Cost-Efficiency trade-off; (c) Power-Efficiency trade-off. Reprinted from [49].

for the same cost, the beta-configuration is preferable because it maximizes the engine efficiency. For the Power-Efficiency trade-off (Fig. 21, c) for the same efficiency, the alpha-configuration is preferable because it maximizes power output, yet it will be more expensive than the beta-type alternative. One can analyze the effect of design parameter values – bore, stroke, hot temperature, and cold temperature – on the system metrics. For this analysis, I selected the alpha-configuration because it offers a higher power output at a similar cost and similar efficiency comparing to other types.

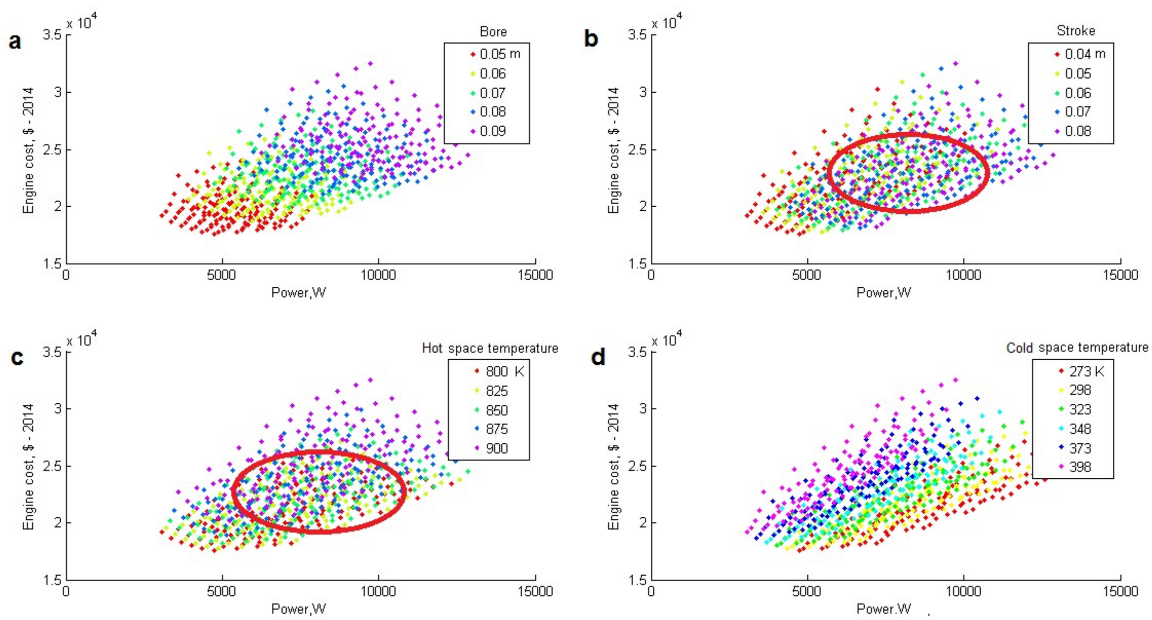


Fig.22. Architectures of the alpha-type engine colored by different design parameters: (a) bore, (b) stroke, (c) hot space temperature, (d) cold space temperature. Reprinted from [49].

To minimize the cost of the engine and maximize power, Fig. 22a demonstrates that most preferable bore values are between 0.06 and 0.08 m. Lower bore values than 0.06 m offer lower cost due to reduced sizing of bulk engine design, but also lower power because the working volumes are also reduced. Conversely, higher bore values than 0.08 m maximize the power but result in a higher engine cost. The effect of the stroke value (Fig. 22b) is more complex than for the borehole diameter. In general, higher stroke values are preferable to maximize power. With the increase of stroke, the working volume increases. However, the area highlighted with the red line shows the design space where different stroke values from 0.04 to 0.08 m could offer more design alternatives with similar power and cost characteristics. In other words, a system designer has more alternative design options, when he selects the engine stroke. Similar conclusions can be made for the hot space temperature (Fig. 22c). The increase in temperature leads to more expensive heat exchanger materials and design. However, the highlighted area allows selecting temperatures from 800 K to 900 K with similar cost and power values. The maximization of power through the change of cold space temperature (Fig. 22d) shows that the lower the temperature, the higher the power and lower the engine cost. At first glance, this result seems surprising. Designing a heat exchanger that can maintain room temperatures (293K) or even lower temperatures (273K) should be more expensive than for higher temperatures like for example for 348 K. However, the increase of temperature of the cold space proportionally increases the amount of rejected heat during the isothermal compression in the Stirling engine. A higher amount of rejected heat implies larger heat exchange areas, which results in more substantial engine costs. Figure 23 helps to understand better the effect of design parameters on the alpha-type engine cost. Bore and stroke demonstrate a linear effect, and temperatures – an exponential effect.

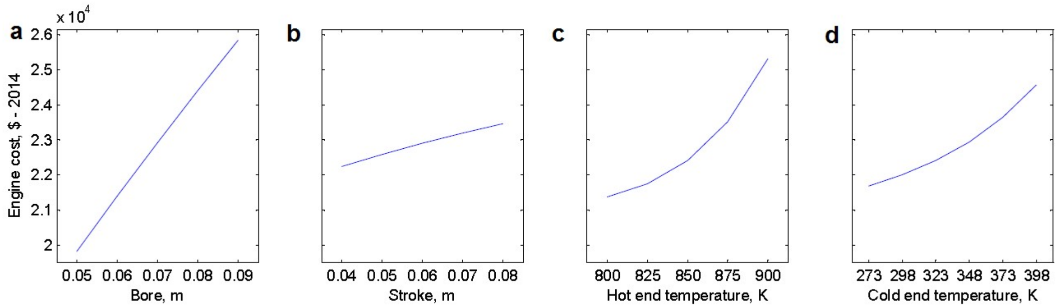


Fig. 23. Main effects of (a) bore; (b) stroke; (c) hot temperature, (d) cold temperature (from left to right) on the alpha-type engine cost. Reprinted from [49].

During the development of the real Stirling machine, our team selected the alpha configuration as an alternative with high potential for scaling power and expanding the technology for different low and high power commercial applications. One of the drawbacks of this decision is lower efficiency of the alpha-type, as shown in Figure 21b. This diminishing of efficiency can be explained by the arrangement of working volumes: the beta-type has more efficient usage of space, when compression and expansion volumes are connected in the same cylinder; and the alpha-type has bigger dead volume and lower volumetric efficiency. Nevertheless, in a trade-off between efficiency and power, a decision to select higher power was made since it offered more market alternative to scale the system. Alpha configuration was one of system-level requirements for the detail design stage.

A similar design methodology can be applied to other ECT to downsize the space of design concepts. The boxes in Figure 20 help to describe this sequence of actions. First, there should be introduced some categorization of design concepts. Second, there should be selected a set of system-level conflicting design parameters that would cause system metrics (in this case, power, efficiency, cost) to change in a conflicting manner. These input design parameters are integrated into a system model with physical relations and cost equations to evaluate different design concepts and their equipment costs. The resulting space of system configurations is downsized using market requirements and visual results similar to those in Figure 20.

IV.10 Novelty and Contribution

- Developed a method of analysis for scientific and patent literature with the use of big data technologies and bibliometric information;
- Identified a historically accurate account of scientific and patenting activities for Stirling engine over the last 58 years;
- Identified statistically significant distribution of research topics about Stirling engines;
- Developed a theoretical framework that defines factors – apart from technical – that need to be taken into consideration when making design choices at the concept stage for commercial development of Stirling engines and other energy conversion technologies.

V Conclusions

- Japan is the number one geographic region in terms of proprietary designs for Stirling engines;
- The time period before 2007 is a good source of knowledge about fundamental questions, especially research conducted in the United States, and after 2007 – about application designs of Stirling engines;
- Design configurations selected by Japanese organizations better reflect the requirements for commercial Stirling engines, while the design configurations selected by the United States organizations are better for Stirling engines in space explorations and military activities;
- China has had original work but also mimicked to some degree knowledge generated in United States and Japan. This differentiation of original and mimicked research and patents helps to identify unique design configurations;

- Information about Stirling engines in scientific publishing and patents are two distinct domains of technical knowledge about Stirling engines. When analyzing design alternatives for Stirling engines, one should explore both domains to find original design solutions;
- The controlled comparison between alpha, beta and gamma configurations was selected for the system level design stage with the objective to compare engine power, efficiency, and cost while controlling bore, stroke, hot and cold temperatures;
- The process of downsizing the number of design concepts and derived design configurations can be completed using the tradespace exploration approach, which enumerates predefined design architectures in the space of conflicting system design metrics and helps to select design alternatives that better satisfy market requirements;
- The alpha configuration was selected as one of system-level requirements for the detail design stage, based on the results of applying tradespace exploration to compare alpha, beta and gamma configurations.

REFERENCES

[1] Katayama, M., Sawada, D. and Yaguchi, H., Toyota Motor Corp, 2014. *Exhaust heat recovery system*. U.S. Patent 8,776,516.

[2] Kim, T. and Kwangweon, A.H.N., Hyundai Motor Co, 2014. *Stirling refrigerator for vehicle*. U.S. Patent Application 13/710,262.

[3] Ichikawa, Y., Akazawa, T., Abe, S., National Maritime Research Institute, Estir, Co., 2013. *Stirling engine control system and ship equipped with Stirling engine*. Japan, Patent WO2013084490A1.

[4] Ozaki, S., Yamamoto, Y. and Yabe, H., Isuzu Motors Ltd, 2016. *Free-piston type stirling engine*. U.S. Patent 9,371,798.

[5] Boote, D.N. and Beile, P., 2005. Scholars before researchers: On the centrality of the dissertation literature review in research preparation. *Educational researcher*, *34*(6), pp.3-15.

[6] Randolph, J.J., 2009. A guide to writing the dissertation literature review. *Practical assessment, research & evaluation*, *14*(13), pp.1-13.

[7] Daim, T.U., Rueda, G., Martin, H. and Gerdsri, P., 2006. Forecasting emerging technologies: Use of bibliometrics and patent analysis. *Technological Forecasting and Social Change*, *73*(8), pp.981-1012.

[8] Gavel, Y. and Iselid, L., 2008. Web of Science and Scopus: a journal title overlap study. *Online information review*, *32*(1), pp.8-21.

[9] Cairelli, J.E., 1994. *SPRE I Free-Piston Stirling Engine Testing at NASA Lewis Research Center* (No. NASA-TM-100241; CONF-880457-5). National Aeronautics and Space Administration, Cleveland, OH (United States). Lewis Research Center.

[10] Hill, D., Goodnight, T., Suarez, V., Lewandowski, E., Qiu, S., Callahan, J. and Richardson, R., 2005. Dynamic Characterization of STC Stirling Convertor and Improved Mechanical Integration in SRG110 Power System. In *3rd International Energy Conversion Engineering Conference* (p. 5746).

[11] Dyson, R.W., Geng, S.M., Tew, R.C. and Adelino, M., 2008. Towards fully three-dimensional virtual Stirling convertors for multi-physics analysis and optimization. *Engineering Applications of Computational Fluid Mechanics*, *2*(1), pp.95-118.

[12] Briggs, M., Gibson, M., Geng, S., Pearson, J. and Godfroy, T., 2012, July. Development Status of the Fission Power System Technology Demonstration Unit. In *10th International Energy Conversion Engineering Conference* (p. 3711).

[13] Niu, Y., Simon, T., Ibrahim, M., Tew, R. and Gedeon, D., 2003, August. Measurements of Unsteady Convective Heat Transfer Rates within a Stirling Regenerator Matrix Subjected to Oscillatory Flow. In *1st International Energy Conversion Engineering Conference (IECEC)* (p. 6013).

[14] Mancini, T., Heller, P., Butler, B., Osborn, B., Schiel, W., Goldberg, V., Buck, R., Diver, R., Andraka, C. and Moreno, J., 2003. Dish-Stirling systems: An overview of development and status. *Journal of Solar Energy Engineering*, *125*(2), pp.135-151.

[15] Chen, G. and McEntee, J., 1991. Development of a 3-kW free-piston Stirling engine with the displacer gas-spring partially sprung to the power piston. In *IECEC'91; Proceedings of the 26th Intersociety Energy Conversion Engineering Conference*, Vol. 5, pp. 233-238.

[16] Hendrickson, M., Podlesak, T., Huth, J., Roychoudhury, S., Brandhorst, H., Kirby, R. and Rose, F., 2006. 160 W Dual Opposed Stirling Engine Power Source for Military Applications. In *4th International Energy Conversion Engineering Conference and Exhibit (IECEC)*, p. 4039.

[17] Ernst, W.D. and Shaltens, R.K., 1997. Automotive Stirling engine development project.

[18] Ernst, W.D., Meacher, J.S. and Bascom, R.C., 1992. Status and emissions results for natural-gas-fired stirling engine. *SAE transactions*, pp.529-540.

[19] Poston, D.I., McClure, P.R., Dixon, D.D., Gibson, M.A. and Mason, L.S., 2014. Experimental Demonstration of a Heat Pipe–Stirling Engine Nuclear Reactor. *Nuclear Technology*, *188*(3), pp.229-237.

[20] Brandhorst Jr, H.W., Kirby, R.L. and Chapman, P.A., 2008, January. Progress in Developing a New 5 Kilowatt Free-Piston Stirling Space Convertor. In *AIP Conference Proceedings*, Vol. 969, No. 1, pp. 602-608. AIP.

[21] Keiter, D.E. and Holliday, E.S., Sunpower Inc, 2008. *Electronic controller matching engine power to alternator power and maintaining engine frequency for a free-piston stirling engine driving a linear alternator*. U.S. Patent 7,453,241.

[22] Wood, J.G., Sunpower Inc, 2018. *Stirling engine or cooler heat exchanger*. U.S. Patent 9,945,322.

[23] Clark, D.A., Lowrie, J.R. and Allderidge, H., Microgen Energy Ltd, 2006. *Heating appliance*. U.S. Patent 7,021,554.

[24] Kagawa, N., 2002. Experimental study of a 3-kW Stirling engine. *Journal of propulsion and power*, *18*(3), pp.696-702.

[25] Watanabe, H., Isshiki, N. and Ohtomo, M., 1996, August. Experimental studies on atmospheric Stirling engine NAS-2. In *IECEC 96. Proceedings of the 31st Intersociety Energy Conversion Engineering Conference*, Vol. 2, pp. 1271-1276. IEEE.

[26] Isshiki, N., Wyczalek, F. and Isshiki, S., 2005. Stirling Air Cycle Silent Emergency Home [APU] Auxiliary Electric Power Concepts: Novel Six Cylinder Single Acting Isshiki Z-Crank and a Free Piston Option. In *3rd International Energy Conversion Engineering Conference*, p. 5619.

[27] Hoshino, Takeshi, Hitoshi Naito, Tsutomu Fujihara, and Kunihisa Eguchi. "Experimental study on Stirling engine generator and solar receiver system for future space applications." In *Collection of Technical Papers. 35th Intersociety Energy Conversion Engineering Conference and Exhibit (IECEC)(Cat. No. 00CH37022)*, vol. 1, pp. 266-271. IEEE, 2000.

[28] Miwa, M., Sumi, T., Biwa, T., Ueda, Y. and Yazaki, T., 2006. Measurement of acoustic output power in a traveling wave engine. *Ultrasonics*, *44*, pp.e1527-e1529.

[29] Hoshi, A., Tezuka, N., Sasaki, S., Fujimoto, I. and Yamada, N., 2008. Development of Practical Stirling Engine for Co-Generation System Using Woody Biomass Fuels. In *Proceedings of ISES World Congress 2007, Vol. I–Vol. V*, pp. 2453-2457. Springer, Berlin, Heidelberg.

[30] Li, Z., Haramura, Y., Tang, D. and Guo, C., 2015. Analysis on the heat transfer characteristics of a micro-channel type porous-sheets Stirling regenerator. *International Journal of Thermal Sciences*, *94*, pp.37-49.

[31] Yoshimura, K., Takai, K., Kitamura, Y., Yamagami, S., Sakamoto, J. and Yasumura, H., Sharp Corp, 2007. *Stirling engine*. U.S. Patent Application 10/561,273.

[32] Tsuneo, E. et al. Stirling engine. Honda Motor Co. Ltd., 2015. Japan, Patent JP 2015132246 A.

[33] Asai, M. and Ban, M., Honda Motor Co Ltd, 2003. *Drive unit for prosthetic limb*. U.S. Patent 6,645,252.

[34] Hironori, F. et al. Vehicular power source device. Honda Motor Co. Ltd., 2009. Japan, Patent JP 2009167824 A.

[35] Wu, F., Chen, L., Sun, F. and Wu, C., 2000. Finite-time exergoeconomic performance bound for a quantum Stirling engine. *International Journal of Engineering Science*, *38*(2), pp.239-247.

[36] Luo, E.C., Ling, H., Dai, W. and Yu, G.Y., 2006. Experimental study of the influence of different resonators on thermoacoustic conversion performance of a thermoacoustic-Stirling heat engine. *Ultrasonics*, *44*, pp.e1507-e1509.

[37] Huang, X., Zhou, G. and Li, Q., 2013. Thermodynamic analysis of onset characteristics in a miniature thermoacoustic Stirling engine. *Journal of Thermal Science*, *22*(3), pp.216-222.

[38] Du, J.L., Tang, D.W. and Li, Z.G., 2011. Radiative Heat Transfer Characteristics Between 5 kW Solar Simulator and Stirling Engine's Heat Receiver. *Joural of Engineering Thermophysics*, *32*, pp.985-988.

[39] Zhou, X.L., Chen, X.L., Zhang, W.G. and Yu, C., 2013. Analysis on Combustion Flame of a Stirling Engine by Using Ethanol as Fuel. *Journal of Shanghai Jiaotong University*, *4*.

[40] Duan, C., Sun, C., Shu, S., Ding, G., Jing, C. and Chang, J., 2015. Similarity design and experimental investigation of a beta-type Stirling engine with a rhombic drive mechanism. *International Journal of Energy Research*, *39*(2), pp.191-201.

[41] Gaichang, Z., Zhensheng, W. *Multi-disc condensation tracking device, solar energy condensation thermal power generation device, solar energy hydrogen production device and solar-powered seawater desalination device.* Dalian Honghai New Energy Dev. Co. Ltd., China, patent CN 202813830 U.

[42] Yu, G. et al. *A portable power generation system*. Institute of Engineering Thermophysics, Chinese Academy of Science, 2016. China, Patent CN 104564419 B.

[43] Xucong, S. et al. *Arrangement structure for heater for Stirling engine and mounting method*. Shanghai Micropowers Co. Ltd., 2017. China, pending patent application CN 108167087 A.

[44] Lu, C. *Stirling solar power system*. China Stirling Engine Mfg. Ltd., 2017. China, patent CN 207349039 U.

[45] Lu, C. *Portable cold insulation box*. China Stirling Engine Ltd., 2017. China, pending patent CN 107062740 A.

[46] Titran, R.H. and Stephens, J.R., 1984. Advanced high temperature materials for the energy efficient automotive Stirling engine. *Journal of materials for energy systems*, *6*(2), pp.114-121.

[47] Shaltens, R.K., 1986. *Stirling powered van program overview* (No. 861303). SAE Technical Paper.

[48] Crude oil prices from 1861. Quandl, online database. Last accessed 05/12/2019. https://www.quandl.com/data/BP/CRUDE_OIL_PRICES-Crude-Oil-Prices-from-1861

[49] Smirnov, D. and Golkar, A., 2015. Stirling engine systems tradespace exploration framework. *Procedia Computer Science*, *44*, pp.558-567.

[50] Schraagen, J.M., 1993. How experts solve a novel problem in experimental design. *Cognitive science*, *17*(2), pp.285-309.

[51] Ross, A.M., Hastings, D.E., Warmkessel, J.M. and Diller, N.P., 2004. Multi-attribute tradespace exploration as front end for effective space system design. *Journal of Spacecraft and Rockets*, *41*(1), pp.20-28.

CHAPTER III
GAME THEORY IN DESIGN OPTIMIZATION

The multi-disciplinary nature of energy-conversion systems requires a group of disciplinary designers working on the detail design of a single system. Unlike the conceptual and system design stages, the detail design stage is decentralized from the organizational point of view. Each designer is qualified in his discipline and applies relevant design techniques; therefore developing design guidance for each discipline is not the objective of this chapter. However, the interplay between the disciplinary designers, the communication strategies, and the organization of a multi-disciplinary design team are challenging aspects of the detail design process with great potential for improvement. In Section I.6.3 of Chapter I, I already mentioned that given finite time and resources in the commercial development, the interplay among designers becomes a decisive factor in a multi-disciplinary design process of energy conversion systems. The previous chapter supplied one of key system-level requirement for the detail design stage – the development of the alpha-type Stirling engine. This chapter conducts the analysis of the detail design process that focuses on the development of energy conversion system based on the alpha-type Stirling engine. The main objective of this analysis is to understand how collaboration of designers affect the outcome of the detail design stage and offer some practical recommendation of organizing the detail design process. The chapter proposes a design method that helps to account for and mitigate the negative influence of this collaboration interplay on design process outcomes. The chapter employs results of the tradespace exploration model described in Section IV.1.9 and extends this model with new algorithms drawn from game theory. The principles of this design optimization method could be applied in the analysis of the detail design process of other energy conversion technologies (ECT). Their design would involve disciplinary engineers. The problems related to their interaction, which are discussed in the following section, would be still relevant.

I INTRODUCTION

Over the past two decades, most research in the design optimization of complex engineering systems [1] has emphasized the use of multi-objective optimization [2]-[7], tradespace exploration [8]-[10] and other enhancing methods in decision theory [11]-[18] in order to obtain globally optimal designs. The existing body of literature suggests that a general problem formulation of global optimization inherently assumes a centralized design team with perfect information exchange, full control of design variables, and transparent mechanism for finding the consensus. However, these inherent assumptions seem to contradict to the real design process structure, thus possibly rendering the outcomes of conventional optimization models inaccurate.

The structure of real design processes seems to differ from the one commonly assumed by system engineers in global optimization models due to several factors. Real design of complex engineering systems in different economic sectors, such as energy, urban environment, aviation, or transportation, requires the involvement of several discipline design teams [19], [20] that frequently work for different organizations. Development projects typically are organized in consortia of multiple entities such as large system integrators, small and medium enterprises, research institutions and universities. Previous research identified several problems related to the cooperation of design teams in multidisciplinary design processes: conflicting outcomes due to design decisions interactions [21]-[23], biased or limited informational exchange due to intentional biases or bounded designers' rationality [24], [25], communication barriers [26], [27], and the protection of sensible design information [28]. A systematic understanding of how these factors affect the result of global design optimization is still lacking. This problem may lead to unrealistic stakeholder expectations, impractical technical requirements, and poor decision-making based on inaccurate optimization modeling results. Drawing upon this gap in scholarship, the present work proposes a new methodology using game theory [29] for a potentially more accurate formulation of a general design optimization problem.

Notations

m-CHP	Micro combined heat and power plant
f_i	Objective function *i* of an engineering system
d_h	Design variable *h* of an engineering system
$\boldsymbol{d}$	Vector of design variables
D	Feasible region of design variables
$\boldsymbol{s}$	Vector of objective functions
S	Feasible space of objective functions
N	Number of discipline designers that design an engineering system
$\boldsymbol{d}^v$	Vector of design variables controlled by a discipline designer *v*
$\boldsymbol{d}^{-v}$	Vector of design variables controlled by discipline designers, except the designer *v*
u_1 , u_2	Generic design variables
G_1 , G_2	Generic objective functions of a system
P	Power output in kW for a m-CHP, objective function
A	Dimensionless affordability of heat exchangers for a m-CHP, objective function
b	Stirling engine bore size in m for a m-CHP, design variable
s	Stirling engine stroke size in m for a m-CHP, design variable
T_h	Stirling engine heating temperature in K for a m-CHP, design variable
T_c	Stirling engine cooling temperature in K for a m-CHP, design variable
Indexes	
i, j, p	Index of an objective function
t	Total number of objective functions
h, b	Index of a design variable
m	Total number of design variables
opt	Optimal vector of objective functions
PO	Pareto-optimal vector of design variables
v	Index of a discipline designer
NE	Nash equilibrium vector of design variables
l, s	Index that denotes a combination of design variable values
n	Total number of combinations of design variable values (system architectures)

I.1 Related work

Game theory is widely used to study economic, social, political and biological phenomena [29]. To date, the application of game theory in engineering design has received scant attention in the research literature. Vincent [30] first proposed the application of game theory in design research. He introduced the notion of multi-objective optimization for a design process decentralized between several discipline designers. Since then, the application of game theory in the design of complex engineering systems has been receiving growing interest. Xiao *et al.* [31] considered working design environment, where designers are in isolation due to organizational barriers, time schedules and geographical constraints. The authors proposed a framework to solve multi-objective multidisciplinary design problems using non-cooperative game theory. The work discussed the solution algorithm and applied the framework to the two-team design of a thin-walled pressure vessel. Several other works studied real-world design examples in non-cooperative game settings. Holmberg *et al.* [32] and Désidéri *et al.* [33] discussed examples of applying game theory to find robust structural designs through the search of Nash equilibria among two competitive disciplines. Della Vecchia *et al.* [34] made a step further and discussed the non-cooperative design of an airplane wing by three design teams. Han and Nehorai [35] extended the design setting from non-cooperative to cooperative two-team design. Most recently, Vermillion and Malak [36] suggested an analytical approach to incentivize collaboration in the design process in game-theoretic settings. Sha [37] analyzed how irrational behavior of designers affected outcomes and underlined the need to inform existing theoretical design models with real-world behavior of designers. Lewis and Mistree [38] directed their attention to the underlying assumption of conventional global optimization techniques and explored different design collaboration modes using game theory. Xiao *et al.* [39] have made an initial effort to analyze the effect of design decision freedom by allowing designers to select unconstrained design decision choices. Interesting insights into sequential game theoretic design were obtained in the work of Bhatia *et al.* [40]. Such scholarly work has gradually extended the possibilities of game theory in design optimization.

However, the generalizability of the published research on this problem remains problematic due to several reasons. Much of the research studies until now have not studied in detail the organizational contradiction between the centralized global optimization design models and the decentralized nature of real design processes. Most of the studies focused on applying game theory in design optimization as an alternative approach to global multi-objective optimization, without an attempt to unify both global and local optimum perspectives of discipline designers. Additionally, the studies focused on the search of Nash equilibria under a different set of assumptions but did not evaluate the applicability of general game theory assumptions to the design process as such critically. Furthermore, the effect of different design authority among discipline designers on the system design outcome is not understood in literature. To address these gaps, we can invoke some well-studied formulations from game theory and global optimization and apply them in the context of a design process.

I.2 NORMAL-FORM GAME FORMULATION OF A DESIGN PROCESS

A normal-form game has been widely applied in economics, politics and biology [29]. It can be instructive to apply the same representation to a general design process with two designers to evaluate potential outcomes (Fig. 1). Each designer faces two fundamental decisions: a system-global or a discipline-local optimum design. In Fig. 1, design outcomes reflect a structure of the game referred to as the Prisoner's Dilemma [29]. In this instance, designers have an incentive to commit to the global optimum, but each designer, in addition, has an incentive for a "free-ride" and may deviate to a local optimum. Other typical game structures can be considered, such as *Bach and Stravinsky*, *Stag Hunt*, *Hawk-Dove*, depending on the design settings [29].

		Designer 1	
		Global	Local
Designer 2	Global	2 , 2	3 , 0
	Local	0 , 3	1 , 1

Fig. 1. Design game formulation with outcomes similar to the Prisoner's Dilemma.

To model and evaluate the outcomes of decisions in such a formulation of a design process, a mathematical link between the decisions and outcomes needs to be established first. Many works have analyzed the mathematical formulation of multi-objective problems. We can use them to provide a similar multi-objective definition of an engineering system.

I.3 Definition of engineering system

Suppose that in the general case there exists a multi-objective engineering system, which is defined by a set of conflicting objective functions f_i , where $f_i \in \mathbf{R}$ and $i = 2,...,t$, $i \in \mathbf{N}$. The index i is higher than one because we are interested in a multi-objective engineering system. The total number t and the type of objective functions is defined by a system engineer and may include for example system's cost, weight, or fuel consumption. The goal of the system engineer is to minimize simultaneously and find a compromise between the values of all objective functions.

Suppose further that every objective function f_i that defines an engineering system depends on a set of design variables d_h with $h \in \mathbf{N}$, which is expounded by the vector of design variables $\boldsymbol{d} = \left(d_1, d_2, ..., d_m\right)^T$ that belongs to the nonempty feasible region $D \subset \mathbf{R}$ with the total number of variables *m*. The feasible region of design variables D and the form of $f_i\left(\boldsymbol{d}\right)$ are also defined by the system engineer. Thus, an engineering system is defined by a vector $\boldsymbol{s}$ that is an image of the design variable vector $\boldsymbol{d}$ and can be denoted as

$\boldsymbol{s} = \boldsymbol{f}(\boldsymbol{d}) = \left(f_1(\boldsymbol{d}), f_2(\boldsymbol{d}), ..., f_t(\boldsymbol{d})\right)^T$. Additionally, an image of feasible design variables into the space of an engineering system is called a feasible space of objective functions $S = \boldsymbol{f}(D)$. The feasible system space contains all possible configurations of an engineering system subject to the constrained region D. After the formulation of the engineering system under design optimization, the system engineer selects the design optimization problem formulation. Conventionally, the system engineer assumes full control of all objective functions and design variables.

I.4 CONVENTIONAL OPTIMAL DESIGN FOR A MULTIDISCIPLINE ENGINEERING SYSTEM

In a general case, the goal of the system engineer is to obtain an optimal system design, which is defined by the system design vector $\boldsymbol{s}_{opt}$. Given the multiple objective functions, the problem of obtaining $\boldsymbol{s}_{opt}$ is typically related to multi-objective optimization methods, tradespace exploration and alternative heuristics approaches. The choice of the method is defined by several factors, such as the knowledge of the system engineer on relevant f_i, the feasible region D, the form of each $f_i(\boldsymbol{d})$ and in a more general case the form of $\boldsymbol{f}(\boldsymbol{d})$.

In the multi-criteria analysis, the optimal system design $\boldsymbol{s}_{opt}$ is commonly referred to as the *Pareto-optimal solution*. More specifically, $\boldsymbol{s}_{opt}$ is regarded as optimal when none of the objective functions can be improved without deteriorating of at least one of them. The vector of design variables $\boldsymbol{d}_{PO} \in \mathbf{R}$ is called Pareto-optimal if there does not exist another vector, $\boldsymbol{d} \in \mathbf{R}$ such that $f_i(\boldsymbol{d}) \le f_i(\boldsymbol{d}_{PO})$ for all $i = 2,...,t$ and $f_j(\boldsymbol{d}) < f_j(\boldsymbol{d}_{PO})$ for at least one index j. Additionally, there exists a notion of a decision-maker to identify a preferred system design among all obtained Pareto-optimal design alternatives. To obtain the set of $\boldsymbol{s}_{opt}$, the system engineer formulates an optimization problem for multi-objective system design in a conventional form, which can be summarized as follows:

$$
\begin{aligned}
&\text{minimize } \boldsymbol{f}(\boldsymbol{d}) \\
&\text{subject to } \boldsymbol{d} \in D \\
&\text{solution :} \\
&\boldsymbol{s}_{opt} = \boldsymbol{f}(\boldsymbol{d}_{PO}), \boldsymbol{d}_{PO} \in D \\
&\text{such that} \\
&f_i(\boldsymbol{d}) \geq f_i(\boldsymbol{d}_{PO}), \forall i \in \{1,2,...,t\} \text{ and} \\
&f_j(\boldsymbol{d}) > f_j(\boldsymbol{d}_{PO}), \text{for at least one } j \in \{1,2,...,t\}.
\end{aligned}
\tag{1}
$$

I.5 PROBLEM FORMULATION AND RESEARCH QUESTION

Expression (1) is predicated on the underlying assumption that all discipline designers who would have participated in the design process, would choose to commit to a global optimum design outcome. Without this assumption, it would be difficult, if not impossible, to minimize all f_i simultaneously while controlling all design decisions at the same time. Most, if not all, global optimization methods inherently assume that designers always commit to the *{Global, Global}* (Fig.1) optimum solution.

However, this formulation of design optimization is incomplete. Real world design typically is conducted by several discipline designers, possibly working for different organizations with different local objectives. Each of them controls some design decisions and the related objective functions. Due to these reasons and due to problems of design team cooperation in multidisciplinary design processes discussed in Section I.A designers may also choose to commit to *{Global, Local}*, *{Local, Global},* or *{Local, Local}* optimum solutions (Fig. 1). The mathematical framework that unifies all possible design outcomes is still missing. Such a framework would provide a complete formulation of a design problem that takes into account designers decision-making. It would also allow the analysis of previously discussed problems of design team cooperation in multidisciplinary design processes. Furthermore, the framework would generalize the application of game theory in design optimization. As a result, it would minimize unrealistic stakeholder expectations, impractical technical requirements, and poor decision-making based on inaccurate optimization modeling results. Therefore, this study is exploring the answer to the question of how, and to which extent, do the design process outcomes with multiple designers diverge from ideal design outcomes characterized by Pareto optimum design solutions (Expression 1)?

This work provides a novel contribution to the field of study by, first, critically evaluating the contradiction between the formulation of a global optimization problem with an inherently assumed centralized design team and the decentralized structure of a real design process. Second, it unifies in a novel design optimization approach both possibilities for multidisciplinary teams to commit to system-global and discipline-local optimum design solutions. Third, it evaluates critically key assumptions of game theory and their applicability to design optimization. Fourth, it studies the implications of varying design authority between discipline designers using a numerical case study.

The remainder of this chapter is structured as follows. Section II introduces the novel optimization framework and describes the system model for the Stirling micro power plant, which is the object of the case study. Section III shows the results of the case study, which are discussed in Section IV. Section V concludes on the most relevant findings of this study.

II FRAMEWORK

II.1 NOVEL APPROACH FOR THE OPTIMAL DESIGN IN A MULTIDISCIPLINE SYSTEM

Suppose that N discipline designers design an engineering system. Each discipline designer v controls design variables $\boldsymbol{d}^v = \left(d_1^v, d_2^v, \ldots, d_b^v\right)^T$, $\boldsymbol{d}^v \subseteq \boldsymbol{d}$. Note that index b is different from index m defined in section I.D because m defines the total number of design variables and b defines variables allocated to a designer v. Strictly speaking, vector $\boldsymbol{d}^v$ may contain any d_h^v, not necessarily those (from 1 to *b*) indicated above. The vector $\boldsymbol{d}^{-v}$ controls the design variables by other discipline designers. In a general case, the selection of value for a design variable by a discipline designer is dependent on selected design variable values (design decisions) of other discipline designers such that $D_v\left(\boldsymbol{d}^{-v}\right) \subseteq D$. Each discipline designer is allocated with objective functions that she is responsible for, such that $\boldsymbol{s}^v = \boldsymbol{f}^v\left(\boldsymbol{d}^v, \boldsymbol{d}^{-v}\right) = \left(f_1^v\left(\boldsymbol{d}^v, \boldsymbol{d}^{-v}\right), f_2^v\left(\boldsymbol{d}^v, \boldsymbol{d}^{-v}\right), ..., f_p^v\left(\boldsymbol{d}^v, \boldsymbol{d}^{-v}\right)\right)^T$ and $\boldsymbol{s}^v \subseteq \boldsymbol{s}$. Indexes 1 to *p* denote one possible allocation of objective functions to a designer *v*. The goal of each discipline designer *v* is to choose such $\boldsymbol{d}^v$ that minimizes vector $\boldsymbol{s}^v$, given the design decisions $\boldsymbol{d}^{-v}$ of other discipline designers. For any $\boldsymbol{d}^{-v}$, the solution set to this minimization problem is denoted $S^v\left(\boldsymbol{d}_{NE}^{-v}\right)$. The solutions are known as equilibrium points, the existence of which is proven by the Nash theorem [41]. The equilibrium point is interpreted here as an engineering system design, for which each discipline designer has nothing to gain regarding his $\boldsymbol{s}^v$ by selecting design decision other than $\boldsymbol{d}_{NE}^v$. The problem of finding an equilibrium point is equivalent to the search of vector $\boldsymbol{d}_{NE}^v$ such that *for every discipline designer v*, $\boldsymbol{d}_{NE}^v \in S^v\left(\boldsymbol{d}_{NE}^{-v}\right)$. This framework can be summarized as:

$$
\begin{aligned}
&\text{minimize } \boldsymbol{f}^v\left(\boldsymbol{d}^v, \boldsymbol{d}^{-v}\right), \forall v \\
&\text{subject to } \boldsymbol{d}^v \in D_v\left(\boldsymbol{d}^{-v}\right) \\
&\text{solution:} \\
&\boldsymbol{s}_{opt}^v = \boldsymbol{f}^v\left(\boldsymbol{d}_{NE}^v, \boldsymbol{d}_{NE}^{-v}\right),\ \boldsymbol{d}_{NE}^v \in D_v\left(\boldsymbol{d}^{-v}\right) \\
&\text{such that} \\
&\boldsymbol{f}^v\left(\boldsymbol{d}_{NE}^v, \boldsymbol{d}_{NE}^{-v}\right) \le \boldsymbol{f}^v\left(\boldsymbol{d}^v, \boldsymbol{d}_{NE}^{-v}\right),\ \forall v
\end{aligned}
\qquad (2)
$$

In this approach, the system solution is obtained from the perspective of each discipline designer. This formulation considers all possibilities for general design decisions – global or local optima – in the design process structure (Fig. 1) and represents a typical design optimization problem more realistically. A system engineer can use the design optimization formulation (Expression 2) and therefore account for the commitment to different design choices, either global or local (Fig. 1). Depending on design variables, objective functions and design authority allocation, the optimal design solution can be Pareto-optimal, but not necessarily so.

II.2 ANALYSIS OF GAME THEORETIC ASSUMPTIONS

Osborne [29] discusses key assumptions entailed in game theoretic representations. The application of these assumptions to a design process is analyzed in the paragraphs 1 to 6:

II.2.1 THE DESIGN PROCESS IS A STRATEGIC GAME

According to this assumption, a number of players participate in a design process, and each player has a set of actions and preferences over this set of actions. This assumption is valid as N discipline designers represent players. Each designer has a set of design decisions $\boldsymbol{d}^v$, which is the set of actions. Each designer has preferences over the set of decisions that are described by the minimization strategy of allocated objective functions $f_i^v\left(\boldsymbol{d}^v, \boldsymbol{d}^{-v}\right)$.

II.2.2 DESIGNERS PURSUE WELL-DEFINED EXOGENOUS OBJECTIVES

This assumption is adequate for the design process, because each discipline designer v is typically allocated with a well-defined objective engineering system function f_i^v, which she attempts to minimize.

II.2.3 DESIGNERS ARE RATIONAL

According to this assumption, a designer v chooses the best design decision, for the allocated system objective function $f_i^v\left(\boldsymbol{d}^v, \boldsymbol{d}^{-v}\right)$, among all design decisions available to him and to other designers $-v$. The validity of this assumption is predicated on whether or not the objective system function f_i mirrors the designer objective function f_i^v. The designer may have alternative subjective preferences, like those analyzed by Sha [37], that he minimizes in parallel with the objective system function, such that $f_i \neq f_i^v$. Here we assume that the system objective function accurately reflects the objective function of the designer and these functions are effectively the same. In future work, the designer's objective function could be formulated as a combination of the engineering system objective function and the subjective preferences, including incentives of the designer.

II.2.4 Perfect information about the design process

Each designer has complete information about the rules of the design process and knows other's designers objective functions. Also, each designer is aware that other designers have complete information about the design process and can form a belief about action choices of other designers and derives from this belief the knowledge of what design decisions $\boldsymbol{d}^{-v}$ other discipline designers will choose [29]. Each designer may not be fully cognizant what specific decisions the other designers would choose in a given design process, but her experience and previous involvement in design processes lead her to believe of other designers' actions. These assumptions may be valid if N discipline designers are considered to be design experts in their discipline with strong experience of participation in design process. This assumption could be a valid condition for industry design processes.

II.2.5 No cooperation between design teams

In the case of absent cooperation, design teams are not willing to make compromises that could erode their f_i^v. Although, this situation may happen in real design, in this study we assume that a non-cooperative design is a limit state that describes a perfect decentralization of a design process and could be used to compare design outcomes with different cooperation protocols or with design outcomes of the opposite limit state – a perfect centralization with Pareto-optimal solutions.

II.2.6 Time is absent from the model

The designers make their decisions once and for all and they make them simultaneously. No designer is informed, when he chooses his design decision, about the decision of other designers. This is a strong assumption, but it allows studying the design process as a strategic game and abstract from complications that arise when a designer is allowed to change his decisions as events unfold in repeated games.

The discussed assumptions may not be valid in more complex design process models. However, in this study, they represent a limit state of perfectly decentralized design, which can be compared with a more complicated formulation of design processes using game theory and which is the subject of future work.

II.3 NASH FRONT

Vincent [30] provided an initial analysis of locations for Pareto-optimal set and Nash equilibrium in the feasible region D for a two-designer optimization process with two generic design cost functions $G_1(u_1,u_2)$ and $G_2(u_1,u_2)$ allocated to each designer correspondingly, where designer 1 controls design variable u_1 and designer 2 – u_2 (Fig. 2).

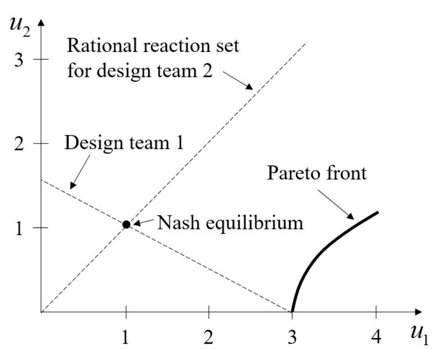


Fig. 2. Analysis of locations for Pareto front and Nash equilibrium design outcomes for the design variables coordinates. Adapted from Vincent [30].

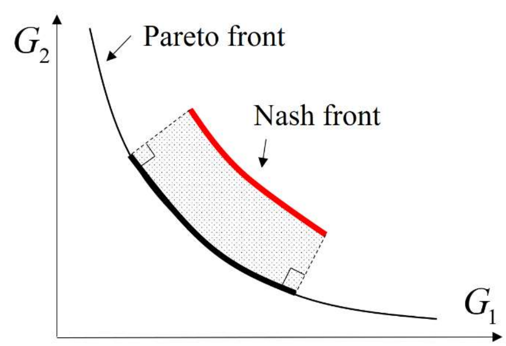


Fig. 3. Non-dominated Pareto front, dominated Nash front and the region between perfect centralization and perfect decentralization of design processes.

However, this analysis used rational reaction sets in the coordinate systems of design variables and did not study how the location of Nash equilibria would change if the control of design variables or design authority was reassigned between design teams. Therefore, it is difficult to evaluate practically how Nash equilibrium affects the design outcomes characterized through objective functions, and the effect of design authority allocation is not known. The notion of the *Nash front* (Fig. 3) is introduced in this work to cover this gap. The Nash front is considered in the coordinate system of objective functions $G_1\left(u_1, u_2, \ldots u_n\right)$ and $G_2\left(u_1, u_2, \ldots u_n\right)$ and is a set that consists of Nash equilibria. Such equilibria are the result of a modeled design process using the proposed game theoretic approach with all possible allocations of design authority between designers (for examples of allocations see Section II.4.4). Specifically, a different design authority characterizes a different number and type of design variables that designers control during a discipline optimization process, which results in different Nash equilibria. The combination of these equilibrium points forms the Nash front.

In this formulation, the Nash front so derived using the classical assumptions of game theory (Sections II.1 and II.2) finds the boundary of design solutions that appear assuming perfect decentralization. In other words, Nash front represents a set of solutions obtained when there is no simultaneous control of all objective functions; they are allocated for local optimization to discipline designers. In addition, there is no centralized control of all design decisions; they are controlled by responsible designers, to which these decisions were allocated. The Pareto frontier – global or local – identifies the design solutions assuming perfect centralization. Such solutions are optimal among all possible or neighboring solutions correspondingly and obtained by optimizing all objective functions simultaneously and by controlling all design decisions in a selected feasible design space. All solutions in between the two fronts correspond to those that result with changing assumptions on the game theory setting. This result is therefore useful as it finds the bounds of all possible solutions between the two asymptotic conditions we identify. The work of Lewis and Mistree [38] provided some examples of different game theory settings for non-cooperative, approximate cooperation, leader-follower, and full cooperation designs. Given the optimization strategy defined for each designer by Expression (2), we may expect a different location of the Nash front relatively to the Pareto front, depending on the number of decision variables in design problem. The next section applies the formulated methodology to evaluate a numerical design optimization of an energy system.

II.4 Case study of a Stirling m-CHP plant

The present numerical case study approach was adopted to provide a rounded, detailed illustration of the difference in results for a conventional design optimization problem [2] and the novel approach proposed in this chapter.

II.4.1 System model of micro-CHP

The Stirling micro combined heat and power plant (m-CHP) is an emerging multi-fuel technology that is based on the Stirling engine and generates electricity and heat at the same time to improve energy conversion efficiency [42].

Two conflicting system metrics ($t = 2$) were assumed in the m-CHP system model: electric power output P in kW and the dimensionless affordability metric of heat exchangers A. In Fig. 4, the blue-dashed area in the lower part represents the boundaries of the mechanical system in the m-CHP, which was related to P. The red-dashed area in the upper part was the thermal system, which includes heat exchanger of the m-CHP and was related to A. The power output metric P was allocated to the mechanical design team. The affordability of heat exchangers A is a normalized metric opposite to the cost of heat exchangers. The metric was allocated to the thermal design team, so that $N = 2$. The affordability metric – as opposed to the cost metric – was selected for simplicity of understanding to have both system metrics maximized by discipline teams. The limitation of the case study to two system metrics and two design teams helped to avoid resourceful calculations and to keep the results at a comprehensible level of complexity. However, there are certain drawbacks associated with this approach: the results could be generalized only for a case of system optimization with two design teams; it is difficult to validate the results of this analysis, because, in real conditions, the design of multi-objective engineering systems would entail high costs of obtaining empirical data for validation. These limitations could be addressed in future research.

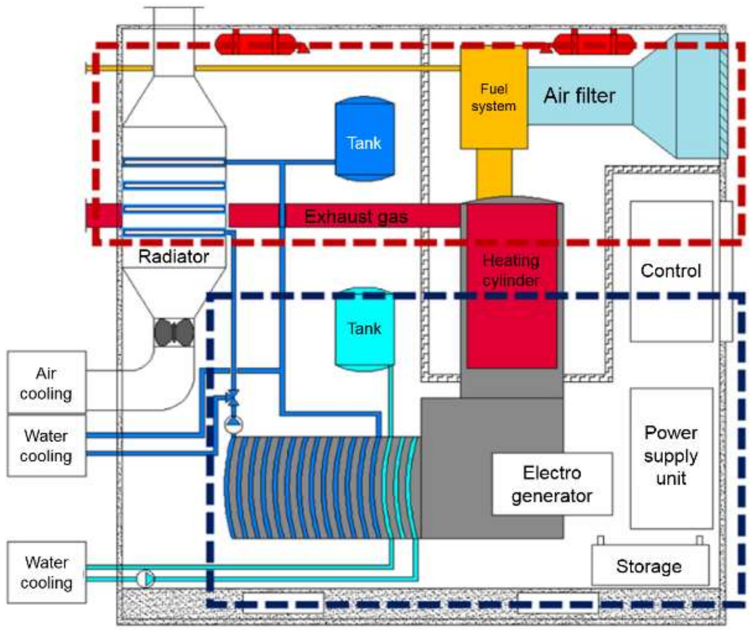


Fig. 4. Stirling m-CHP. Red dashed line is the boundaries of the thermal system, blue dashed line is the boundaries of the mechanical system [9].

For simplicity of demonstration, the m-CHP system had four design variables ($n=4$): b was the Stirling engine bore size (bore) in meter, s was the engine stroke size (stroke) in meter, T_h was the engine heating temperature in K, and T_c was the engine cooling temperature in K. The system model defined the relationship between design variables and system metrics using corresponding objective functions, such that $P=f_1\left(b,s,T_h,T_c\right)$ and $A=f_2\left(b,s,T_h,T_c\right)$. To formulate these system objective functions, the fundamental analytical equations for the Stirling engine were used [43] with the application of cost functions [44] and heat transfer parametric functions [45] for heat exchangers in Stirling engines. The modelling equations and model validation are in the Appendix to this chapter.

II.4.2 General approach

The present section discusses calculation steps that were conducted in order to compare numerical calculation results of the conventional optimization problem formulation (Expression 1) and results of the novel game-theoretic formulation (Expression 2). The chapter applied the tradespace exploration approach as an example of the conventional design optimization problem to find Pareto-optimal set for m-CHP system design. However, a different approach could also be used, for example, multi-objective optimization, and it is the subject of future work. In parallel, a novel game-theoretic framework was applied. This process – named in this work *gamespace exploration* similar to tradespace exploration – is shown in Fig. 5 and unfolded in five steps: 1) Enumeration of architectures, 2) Calculation of metrics (objective functions), 3) Allocation of metrics to gamespace matrix, 4) Tradespace calculation, and 5) Integration of Pareto set and Nash equilibria.

II.4.3 Architectures and the feasible space of m-CHP design variables

The enumeration of architectures (Step 1) consisted of the enumeration of possible combinations of system design variable values. In Fig. 5, the indexes $[1;n]$ define the number of combinations (system architectures) for design variable values. For the discussion of enumeration techniques in tradespace exploration, a reader may refer to [10]. In this chapter, a full-factorial enumeration was assumed for two different value ranges of design variables (Table 1): for 16 m-CHP architectures and 49,200 m-CHP architectures, which are hereafter referred to Case 1 and Case 2 correspondingly. In real system design, the architecture enumeration is subject to physical or technological constraints [46], knowledge about the system [47], technical standards [48], or existing engineering cultural practices [49]. This chapter assumed exhaustive full factorial enumeration and no enumeration constraints, without loss of generality.

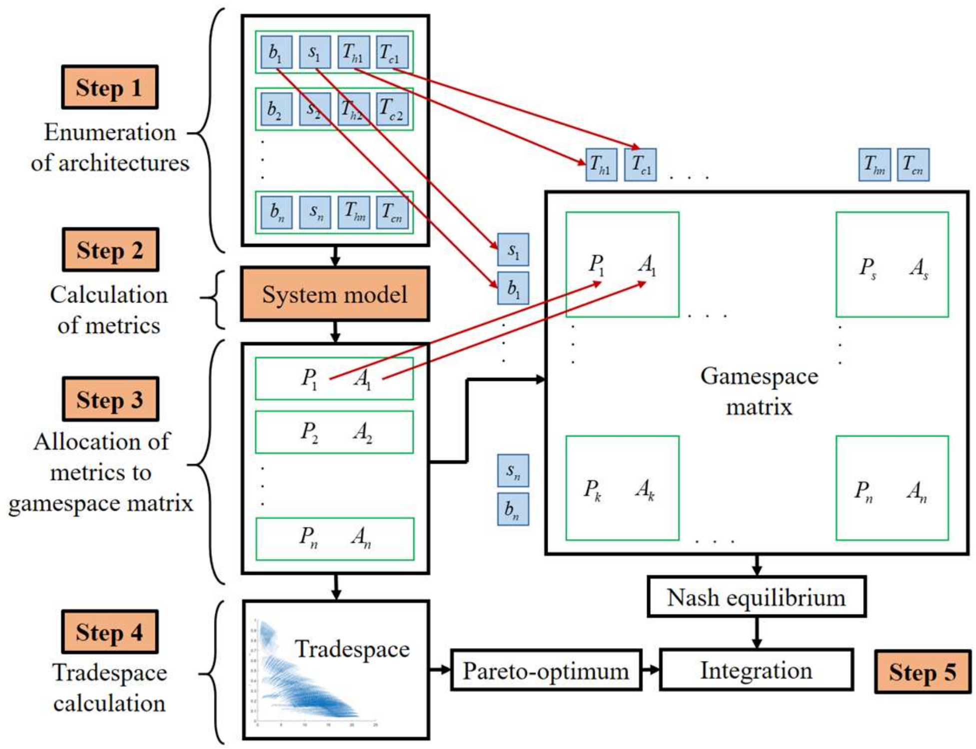


Fig. 5. Gamespace exploration method and its integration with tradespace exploration.

Table 1 shows the ranges of design variable values that were used in Case 1 and Case 2. The ranges were defined by the lowest and the highest value of design variables and by their value increment. The Cases represent two instances with different feasible spaces of design variables $D \subset \mathbf{R}$. In Case 1 with 16 m-CHP architectures, there was one increment type of a design parameter. For example, for T_h the lowest temperature used in the model was 600K and the highest temperature 1005K with the increment of 405K. By applying the full factorial design, the four design variables with two values each resulted in 16 design variable configurations or 16 m-CHP system architectures. For 49,200 m-CHP architectures, the temperature limits remained similar to Case 1, but the increment was 10K. This resulted in many design values and a high number of system architectures. Both, Case 1 and Case 2, as well as intermediate cases or cases with alternative ways of defining feasible spaces of design variables D may occur in a real system design.

Table 1. Range of Design Variable Values for Two Calculations Cases

Design parameter	16 architectures Case 1	49 200 architectures Case 2
T_h , K	600 and 1005	600 to 1005 with increment of 10
T_c , K	301 and 400	301 to 400 with increment of 9
b , m	0.03 and 0.3	0.03 to 0.3 with increment of 0.03
s , m	0.02 and 0.11	0.02 to 0.11 with increment of 0.01

II.4.4 Metrics and the feasible space of the m-CHP engineering system

The calculation of metrics (Step 2) characterized all enumerated architectures under system-level metrics *P* and *A* using the corresponding objective functions f_1 and f_2. In a general case, depending on the design of interest, these metrics could be, for instance, proxy metrics of performance, cost, or risk. This step defined the feasible space S of the m-CHP plant.

Step 3 was the allocation of metrics to the gamespace matrix. The calculated metric values were arranged in the matrix as if they were conceived as results of designers' decisions in a normal-form game. In Fig. 5, metric outcomes $P_l = f_1\left(b_l, s_l, T_{hl}, T_{cl}\right)$ were allocated to design team 1 with design decisions in rows, and metric outcomes $A_l = f_2\left(b_l, s_l, T_{hl}, T_{cl}\right)$ were allocated to design team 2 with design decisions in columns, where $l = \left[1; n\right]$. We named the matrix with calculated metric values that represents a normal-form game formulation – *the gamespace matrix*. The matrix can be interpreted as follows: on each design decision of Team 1, which are known from the perfect information assumption II.2.4, Team 2 has a set of alternatives, which it can select against each decision of Team 1, to maximize its metric. For example, in spacecraft design, a power system team can decide the number of solar arrays and the battery capacity to provide the required spacecraft power supply and to minimize their power subsystem cost. Knowing possible alternatives for the number of solar arrays and the battery capacity, the spacecraft structure team will select esign decisions to maximize the structural reliability of the spacecraft. However, the assumption II.2.5 for the absent cooperation suggests that, although the teams can communicate, they have no intention to compromise and select decisions that would deteriorate their metric. This may be seen as a game-theoretic representation of a limit case in engineering tradeoff discussions and negotiations such as those encountered in conceptual design studies.

Table 2. Allocation of design variables

Allocation number	Mechanical team	Thermal team
1	$[b, s]$	$[T_h, T_c]$
2	$[s]$	$[b, T_h, T_c]$
3	$[b]$	$[s, T_h, T_c]$
4	$[b, s, T_h]$	$[T_c]$
5	$[b, s, T_c]$	$[T_h]$

II.4.5 REALLOCATION OF DESIGN AUTHORITY

The design variables were allocated to the design teams in different combinations in order to analyze different Nash equilibria and to study the implications of the design authority reallocation (Table 2). Different Nash equilibria that resulted from the reallocation of design variables formed a set of Nash equilibrium system designs, which is the *Nash front* that was introduced in Section II.3. For the case of 16 m-CHP architectures and according to Table 2, in point 1, the mechanical team had authority over two design variables $[b, s]$ with two discrete values resulting in four design decisions: $\{b = 0.03\text{m}, s = 0.02\text{m}\}$, $\{0.3\text{m}, 0.11\text{m}\}$, $\{0.03\text{m}, 0.11\text{m}\}$ and $\{0.3\text{m}, 0.02\text{m}\}$. The thermal design team by analogy owned $[T_h, T_c]$ with two numerical values for each and also had four design decisions $\{T_h = 600\text{K}, T_c = 301\text{K}\}$, $\{1005\text{K}, 400\text{K}\}$, $\{600\text{K}, 400\text{K}\}$ and $\{1005\text{K}, 301\text{K}\}$. As a result, sixteen m-CHP designs were possible, subject to design decisions of mechanical and thermal teams. For other allocation of design variables in Table 2, design teams had a different distribution of possible design decisions. The allocation of design variables in Table 2 was applied for both Case 1 and

Case 2. Tradespace calculation (Step 4) allowed building a tradespace of all possible design architectures for Case I and Case 2. Finally, in Step 5, based on the tradespace and gamespace matrix and using Expression 1 and Expression 2, all Pareto-optimal and all Nash equilibrium architectures were identified and integrated into the same plot for the analysis of results.

III RESULTS

Fig. 6 and 7 depict the space of 16 and 49,200 m-CHP architectures correspondingly, which were calculated using the system model and the approach described in Fig. 5. The Nash equilibria architectures were numbered and highlighted with red. Each number was related to a specific allocation of design variables to design teams (Table 2). For Case 2 and Type 1 allocation of design variables (Table 2), Fig. 7 also depicts the absolute difference of \$8200 in the affordability of heat exchangers A for a fixed power output $P = 18\ \text{kW}$ between a global optimum m-CHP architecture and the Nash equilibrium m-CHP architecture. For five different allocations of design variables (Table 2) in the proposed design optimization formulation, the average projected increase of the system capital cost was 22% for fixed power outputs comparing to Pareto-optimal design solutions of the conventional design optimization approach. Similarly, the average reduction of the system power output at fixed affordability for Nash equilibrium solutions amounted to 16%. Table 3 shows how the Nash equilibrium values for objective functions change, depending on the allocation of design variables and on the number of m-CHP architectures. Table 4 shows a gamespace matrix defined in Step 3 of the general approach for the m-CHP design optimization with two design teams and the allocation of design variables according to Type 1 in Table 2.

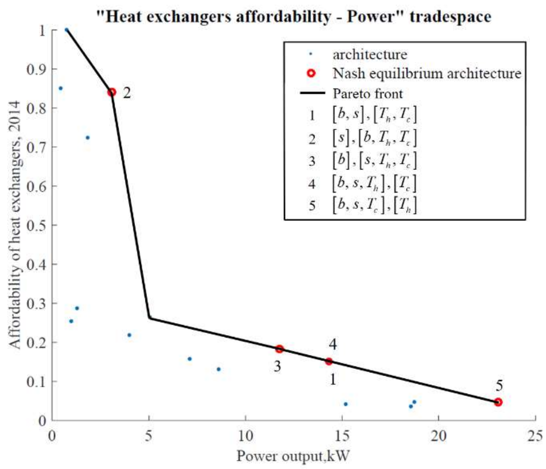


Fig. 6. Tradespace of 16 m-CHP architectures, Case 1.

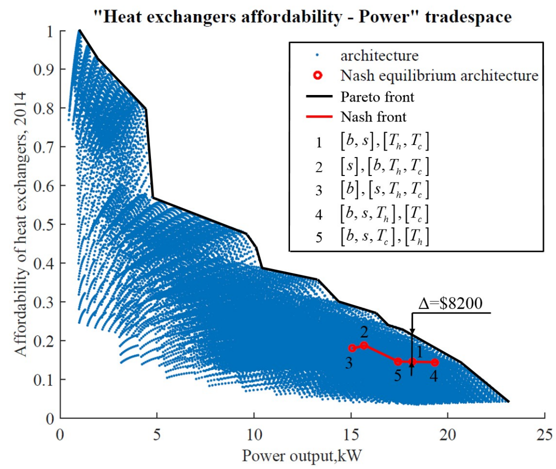


Fig. 7. Tradespace of 49,200 m-CHP architectures, Case 2.

Table 3. Metric values in Nash front

Allocation number	**16 architectures Case 1**		**49,200 architectures Case 2**	
	P **, kW**	A	P **, kW**	A
1	14.3	0.15	18.2	0.15
2	2.9	0.85	15.7	0.19
3	11.8	0.18	15.1	0.18
4	14.3	0.15	19.3	0.14
5	23.0	0.05	17.4	0.14

Table 4. Gamespace matrix for 16 micro-CHP architectures

			Design decisions of the thermal team [heating temperature, cooling temperature], K			
	Heating temperature, K		600	600	1000	1000
	Cooling temperature, K		300	400	300	400
Design decisions of the mechanical team [bore, stroke], m	Bore, m	Stroke, m				
	0.03	0.02	0.8[a], 1[b] P[c]	1.3, 0.31	0.4, 0.85	0.9, 0.27
	0.3	0.02	11.8, 0.18 P	18.7, 0.05 P	7.1, 0.16	15.1, 0.05
	0.03	0.1	2.9, 0.85 P	4.7, 0.28	1.7, 0.73	3.7, 0.24
	0.3	0.1	14.3, 0.15 P, NE[d]	23.0, 0.05 P	8.6, 0.13	18.4, 0.04

[a]Left metric – power output (kW); [b]Right metric – affordability of heat exchanger; [c]P – Pareto-front architectures; [d]NE – Nash equilibrium architecture.

IV DISCUSSION

IV.1 GAME THEORY IN DESIGN OPTIMIZATION

The central question of this study was to analyze how, and to which extent, do the design process outcomes with multiple designers diverge from ideal design outcomes characterized by Pareto optimum design solutions (Expression 1)? Our analysis identified several factors that explain the mechanism of such divergence. The first factor is a different definition of an optimal design. Expression 2 – on the contrary to the centralized and monopolistic approach of finding Pareto-optimal solution in Expression 1 – suggests the optimization approach from the view of discipline designers, where each of them attempts to optimize her local objective taking into account possible design decisions of other designers. This approach explains the possibility to obtain design solutions that are different from Pareto-optimal solutions. This result also suggests a more practical recommendation. Before the start of the detail design process, in a joint meeting between disciplinary designers, each expert should describe his objective function, controlled design decisions and the strategy to maximize the objective function. This discussion will surface conflicts between design parameters and help to plan a joint strategy of maximizing conflicting objectives. The implication of this discussion can be a reduced difference between the design outcome and the Pareto-optimal target. There are several other factors define the extent, to which these solutions diverge.

One of them is the assumption of a non-cooperative design process. In this case, each designer is interested only in the maximization of her metric. This strict specification of the optimization procedure may limit the innovation potential of design optimization based on game theory. Because of this strict specification, a discipline designer does not accept even a negligible penalty to her metric, which is caused by selecting a design decision that can significantly improve metrics of other designers and bring the system design closer to the

Pareto-optimal solution. This assumption is a limit case that defines a worst-case scenario, and the results obtained for the case study incorporate this assumption. Xiao *et al.* [39] proposed an approach to model the unconstrained choice of design decisions. For practical means, this result imply that the design process policy should motivate designers to maximize their own objective function, but receive higher incentive from penalizing their own function if it can lead to a larger gain for the objective function of other discipline experts.

Another factor is the feasible region of design variables. Case 1 and Case 2 describe two different feasible spaces of design variables. A different number of design decision alternatives for a similar optimization problem *ceteris paribus* result in a different divergence between the Pareto optimal and Nash equilibrium designs (Fig. 6 and Fig. 7). The increase in the number of possible design alternatives for each designer provides an opportunity to further maximize the objective function of a designer according to the local optimization strategy defined for each designer in Expression 2. The resulting reaction sets may define a Nash front that coincides with the Pareto front (as in Fig. 6) or be distant from it (Fig. 7). Therefore, the design decision region affects the degree of divergence between the Nash front and the Pareto front. Before the commencement of the design process, it is important to run the gamespace simulation with a different number of design decision alternatives to understand how this variable affect the divergence from the Pareto-front. If the divergence increases, the designers should have lesser options for design parameter values.

One more factor is the reallocation of the design authority between discipline designers. By controlling different sets of design decisions, discipline designers obtain different optimization results using a local optimization technique (Expression 2). Fig. 7 shows five different equilibrium points for five different allocations of design variables. The collection of Nash equilibrium points for different design authority allocation – the Nash front – helps to visually (Fig. 7) demonstrate the degree of divergence, subject to the local optimization approach (Expression 2), the cooperation degree and the feasible region of design variables. The reallocation of design authority (Table 2) causes different Nash equilibrium solutions. For the allocation of Type 2 and 3, where the thermal team controls three out of four design variables, the affordability *A* (thermal team's objective function) of the Nash equilibrium designs for both 16 and 49,200 architectures has the highest values, and the power output *P* (mechanical team's objective function) has the lowest values. The opposite is true for the allocation Types 4 and 5 for 16 architectures and for Type 4 for 49,200 architectures, where the mechanical team controls three out of four design variables and obtains the highest values of power output, whereas the thermal team obtains the lowest values of affordability. This result suggests a disproportional influence of design teams with dominating design authority on system design outcomes. Also, this result may partially explain why real design projects frequently do not meet the original technical requirements [50]-[52]: there is an inadequate analysis of the authority allocation effect on design outcomes. None of the research works known to authors investigated the impact of the design authority allocation on design outcomes, which makes it a fruitful subject for further investigations. For practical purposes, this result imply that if possible the design authority should be distributed between designers equally.

A practical limitation of the obtained results is the two-team design process configuration. Although Expression 2 generalizes the game theoretic design optimization approach for *N* design teams, more studies are needed to validate the game theoretic approach. Only one study [34] known to authors have attempted to analyze the outcomes numerically for the design process with three design teams. Furthermore, even if future work obtained solutions for *N* design teams, the results would still be related to the benefits of individual disciplines. Most real-world design processes undergo the emergence of clusters or social groups of design teams from different disciplines. The effect of these clusters on design outcomes could also be evaluated in future work. Recently Bu *et al.* [53] have paved the way for understanding the emergence of social groups using a game theoretic approach.

The obtained Nash front represents a different paradigm of design optimization that, unlike a conventional approach, assumes a decentralized control of objective functions and design decisions whereas reproducing a more realistic structure of multidisciplinary system design. A significant difference in system capital cost and in power output for the Nash front comparing to the Pareto front that amounted to 22% and 16% further suggests the need to implement this approach in design optimization. Other studies that analyzed different two-team design cases and provided alternative data to derive the difference between metric values for Pareto solutions and Nash solutions are scarce. Vincent [30] analyzed the two-team design for two different cases of generic design cost functions. The discrepancy between Pareto and Nash solutions amounted to 49% and 38% in the first case and 71% and 84% in the second case. Della Vecchia *et al.* [34] obtained a set of Nash equilibria by applying a principle of repeated games with a fixed authority allocation for two-team wing design. For the worst-case design, which better characterizes a non-cooperative design, the average discrepancies amounted to 11% and 16% for the wing drag coefficient and specific wing weight as objective functions correspondingly. Lewis and Mistree [38] analyzed a two-team design of an aircraft. The average difference for six metric values of design goals between the non-cooperative case and the case of full cooperation amounted to 13%. Literature results indicate that the discrepancies between the global optimum design and Nash equilibrium design have significant values ranging from 11% to 84%. Although the referenced studies are scant and have different methodologies, they warrant further investigation of this question.

Given a broad application of game theory in different fields of science, including economics, it is also interesting to surface some criticism that game theory has received in this field and discuss how the criticism applies to the solutions in design optimization. For the reference, I use a recent review of Guerrien on the current state of game theory [54] and focus on two main criticism points: Nash equilibrium is not self-evident for rational players, otherwise they would play it every time; and game theory cannot describe any type of player's behavior. These points can suggest that Nash equilibrium is not an appropriate concept for the definition of an optimal design outcome in a decentralized design process settings.

The result of the decentralized design process is a consensus of design decisions made by disciplinary engineers. The game-theoretic formulation of the decentralized design optimization problem (Eq. 2, lines 1 and 2) does not limit the outcome of the optimization to Nash equilibrium only. Other equilibrium formulations are possible. In author's point of view, in the first approximation it is reasonable to assume that in a decentralized design process a consensus that encourages every discipline designer to maximize his own objective function (within the constraints applied by a similar optimization strategy of other discipline designers) can be called "an optimum". This consensus is also a general definition of the Nash equilibrium, and the formulation of the optimization problem assumes so (Eq. 2, lines 4 and 5). This definition helps finding optimal outcomes for the decentralized architecture of the design process, similar to Pareto-optimal solutions with the centralized design process. The assumptions embedded in the search for Nash equilibrium – such as no cooperation and perfect information – does simplify its notion and render the result as somewhat idealized. As mentioned in Section II.3, it is an asymptotic case. If one wishes that the result of optimization with decentralized designers coincided with absolute optimum (Pareto solution), it is necessary to adjust the definition of objective functions for designers with incentives for Pareto-optimal outcomes, assumptions about cooperation, and available information. These tasks would not only optimize technical aspects of the system, but would also optimize the design process aspects related to the interaction between disciplinary designers. These design process aspects could be implemented as requirements – along with technical requirements to system metrics and design parameters – during the detail design stage. With these preliminary comments, two criticism points raised in literature against Nash optimization are discussed below.

Many studies implicitly assumed that the Nash equilibrium is an obvious way to play. In other words, it is not self-evident for rational players how to reach the Nash equilibrium. This criticism can be also applied to the design process. There, the designers may not have the necessary information that they would use to adjust their behavior and reach an equilibrium outcome. In this case, the designers may not play the design game that would result in a Nash solution because it is not obvious for them how to find Nash design. However, this criticism point does not make Nash equilibrium less important or less optimal, even if it is not evident for players how to achieve it. The task of a system engineer that performs Nash optimization before the detail design is to develop procedural recommendations for the design process that would help discipline experts reach the consensus, preferably as close to the Pareto-optimal solution as possible. Another criticism point is attributed to the assumption of game theory that any behavior can be explained mathematically. In design optimization, this assumption also means that any act of rational designers in a design process can be explained with mathematical modeling. Section II.2.3 provides initial comments on the problem of formulating the objective function for each designer. Indeed, expressing the objective function, which mirrors the preference of discipline designers, is a challenge that requires careful evaluation, especially because the designers use these objective functions to reach a consensus in the form of Nash equilibrium.

These considerations suggest that the employment of Nash equilibrium in design optimization as an optimum outcome is a reasonable concept and not a problem as such. The challenge is in careful modelling of objective functions, cooperation process, and availability of information to converge the outcomes of Nash optimization as close to the Pareto solution as possible.

The last aspect of the results that is interesting to comment is the values of design parameters for Pareto-optimal and Nash front solutions. In Fig. 7, the Nash front is located between power outputs 15 kW and 20 kW. The Pareto-optimal values of the stroke for this range are constant and equal − independent to the design architecture − to the maximum available stroke value 0.11 m. The bore values increase from 0.12 m to 0.2 m for higher values of power output and lower affordability. The hot and cold temperatures are unchanged and equal to the minimum available values of 600 K and 301 K, respectively. The affordability takes the highest value for given bore and stroke when hot and cold temperatures are the lowest. The cost of the heater depends on the area of the heat exchange and the value of hot temperature (Eq. 34, Appendix). The area of the heat exchanger also depends on the magnitude of hot temperature: the higher the hot temperature, the larger the heat flow from the heat source. More substantial heat flow requires a bigger surface of the heater to accept the heat flow and avoid overheating. Also, the higher the hot temperature, the larger the exponential component in the cost equation that characterizes the more expensive materials resistant to temperature. Therefore, a lower cost of the heater can be achieved at lower hot temperatures. The cost equation for the cooler (Eq. 37, Appendix) contains only the dependence on the surface area. Higher cooling temperature implies higher rejected heat flow and a larger cooler area, which increases its cost. If the cooler temperature is lower, then the cost of the cooler is lower as well. Therefore, the designs with the most affordable heat exchangers are the designs with the lowest hot and cold temperatures. The source of power maximization when temperatures are minimal is the increase of the system working volume, which can be achieved through the increase of bore and stroke. For the Pareto-optimal designs, the stroke takes a maximum available value of 0.11 m, and the bore increases from 0.12 m to 0.2 m with increasing power output from 15 kW to 20 kW and the reducing affordability of heat exchanger from 0.19 to 0.14. Due to sub-optimality of Nash front solutions, the values of design parameters in the game settings would be selected against the discussed above logic to maximize the system power and affordability of heat exchangers. Based on the example discussed above, the strategy for the selection of hot and cold temperatures is minimization and for the bore and stroke – maximization. These strategies can be communicated to design teams before the commencement of the detail design process as an additional measure to minimize the difference between real and intended Pareto-optimal system metrics.

IV.2 Design of energy conversion technologies

The examples of alternative studies referenced in Section IV.1 prove that the question of interplay between disciplinary designers is not limited to Stirling systems. It is obvious, that the methodology and results of this chapter could be applied in the commercial development of other energy conversion systems.

IV.3 Novelty and Contribution

- Developed a formal approach to apply game theory in the process of engineering design to account for the influence of disciplinary designers on results of detail design outcomes;
- Developed the notion of Nash front as an alternative concept to the Pareto front, which accounts for the influence of design authority reallocation in a multidisciplinary design process;
- Calculated the difference between Pareto-efficient solutions and Nash-efficient solutions for a two-team design process for the Stirling machine;
- Proposed an algorithm to apply the game theoretic framework to the design of other energy conversion systems apart from Stirling machines.

V CONCLUSIONS

- One of problematic questions in choosing between alternative design decisions during the detail design stage is how to reach a design consensus between disciplinary experts in a design process of a multi-disciplinary system. This chapter offered a method to reduce negative implications of this problem for the system design outcome;
- This chapter proposed a unified approach for the multi-objective design optimization using game theory for collaborative design at a detail design stage. This approach accounts for situations when discipline designers may choose to commit to either global or discipline-local optimum solutions. This design methodology applies game theoretic concepts to inform and improve multi-objective design optimization and collaborative design process;
- A general mathematical formulation of the approach was proposed for N discipline designers. The notion of Nash front was introduced as a set of design solutions consisting of Nash equilibria derived based on different design authority allocation between discipline designers;
- A case study of the Stirling engine-based m-CHP design optimization was studied with the application of the novel approach for design with two discipline teams. Five different cases of the design authority allocation to teams were studied. The average difference in m-CHP capital cost and power output between Pareto-optimal solutions and Nash front solutions was 22% and 16% correspondingly;
- This chapter provides the formulation of the limit case of a design optimization problem, where strict assumptions on the game theory setting were assumed, such as the absence of cooperation and non-repeated design process. It was intended that these assumptions define boundary conditions, thus identifying the asymptotic limit bounds (the Nash front and the Pareto front) of all possible design solutions;

- Future work will entail the exploration of extended approaches where cooperating designers' coalitions, as well as repeated-game design processes, are considered. Relaxing the assumptions on the game theoretical formulation will lead to designs that closely correspond to realistic design settings. Non-quantifiable parameters such as regulatory constraints, hidden agendas, and other organizational considerations will still affect the outcomes of the design process. By moving beyond the simple global optimization approach, the authors believe new insights will be developed in design processes in many engineering domains;
- Several practical recommendations can be proposed for detail design stages in future based on findings in this chapter. The implication of these recommendations can be a reduced difference between the real design outcome and the Pareto-optimal target:
    a) In a joint meeting between disciplinary designers, each expert should describe his objective function, controlled design decisions and the strategy to maximize the objective function. This discussion would surface conflicts between design parameters and help to plan a joint strategy for maximizing conflicting objectives;
    b) The design process policy should motivate designers to maximize their own objective function, but receive higher incentive for penalizing their own function if it can lead to a larger gain for the objective function of other discipline experts;
    c) Before the commencement of the design process, it is important to run the gamespace simulation for the system model with a different number of design decision alternatives to understand how this variable would affect the divergence from the Pareto-front. If the divergence increases, the designers should have lesser options for design parameter values;

d) The design authority should be distributed between designers equally.

- The developed design methodology can be applied to manage the detail design stage for commercial development of other energy conversion systems, apart from Stirling machines.

APPENDIX

MICRO-CHP SYSTEM MODEL EQUATIONS AND VALIDATION

This Appendix reports the equations for the system model of a micro heat and power plant based on the Stirling engine and results of the model validation.

A1. THERMODYNAMIC SUB-MODEL

The equations were adapted from Martini [43]. Swept volume in expansion space V_{L} and in compression space V_{K}, m^3:

$$V_{\mathrm{L}} = s \cdot \frac{\pi \cdot b^2}{4} \tag{3}$$

$$V_{\mathrm{K}} = V_{\mathrm{L}} \tag{4}$$

Instantaneous hot volume H and instantaneous cold volume C, m^3:

$$H = \frac{V_{\mathrm{L}}}{2} \cdot \left[1 - \sin\left(\theta\right)\right] + H_{\mathrm{D}} \tag{5}$$

$$C = \frac{V_{\mathrm{K}}}{2} \cdot \left[1 - \sin\left(\theta - \varphi\right)\right] + C_{\mathrm{D}} \tag{6}$$

Where $\theta \in \left[0; 2\pi\right]$ is the shaft crank angle, H_{D} is a dead volume in the hot volume, φ is a phase angle, and C_{D} is a dead volume in the cold volume.

Total gas volume V, m^3:

$$V = H + C + R_{\mathrm{D}} \tag{7}$$

Where R_{D} is a regenerator (dead) volume.

Using temperatures of the hot and cold spaces, T_{h} and T_{c} correspondingly in K, the formula for the regenerator temperature T_{r}, K reads:

$$T_{\mathrm{r}} = \left(T_{\mathrm{h}} + T_{\mathrm{c}}\right) / 2 \tag{8}$$

Number of cycles per second f , Hz:

$$f = N/n_{\mathrm{s}} \tag{9}$$

Where N is a number of revolutions per second and n_{s} is a number of revolutions per stroke.

Mean effective pressure p_{m} , Pa:

$$p_{\mathrm{m}} = P^{\mathrm{r}} / \left(f \cdot B^{\mathrm{r}} \cdot V_{\mathrm{L}} \right) \tag{10}$$

Where P^{r} and B^{r} is a power output and performance parameter (Beal number) for a reference micro-CHP Stirling engine.

Engine gas inventory M , moles:

$$M = \frac{P_{\mathrm{m}} \cdot V_{\mathrm{m}}}{R} \cdot \frac{\ln\left(T_{\mathrm{h}}/T_{\mathrm{c}}\right)}{\left(T_{\mathrm{h}} - T_{\mathrm{c}}\right)} \tag{11}$$

Where V_{m} is a mean volume and R is a gas constant for the working gas.

Instantaneous working gas pressure p , Pa:

$$p = \frac{M \cdot R}{H/T_{\mathrm{h}} + C/T_{\mathrm{c}} + R/T_{\mathrm{r}}} \tag{12}$$

Cycle work W (calculated for $p = f\left(V\right)$ as the area inside the closed curve), J:

$$W = \left(\int_{V\max}^{V\min} pdV \right)_{\text{expansion}} - \left(\int_{V\min}^{V\max} pdV \right)_{\text{compression}} \tag{13}$$

Swept volume ration K :

$$K = V_{\mathrm{K}} / V_{\mathrm{L}} \tag{14}$$

Total dead volume V_{D} , m^3 :

$$V_{\mathrm{D}} = H_{\mathrm{D}} + R + C_{\mathrm{D}} \tag{15}$$

Schmidt’s work equation parameters:

$$R_{\mathrm{V}}=V_{\mathrm{D}}/V_{\mathrm{L}};\ A_{\mathrm{U}}=T_{\mathrm{c}}/T_{\mathrm{h}};\ V_{\mathrm{T}}=(1+K)V_{\mathrm{L}};\ S=2R_{\mathrm{V}}A_{\mathrm{U}}/(A_{\mathrm{U}}+1);$$
$$E_{\mathrm{T}}=\arctan\left[\frac{K\sin(\varphi)}{A_{\mathrm{U}}+K\cos(\varphi)}\right];\ D_{\mathrm{L}}=\frac{\sqrt{A_{\mathrm{U}}^2+2A_{\mathrm{U}}K\cos(\varphi)+K^2}}{A_{\mathrm{U}}+K+2S} \tag{16}$$

Schmidt's equation for cycle work W_{S}, J:

$$W_{\mathrm{S}}=p_{\max}V_{\mathrm{T}}\cdot\frac{\pi(1-A_{\mathrm{U}})}{(K+1)}\cdot\sqrt{\frac{1-D_{\mathrm{L}}}{1+D_{\mathrm{L}}}}\cdot\frac{D_{\mathrm{L}}\sin(E_{\mathrm{T}})}{1+\sqrt{1-D_{\mathrm{L}}^2}} \tag{17}$$

Where $p_{\max}$ is the maximum instantaneous pressure in the cycle.

Meijer parameters S_{M} and D_{LM}:

$$S_{\mathrm{M}}=\frac{V_{\mathrm{D}}T_{\mathrm{c}}}{V_{\mathrm{L}}T_{\mathrm{r}}} \tag{18}$$

$$D_{\mathrm{LM}}=\frac{\sqrt{A_{\mathrm{U}}^2+2A_{\mathrm{U}}K\cos(\varphi)+K^2}}{A_{\mathrm{U}}+K+2S_{\mathrm{M}}} \tag{19}$$

Schmidt's equation for cycle work with Meijer parameter W_{SM}, J:

$$W_{\mathrm{SM}}=p_{\max}V_{\mathrm{T}}\cdot\frac{\pi(1-A_{\mathrm{U}})}{(K+1)}\cdot\sqrt{\frac{1-D_{\mathrm{LM}}}{1+D_{\mathrm{LM}}}}\cdot\frac{D_{\mathrm{LM}}\sin(E_{\mathrm{T}})}{1+\sqrt{1-D_{\mathrm{LM}}^2}} \tag{20}$$

Finkelstein' equation for cycle work W_{F}, J:

$$W_{\mathrm{F}}=\frac{2\pi K(1-A_{\mathrm{U}})\sin(\varphi)MRT_{\mathrm{c}}}{(A_{\mathrm{U}}+K+2S_{\mathrm{M}})^2\sqrt{1-D_{\mathrm{LM}}^2}\left(1+\sqrt{1-D_{\mathrm{LM}}^2}\right)} \tag{21}$$

Cycle work $\bar{W}$, J:

$$\bar{W}=median(W,W_{\mathrm{S}},W_{\mathrm{SM}},W_{\mathrm{F}}) \tag{22}$$

Heat input W_{in}, J:

$$W_{\mathrm{in}}=MRT_{\mathrm{h}}\ln\left(\frac{V_{\max}}{V_{\min}}\right)+MC_{\mathrm{v}}(T_{\mathrm{h}}-T_{\mathrm{r}}) \tag{23}$$

Where $V_{\max}$ and $V_{\min}$ are maximum and minimum instantaneous volumes, and C_{v} isochoric heat capacity of the working gas.

Power output P (adjusted to match the power output of the reference engine [9]), W:

$$P = \overline{W} \cdot \frac{f}{a} \tag{24}$$

Where a is the reference engine correction parameter. The value of parameter a is derived after the validation of the model (Table A2) to minimize the error and match the power output of the reference engine.

A2. HEAT EXCHANGER SUB-MODEL

Piston speed u_{p}, $\mathrm{m \cdot s^{-1}}$

$$u_{\mathrm{p}} = 2 \cdot s \cdot \frac{f}{a} \tag{25}$$

Heat transfer coefficient for heater α_{h}, $\mathrm{J \cdot kg^{-1} \cdot K^{-1}}$ [45]:

$$\alpha_{\mathrm{h}} = 0.042 \cdot b^{-0.42} \cdot u_{\mathrm{p}}^{0.58} \cdot p_{\mathrm{m}}^{0.58} \cdot T_{\mathrm{h}}^{-0.19} \tag{26}$$

Heat transfer coefficient for cooler α_{c}, $\mathrm{J \cdot kg^{-1} \cdot K^{-1}}$ [45]:

$$\alpha_{\mathrm{c}} = 0.0236 \cdot b^{-0.47} \cdot u_{\mathrm{p}}^{0.53} \cdot p_{\mathrm{m}}^{0.53} \cdot T_{\mathrm{c}}^{-0.11} \tag{27}$$

Power input to the cycle P_{in}, W:

$$P_{\mathrm{in}} = W_{\mathrm{in}} \cdot \frac{f}{a} \tag{28}$$

Wetted surface area of the heater $A_{\mathrm{h}}^{\mathrm{w}}$, m^2:

$$A_{\mathrm{h}}^{\mathrm{w}} = \frac{P_{\mathrm{in}}}{\alpha_{\mathrm{h}} \cdot \left(T_{\mathrm{h}} - T_{\mathrm{c}}\right)} \tag{29}$$

Amount of heat rejected from the cycle W_{out}, J:

$$W_{\mathrm{out}} = MRT_{\mathrm{c}} \ln\left(\frac{V_{\max}}{V_{\min}}\right) \tag{30}$$

Power rejected from the cycle P_{out}:

$$P_{\mathrm{out}} = W_{\mathrm{out}} \cdot \frac{f}{a} \tag{31}$$

Wetted surface area of the cooler $A_{\mathrm{c}}^{\mathrm{w}}$, m^2:

$$A_{\mathrm{c}}^{\mathrm{w}} = \frac{P_{\mathrm{out}}}{\alpha_{\mathrm{c}} \cdot \left(T_{\mathrm{h}} - T_{\mathrm{c}}\right)} \tag{32}$$

Wetted surface area in the regenerator $A_{\mathrm{r}}^{\mathrm{w}}$, m^2 [55]:

$$A_{\mathrm{r}}^{\mathrm{w}} = 735.239 \cdot V_{\mathrm{L}}^{2/3} \tag{33}$$

A3. SYSTEM COST SUB-MODEL

Equations were adapted from the work of Garcia *et al*. [55].

Bulk engine cost C_{e}, \$:

$$C_{\mathrm{e}} = C_{\mathrm{e}}^{\mathrm{r}} \cdot \left[V_{\mathrm{L}}^{\mathrm{r}} \cdot \left(\frac{V_{\mathrm{L}}}{V_{\mathrm{L}}^{\mathrm{r}}} \right)^{0.2} \cdot p_{\mathrm{m}}^{\mathrm{r}} \cdot \left(\frac{p_{\mathrm{m}}}{p_{\mathrm{m}}^{\mathrm{r}}} \right)^{0.2} \right] \tag{34}$$

Where $C_{\mathrm{e}}^{\mathrm{r}}$, $V_{\mathrm{L}}^{\mathrm{r}}$, $p_{\mathrm{m}}^{\mathrm{r}}$ are bulk cost, swept volume and mean pressure for the reference engine correspondingly.

Heater cost C_{h}, \$:

$$C_{\mathrm{h}} = C_{\mathrm{h1}}^{\mathrm{r}} \cdot A_{\mathrm{h}}^{\mathrm{wr}} \left[\left(\frac{A_{\mathrm{h}}^{\mathrm{w}}}{A_{\mathrm{h}}^{\mathrm{wr}}} \right)^{0.5} \cdot \frac{1}{2} \cdot \left(1 + \mathrm{e}^{C_{\mathrm{h2}}^{\mathrm{r}}\left(T_{\mathrm{h}} - T_{\mathrm{h}}^{\mathrm{r}}\right)} \right) \right] \tag{35}$$

Where $C_{\mathrm{h1}}^{\mathrm{r}}$, $A_{\mathrm{h}}^{\mathrm{wr}}$ and $T_{\mathrm{h}}^{\mathrm{r}}$ are heater cost, heater wetted area and heating temperature for the reference system, $C_{\mathrm{h2}}^{\mathrm{r}}$ is the cost parameter in $\$ \cdot \mathrm{K}^{-1}$ for the reference system.

Regenerator cost C_{r}, \$:

$$C_{\mathrm{r}} = C_{\mathrm{r1}}^{\mathrm{r}} \cdot A_{\mathrm{r}}^{\mathrm{wr}} \left[\left(\frac{A_{\mathrm{r}}^{\mathrm{w}}}{A_{\mathrm{r}}^{\mathrm{wr}}} \right)^{0.6} \cdot \frac{1}{2} \cdot \left(1 + \mathrm{e}^{C_{\mathrm{r2}}^{\mathrm{r}}\left(T_{\mathrm{r}} - T_{\mathrm{r}}^{\mathrm{r}}\right)} \right) \right] \tag{36}$$

Where $C_{\mathrm{r1}}^{\mathrm{r}}$, $A_{\mathrm{r}}^{\mathrm{wr}}$ and $T_{\mathrm{r}}^{\mathrm{r}}$ are regenerator cost, wetted area and temperature for the reference system correspondingly, $C_{\mathrm{r2}}^{\mathrm{r}}$ is the cost parameter in $\$ \cdot \mathrm{K}^{-1}$ for the reference system.

Cooler cost C_{c}, \$:

$$C_{\mathrm{c}} = C_{\mathrm{c}}^{\mathrm{r}} \cdot A_{\mathrm{c}}^{\mathrm{wr}} \left(\frac{A_{\mathrm{r}}^{\mathrm{w}}}{A_{\mathrm{r}}^{\mathrm{wr}}} \right)^{0.4} \quad (37)$$

Where $C_{\mathrm{r}}^{\mathrm{r}}$ and $A_{\mathrm{c}}^{\mathrm{wr}}$ are cooler cost and wetted area for the reference system correspondingly.

Dimensionless heat exchangers affordability metric A:

$$A = \frac{1}{C_{\mathrm{h}} + C_{\mathrm{r}} + C_{\mathrm{c}}} \Big/ \max\left(A\right) \quad (38)$$

Where $\max\left(A\right)$ is the highest affordability among all evaluated system architectures.

Table 5. Constant parameters of the system model.

Parameter	**Unit**	**Value**
H_{D}	m^3	$63.4 \cdot 10^{-6}$
φ	rad	1.571
C_{D}	m^3	$36.9 \cdot 10^{-6}$
R_{D}	m^3	$68.7 \cdot 10^{-6}$
N	rps	25
n_{s}	rps, “s” – stroke	1
P^{r}	W	7500
B^{r}	-	0.01715
R	$\mathrm{J} \cdot \mathrm{g\text{-}mol}^{-1} \cdot \mathrm{K}^{-1}$	8.314
C_{v}	$\mathrm{J} \cdot \mathrm{K}^{-1} \cdot \mathrm{mol}^{-1}$	12.471
a	-	2.675
$C_{\mathrm{e}}^{\mathrm{r}}$	$\$ \cdot \mathrm{Pa}^{-1} \cdot \mathrm{m}^{-3}$	4.99
$V_{\mathrm{L}}^{\mathrm{r}}$	m^3	$194.55 \cdot 10^{-6}$
$p_{\mathrm{m}}^{\mathrm{r}}$	Pa	$9.5 \cdot 10^{6}$
$C_{\mathrm{h1}}^{\mathrm{r}}$	$\$ \cdot \mathrm{m}^{-2}$	50480
$A_{\mathrm{h}}^{\mathrm{wr}}$	m^2	0.14
$T_{\mathrm{h}}^{\mathrm{r}}$	K	898
$C_{\mathrm{h2}}^{\mathrm{r}}$	$\$ \cdot \mathrm{K}^{-1}$	0.02
$C_{\mathrm{r1}}^{\mathrm{r}}$,	$\$ \cdot \mathrm{m}^{-2}$	2329
$A_{\mathrm{r}}^{\mathrm{wr}}$	m^2	2.47
$T_{\mathrm{r}}^{\mathrm{r}}$	K	600
$C_{\mathrm{r2}}^{\mathrm{r}}$	$\$ \cdot \mathrm{K}^{-1}$	0.02

A4. VALIDATION

Table 6. Model validation using the parameters of the Stirling engine V161. Adapted from [9].

Parameter	Unit	Model	Reference case, Stirling V161	Error, %
Swept volume	cm^3	194.66	194.55	<1%
Maximum pressure	MPa	15.03	15.00	<1%
Mean pressure	MPa	9.26	9.5	3%
Nominal electric output	kW	7.73	7.50	3%
Electrical efficiency	%	21.3	24.0	13%
Engine capital cost	$	23 354 (100%)	21 400 (100%)	8%
Engine bulk cost (share %)	$	9 179 (39.3%)	7 884 (36.8%)	14%
Heater cost	$	7 126 (30.5%)	6 089 (28.5%)	15%
Regenerator cost	$	4 112 (17.6%)	4 916 (23.0%)	20%
Cooler cost	$	2 937 (12.6%)	2 509 (11.7%)	15%

REFERENCES

[1] E. Crawley, B. Cameron, and D. Selva, *System architecture: strategy and product development for complex systems*, Prentice Hall: Pearson, 2015.

[2] C-L. Hwang and Abu Syed Md Masud, "Multiple objective decision making methods and applications: a state-of-the-art survey," *Springer Science & Business Media*, vol. 164, 2012.

[3] W. Tan, B.J. Sauser, J.E. Ramirez-Marquez and R.B. Magnaye, "Multiobjective optimization in multifunction multicapability system development planning," *IEEE Transactions on Systems, Man, and Cybernetics: Systems,* vol. 43, no. 4, pp.785-800, 2013.

[4] R. Datta, S. Pradhan and B. Bhattacharya, "Analysis and design optimization of a robotic gripper using multiobjective genetic algorithm," *IEEE Transactions on Systems, Man, and Cybernetics: Systems*, vol. 46, no. 1, pp.16-26, 2016.

[5] C.H. Chen and J.H. Chou, "Multiobjective optimization of airline crew roster recovery problems under disruption conditions," *IEEE Transactions on Systems, Man, and Cybernetics: Systems*, vol. 47, no. 1, pp.133-144, 2017.

[6] Y.R. Naidu and A.K. Ojha, "Solving multiobjective optimization problems using hybrid cooperative invasive weed optimization with multiple populations," *IEEE Transactions on Systems, Man, and Cybernetics: Systems*, vol. 48, no. 6, pp.821-832, 2018.

[7] W. Lakhdhar, R. Mzid, M. Khalgui, Z. Li, G. Frey, and A. Al-Ahmari, "Multiobjective optimization approach for a portable development of reconfigurable real-time systems: From specification to implementation," *IEEE Transactions on Systems, Man, and Cybernetics: Systems*, 2018.

[8] A. M. Ross and D. E. Hastings, "The tradespace exploration paradigm," *INCOSE Int. Symp*., vol. 15, no. 1, pp. 1706–1718, Jul., 2005.

[9] D. Smirnov and A. Golkar, “Stirling engine systems tradespace exploration framework,” *Procedia Comput. Sci.*, vol. 44, pp. 558–567, 2015.

[10] A. M. Ross, “Multi-attribute tradespace exploration with concurrent design as a value-centric framework for space system architecture and design,” Ph.D. dissertation, Massachusetts Institute of Technology, 2003.

[11] R. Alonso and N. Matouschek, “Optimal delegation,” *Rev. Econ. Stud.*, vol. 75, no. 1, pp. 259–293, 2008.

[12] M. Bristow, L. Fang, and K. W. Hipel, “Agent-Based Modeling of Competitive and Cooperative Behavior Under Conflict,” *IEEE Trans. Syst. Man Cybern. Syst.*, vol. 44, no. 7, pp. 834–850, Jul., 2014.

[13] E. Herrera-Viedma, F. Herrera, and F. Chiclana, “A consensus model for multiperson decision making with different preference structures,” *IEEE Trans. Syst. Man Cybern. - Part Syst. Hum.*, vol. 32, no. 3, pp. 394–402, May, 2002.

[14] M. A. Cardin, J. Yixin, H. K. H. Yue, and F. Haidong, “Training Design and Management of Flexible Engineering Systems: An Empirical Study Using Simulation Games,” *IEEE Trans. Syst. Man Cybern. Syst.*, vol. 45, no. 9, pp. 1268–1280, Sep., 2015.

[15] G. Bordogna, M. Fedrizzi, and G. Pasi, “A linguistic modeling of consensus in group decision making based on OWA operators,” *IEEE Trans. Syst. Man Cybern. - Part Syst. Hum.*, vol. 27, no. 1, pp. 126–133, Jan., 1997.

[16] D. Ben-Arieh and Z. Chen, “Linguistic-labels aggregation and consensus measure for autocratic decision making using group recommendations,” *IEEE Trans. Syst. Man Cybern. - Part Syst. Hum.*, vol. 36, no. 3, pp. 558–568, May, 2006.

[17] K. W. Li, K. W. Hipel, D. M. Kilgour, and L. Fang, “Preference uncertainty in the graph model for conflict resolution,” *IEEE Trans. Syst. Man Cybern. - Part Syst. Hum.*, vol. 34, no. 4, pp. 507–520, Jul., 2004.

[18] R. Mallubhatla, K. R. Pattipati, D. L. Kleinman, and Z. B. Tang, “A model of distributed team information processing under ambiguity,” *IEEE Trans. Syst. Man Cybern.*, vol. 21, no. 4, pp. 713–725, Jul., 1991.

[19] S. J. Kapurch, “NASA systems engineering handbook,” DIANE Publishing, 2010.

[20] T. W. Simpson and J. R. R. A. Martins, “Advancing the design of complex engineered systems through multidisciplinary design optimization: Report from an NSF workshop,” AIAA 2012-1585, 2012.

[21] A. Yassine, B. Maddah, and N. Nehme, “Optimal information exchange policies in integrated product development,” *IIE Trans.*, vol. 45, no. 12, pp. 1249–1262, Dec., 2013.

[22] V. Chanron and K. Lewis, “A study of convergence in decentralized design processes,” *Res. Eng. Des.*, vol. 16, no. 3, pp. 133–145, Dec., 2005.

[23] M. R. Danesh and Y. Jin, “An agent-based decision network for concurrent engineering design”, *Concurr. Eng.*, vol. 9, no. 1, pp. 37–47, Mar., 2001.

[24] J.-C. Pomerol and F. Adam, “Practical decision making – from the legacy of Herbert Simon to decision support systems,” *Decis. Support Uncertain Complex World IFIP TC8WG83 Int. Conf.*, 2004.

[25] J. Austin-Breneman, B. Y. Yu, and M. C. Yang, “Biased information passing between subsystems over time in complex system design,” *Journal of Mechanical Design*, vol. 138, no. 1, Aug. 2014.

[26] M. Kalsi, K. Hacker, and K. Lewis, “A comprehensive robust design approach for decision trade-offs in complex systems design,” *J. Mech. Des.*, vol. 123, no. 1, p. 1, 2001.

[27] A. Yassine and D. Braha, “Complex concurrent engineering and the design structure matrix method,” *Concurr. Eng.*, vol. 11, no. 3, pp. 165–176, Sep., 2003.

[28] F. Li, T. Wu, and M. Hu, "Design of a decentralized framework for collaborative product design using memetic algorithms," *Optim. Eng.*, vol. 15, no. 3, pp. 657–676, Sep., 2014.

[29] M. J, Osborne, "An introduction to game theory", vol. 3, no. 3, New York: Oxford university press, 2004.

[30] T. L. Vincent, "Game theory as a design tool," *J. Mech. Des.*, vol. 105, no. 2, pp. 165–170, 1983.

[31] M. Xiao, X. Shao, L. Gao, and Z. Luo, "A new methodology for multi-objective multidisciplinary design optimization problems based on game theory," *Expert Systems with Applications*, vol. 42 no. 3, pp.1602-1612, 2015.

[32] E. Holmberg, C.J. Thore, and A. Klarbring, "Game theory approach to robust topology optimization with uncertain loading," *Structural and Multidisciplinary Optimization*, vol. 55 no. 4, pp.1383-1397, 2017.

[33] J.A. Désidéri, R. Duvigneau, and A. Habbal, "Multiobjective Design Optimization using Nash Games," *Computational Intelligence in Aerospace Sciences, Progress in Astronautics and Aeronautics*, pp. 583-641, 2014.

[34] P. Della Vecchia, L. Stingo, S. Corcione, D. Ciliberti, F. Nicolosi, A. De Marco, and G. Nardone, "Game theory and evolutionary algorithms applied to MDO in the AGILE European project," *18th AIAA/ISSMO Multidisciplinary Analysis and Optimization Conference*, p. 4330, 2017.

[35] K. Han and A. Nehorai, "Jointly optimal design for MIMO radar frequency-hopping waveforms using game theory," *IEEE Transactions on Aerospace and Electronic Systems*, vol. 52, no. 2, pp.809-820, 2016.

[36] S.D. Vermillion and R.J. Malak, "A Game Theoretical Perspective on Incentivizing Collaboration in System Design," *Disciplinary Convergence in Systems Engineering Research*, pp. 845-855, 2018.

[37] Z. Sha, K.N. Kannan, and J.H. Panchal, "Behavioral experimentation and game theory in engineering systems design," *Journal of Mechanical Design*, vol. 137, no. 5, p.051405, 2015.

[38] K. Lewis and F. Mistree, "Collaborative, sequential, and isolated decisions in design," *Journal of Mechanical Design*, vol. 120, no. 4, pp.643-652, 1998.

[39] A. Xiao, S. Zeng, J.K. Allen, D.W. Rosen, and F. Mistree, "Collaborative multidisciplinary decision making using game theory and design capability indices," *Research in Engineering Design*, vol. 16, no. 1-2, pp.57-72, 2005.

[40] G.V. Bhatia, H. Kannan, and C.L. Bloebaum, "A Game Theory approach to Bargaining over Attributes of Complex Systems in the context of Value-Driven Design," *54th AIAA Aerospace Sciences Meeting*, p. 0972, 2016.

[41] C. Dutang, "Existence theorems for generalized Nash equilibrium problems: an analysis of assumptions," *Journal of Nonlinear Analysis and Optimization*, vol. 4, no. 2, pp. 115-126, 2016.

[42] *Micro-CHP technology assessment and benchmarking*, EPRI, Palo Alto, CA, 1018977, 2009.

[43] W. R. Martini, "Stirling engine design manual," 2nd edition, Jan., 1983.

[44] A.C. Ferreira, R.F. Oliveira, M.L. Nunes, L.B. Martins, and S.F. Teixeira, "Modelling and cost estimation of stirling engine for CHP applications," *Proceedings of the 2014 International Conference on Power Systems, Energy, Environment*, p. 978, 2014.

[45] F. Toda and S. Iwamoto, "Heat Transfer on a Small Stirling Engine," *Marine Engng Soc Japan Bull*, vol. 19, no. 1, March, 1991.

[46] K. Hölttä-Otto and O. de Weck, "Degree of Modularity in Engineering Systems and Products with Technical and Business Constraints," *Concurr. Eng.*, vol. 15, no. 2, pp. 113–126, Jun., 2007.

[47] F. C. Keil, “Constraints on knowledge and cognitive development,” *Psychol. Rev.*, vol. 88, no. 3, pp. 197–227, 1981.

[48] W. E. Steinmueller, “The role of technical standards in coordinating the division of labour in complex system industries. Part II., The Business of Systems Integration,” Oxford University Press, 2003.

[49] G. Kunda, “Engineering culture: Culture and control in a high-tech organization,” Ph.D. dissertation, Massachusetts Institute of Technology, 1986.

[50] W. Gruhl, “Lessons learned, cost/schedule assessment guide,” Internal presentation, NASA Comptroller’s office, 1992.

[51] A.K. Kludze, “The impact of systems engineering on complex systems,” *Conference on Systems Engineering Research*, 2004.

[52] R.S. Miller, D.R. Floricel, Lessard, “The strategic management of large engineering projects,” MIT Press, 2000.

[53] Z. Bu and H. Li and J. Cao and Z. Wang and G. Gao, "Dynamic Cluster Formation Game for Attributed Graph Clustering," *IEEE Transactions on Cybernetics*, vol. 49, no. 1, pp.328-341, 2019.

[54] Guerrien, B. “On the current state of game theory,” *Real-world economics review*, vol. 83, Wolrd Economics Association, p.35, 2018

[55] D. García, M.A. González, J.I. Prieto, S. Herrero, S. López, I. Mesonero, and C. Villasante, “Characterization of the power and efficiency of Stirling engine subsystems,” *Applied energy*, vol. 121, pp.51-63, 2014.

CHAPTER IV
BIBLIOMETRIC STUDY OF STIRLING REFRIGERATORS

After the detailed design was complete, and the components of the experimental system manufactured, the next step was the assembly and initial testing. I discussed in Section I.5 of Chapter I, how the break-in process for the mechanical drive of the Stirling machine led to the observation of the refrigeration effect. This stage of the project was crucial for the technological pathway proposed in this thesis. If this project were developed in the framework of scientific or engineering development to obtain the operational Stirling engine, then most likely, after the break-in tests, the activities would focus on achieving expected engine performance. However, we did not keep the focus on the engine application. Our project was commercial and was primarily concerned with developing a system with a high market value within allocated time and resources. Several engineering challenges still had to be solved to operate the designed Stirling machine as the engine. They included the optimization of the piston-cylinder seal and the system to transfer heat to the engine from the external source. After the observation of the refrigeration effect, in parallel with the initial tests, we started the evaluation of the refrigeration application. Higher market potential and finite project nature motivated and indeed led to the reorientation of the development focus towards the refrigeration system. To formulate the development strategy for the test and refinement stage, it was important to perform a similar analysis of the technical literature for the refrigeration application using big data approach.

I INTRODUCTION AND METHODOLOGY

The present chapter applies the methodology developed in Chapter II and reports on the results of analyzing scientific and patent literature using the big data approach. From the development point of view, it is essential to understand the scientific and patent landscape for Stirling refrigerators to improve the existing design of the Stirling refrigerator and find a technical niche, where the designed Stirling machine can be used. Typically, scientific and engineering literature treats the domain of refrigerators or heat pumps based on the Stirling cycle differently from its sibling area of Stirling engines. Why it happens is a fascinating question. From the one hand, the system operates a similar thermodynamic cycle. This point was clarified in Section I.2 of Chapter I. However, due to different operational conditions, the technical requirements to materials, heat exchangers and mechanical structure are different. This difference leads to a different optimal design. The general approach and limitations of the present method of study are discussed in detail in Section II of Chapter II. In this chapter, I use a different combination of search terms that reflects varying terminology in the field: *"Stirling AND (refrigerator OR cooler OR cryocooler OR "heat pump") AND NOT "Stirling engine""*. For the country analysis of Stirling refrigerators, I focused three countries – United States, Japan and China. Although the literature as such forms a different research field, the geographical regions that are leading in the development of the Stirling refrigerators are similar to those that advance in the field of Stirling engines.

II RESULTS

According to Scopus, during the period between 1960 and 2006 in the world was published 692 scientific documents about Stirling refrigerators. In the period between 2007 and 2018 the same number was 699 documents. This means that the last twelve years generated

50% of scientific knowledge about Stirling refrigerators referenced in Scopus over the last 58 years. Figure 1 depicts the publishing trends of scientific and patent documents over that time.

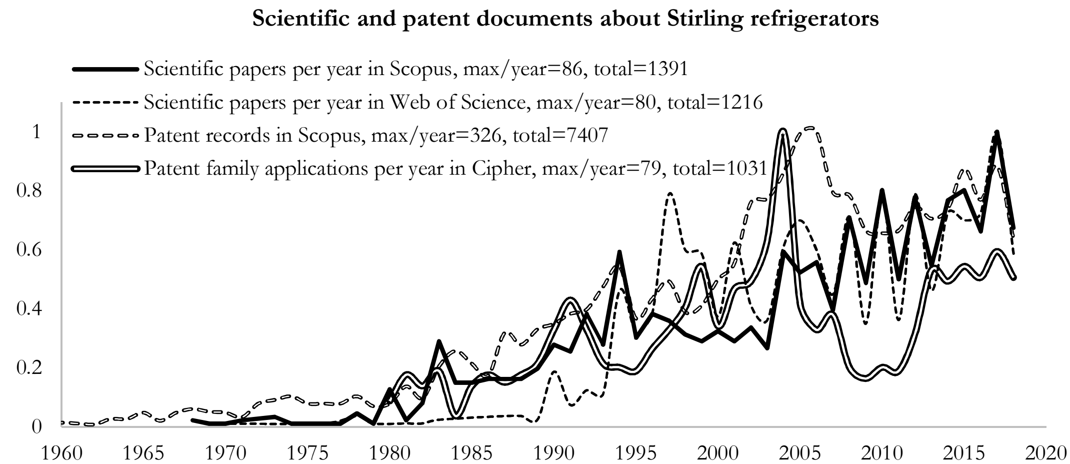


Fig. 1. Publishing activity for scientific papers and patent documents for the search term "Stirling refrigerator [etc.]" based on data from Scopus, Web of Science and Cipher for a period between 1960 and 2018. Absolute numbers are normalized by the maximum number of documents per year.

Figure 2 shows the affiliation of countries with published research for the two analyzed periods.

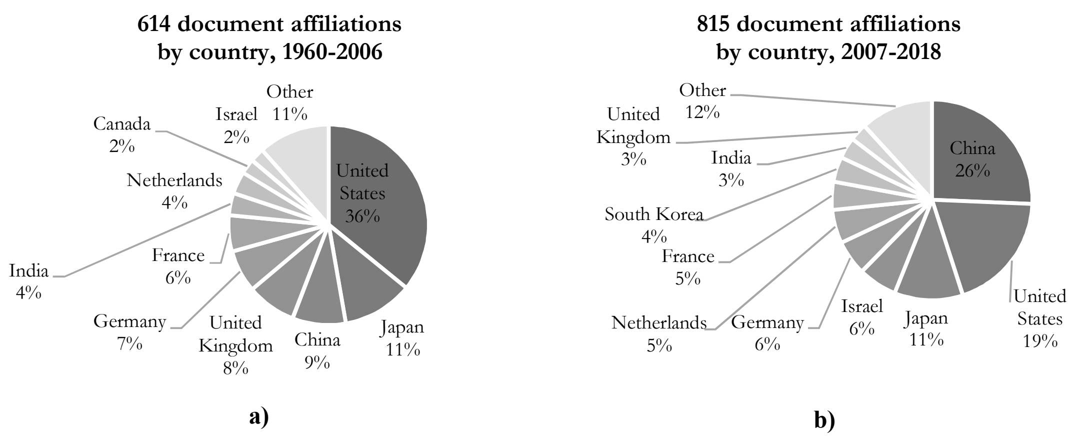


Fig. 2. Document affiliations by country in Scopus with identifiable countries of origin: a) 1960-2006, total 614 document affiliations; b) 2007-2018, total 815 document affiliations.

Figure 3 demonstrates in what sources the scientific research about Stirling engines was published.

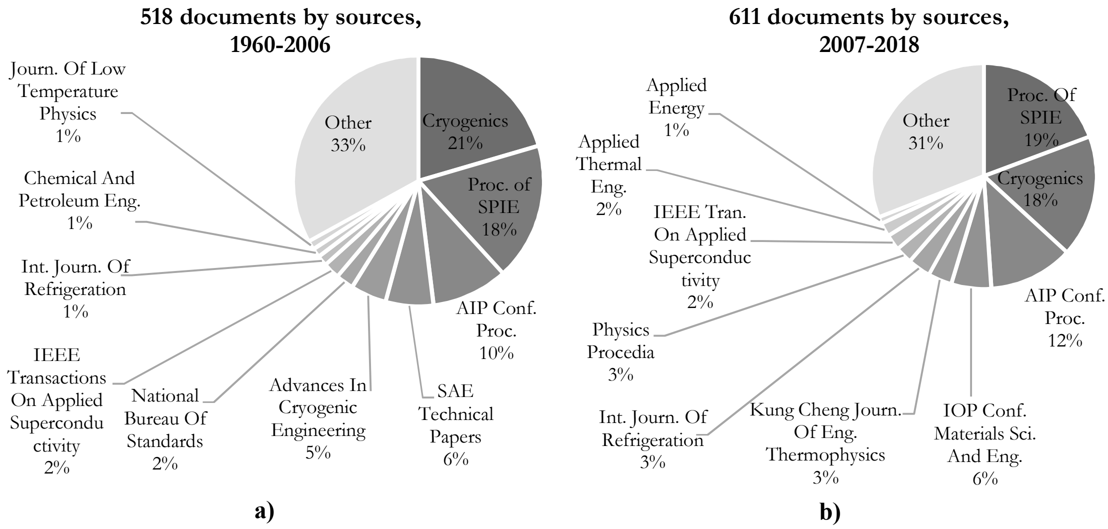


Fig. 3. Documents by source in Scopus with identifiable sources: a) 1960-2006, total 518 documents; b) 2007-2018 total 611 documents

Figure 4 shows the geographic distribution for the cumulative number of granted patents and patent expenditure since 2000.

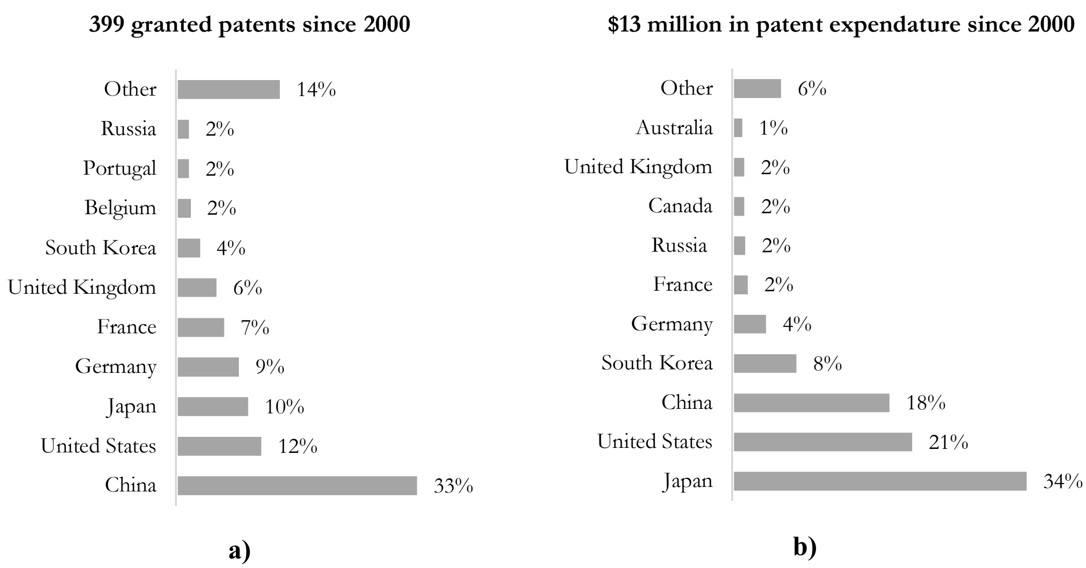
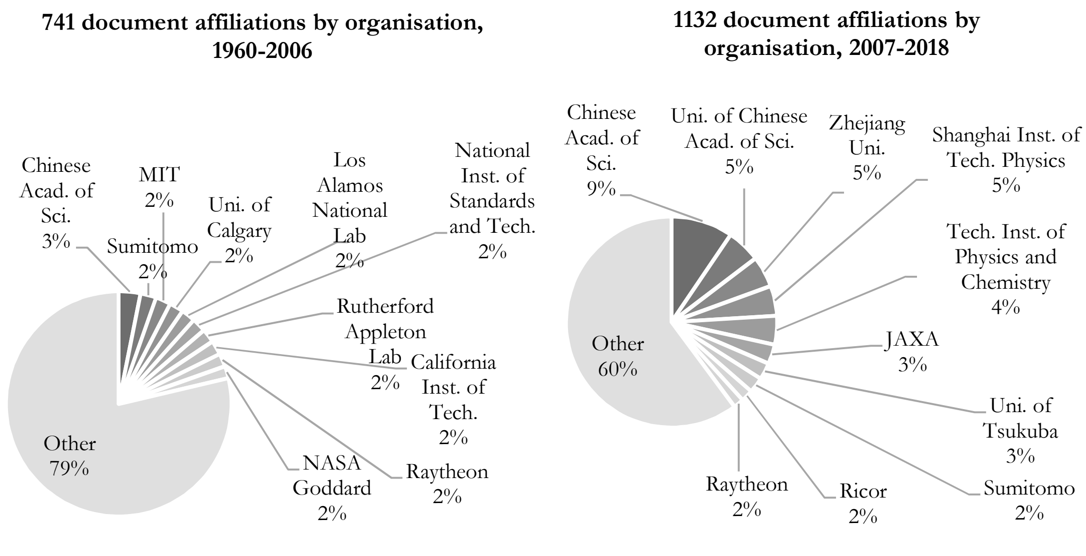


Fig. 4. Geograthic distribution data in Cipher in 2018, cummulative since 2000 for: a) 399 granted patents and b) $13 million in patent expendature.

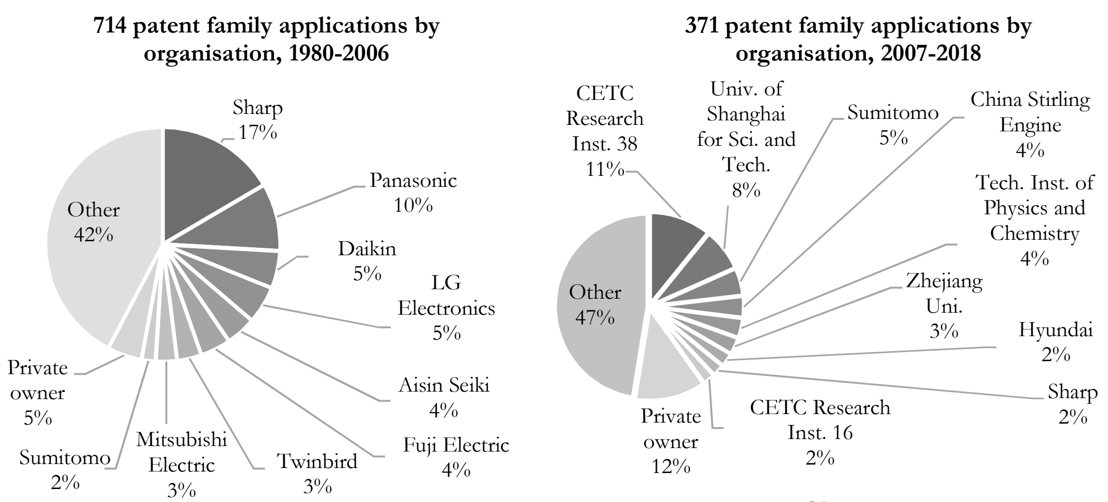
Figure 5 helps to look into the structure of organizations affiliated with 692 scientific documents published before 2007 and 699 documents for the period between 2007 and 2018.



a) b)

Fig. 5. Document affiliations by organization in Scopus: a) 1960-2006, total 741 document affiliations; b) 2007-2018, total 1132 document affiliations.

Figure 6 shows patent family applications by organization for the period between 1980 and 2006 and the period from 2007 to 2018.



a) b)

Fig. 6. Patent family applications by organization in Cipher: a) total 714 families, 1980-2006; b) 2007-2018, total 371 families.

II.1 UNITED STATES

According to Scopus, during the period between 1960 and 2006 in the United States were published 220 scientific documents with identifiable country of origin about Stirling engines. In the period between 2007 and 2018 the same number was 158 documents. Figure 7 shows the document affiliation for the two time periods.

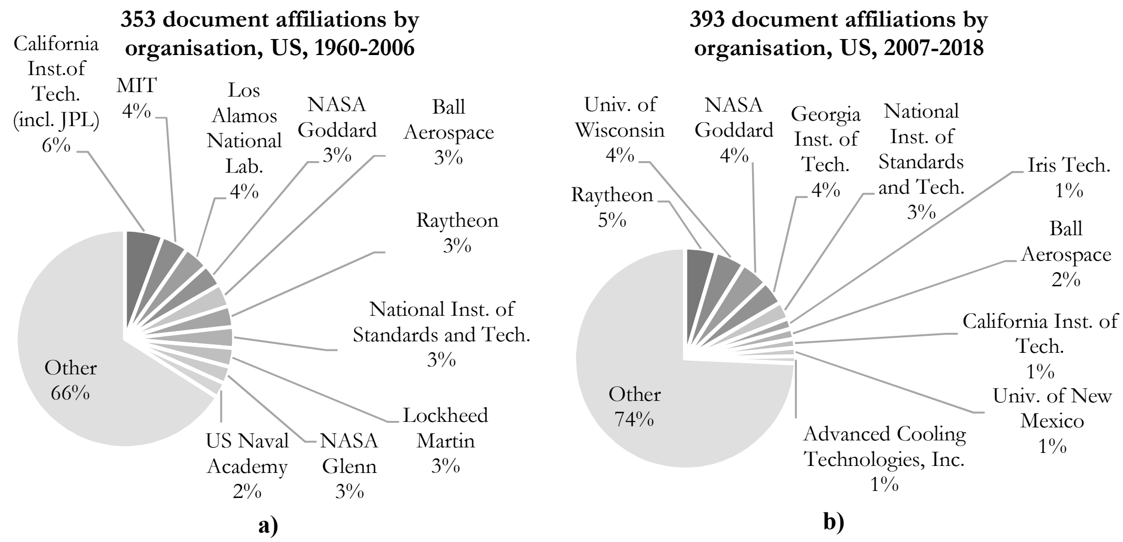


Fig. 7 Document affiliations in United States by organization in Scopus: a) 1960-2006, total 353 document affiliations; b) 2007-2018, total 393 document affiliations.

Between 1990 and 2006 California Institute of Technology and primarily Jet Propulsion Laboratory conducted a series of studies for different configurations of Stirling cryocoolers for cooling aerospace instruments. These studies started with the characterization of the Oxford-type Stirling cryocooler manufactured by British Aerospace [1] in the beginning of 1990s. Later studies investigated different configurations of cryocoolers with the aim to developed an advanced cryocooler for the temperature range of 4 to 10 K [2]. In the middle of 2000s the work analyzed the operation of Stirling cryocoolers in different applications to cool aerospace instruments, for example sapphire oscillator [3], tunable diode laser [4], or atmospheric infrared sounder [5]. Massachusetts Institute of Technology primarily focused on the development of two technologies – the superfluid Stirling refrigerator [6] and the cryocooler based on the

Collins cycle [7]. Los Alamos National Laboratory make advancements in two types of refrigerators – superfluid Stirling refrigerators [8] and pulse tube acoustic refrigerators [9], with some other studies to improve different elements of refrigerators. Research at NASA Goddard Space Flight Center focused mainly on studying the operation of Stirling cryocoolers as cooling devices for aerospace electronic instruments. This work described specific designs, system characteristics and compatibility of Stirling cryocoolers with different types of aerospace instruments, for example with the X-Ray spectrometer [10], the High Energy Solar Spectroscopic Imager [11], or the Alpha Magnetic Spectrometer [12]. Similar to NASA Goddard, Ball Aerospace primarily reported about the results of applying Stirling cryocooler for sensor electronics, such as the Germanium gamma-ray detector [13]. Several papers discussed the design and performance of the hybrid system to reach 4 K, which consists of Stirling and Joule-Thomson cryocoolers [14]. Raytheon focused on different aspects of Stirling cryocooler, including hybrid systems [15], entropy analysis in cryocoolers [16], and active temperature control [17].

After 2006, Raytheon put a strong focus on the properties of regenerators [18], the operation of hybrid systems [19] and the improvements of pulse tube cryocoolers [20]. University of Wisconsin Madison systematically studied the pulse tube cryocoolers [21] and the application of cryocoolers in electronic instruments [22]. NASA Goddard exclusively reported the results of operating Stirling cryocoolers within orbital instrument systems, including The Astro-H Soft X-ray Spectrometer [23], the Thermal Infrared Sensor [24] and Single photon HgCdTe avalanche photodiode [25]. Georgia Institute of Technology focused on research related to regenerators [26], development of a miniature Stirling cryocooler for a satellite [27] and pulse tube cryocoolers [28]. Figure 8 shows the distribution of research topics in the sample of documents published by leading organizations in each corresponding time period.

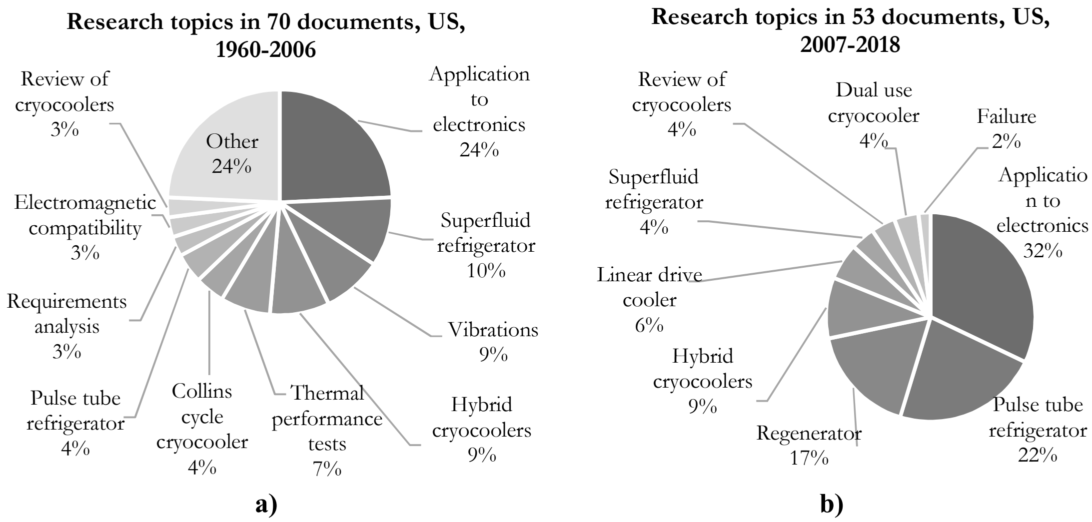


Fig. 8. Distribution of research topics in scientific documents by top publishing organizations in United States: a) 1966-2006, 70 analyzed documents from 220 total b) 2007-2018, 53 analyzed documents from 158 total.

Figure 8b shows that the Stirling cryocooler has reached maturity for space electronics applications. Figure 9 analyzes the ownership structure of active granted patent families in 2018 and indicates organizations that defined the growth in granted patent families over the last decade.

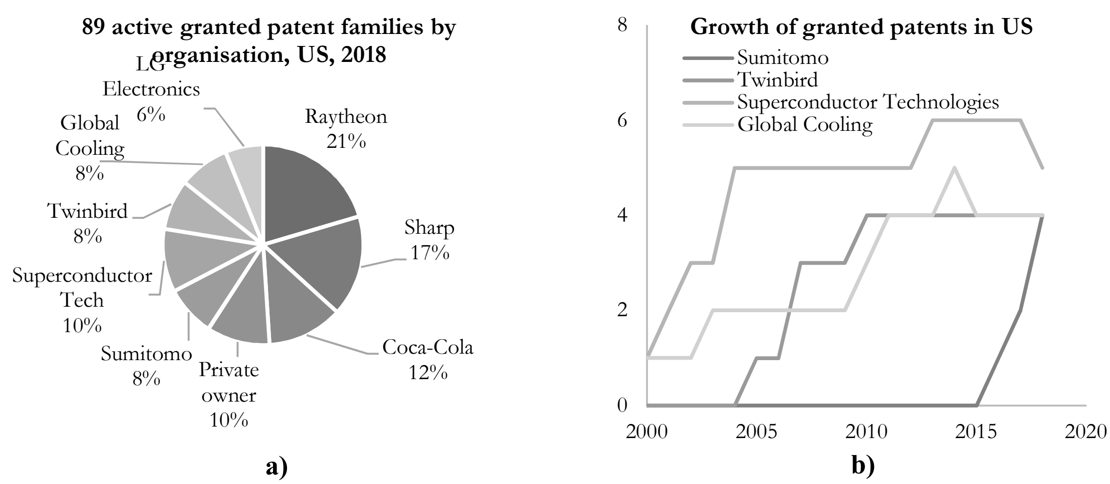


Fig. 9. Granted patent families in United States: a) distribution of 89 active granted patent families by organization, active in 2018 b) Organizations that defined growth in granted patent families in US over the last two decades.

Superconductor Technologies has developed a proprietary design of a Stirling cryocooler [29] to cool high temperature superconductor devices. In addition the company also patented the digital control and the regenerator material. Earlier patents of Twinbird in US included designs for the cold container cooled by the Stirling refrigerator [30], the coolant loop that removes cold from the container and the design of heat-exchanging external fins. In 2010 the company patented together with Global Cooling an original Stirling refrigerator design with the free-piston beta configuration [31]. Global Cooling protected an original design for a cabinet-type refrigeration system owned jointly with The Coca-Cola Company [32], the balancing system and has developed the free-piston gamma configuration with improved stability, efficiency and control [33]. Recent patents of Sumitomo Heavy Industries protected inventions for two types of Stirling refrigerators – the pulse tube [34] and beta-type [35].

II.2 JAPAN

According to Scopus, during the period between 1960 and 2006 in Japan were published 70 scientific documents with identifiable country of origin about Stirling refrigerators, 3.1 times less than in United States. In the period between 2007 and 2018 the same number was 90 documents, or 1.7 times less than in US. Figure 10 shows the document affiliation for the two time periods.

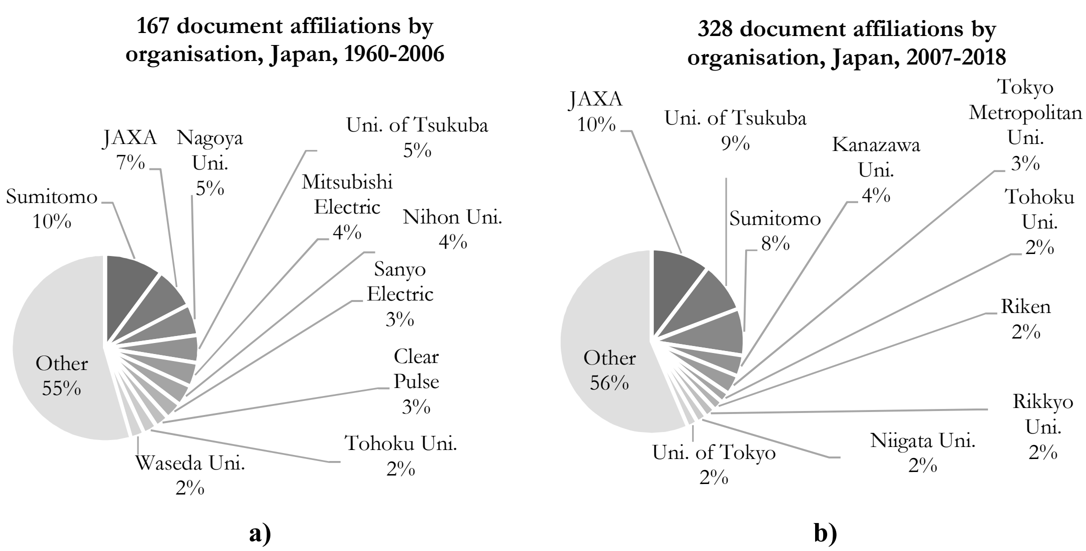


Fig. 10. Document affiliations in Japan by organization in Scopus: a) 1960-2006, total 167 document affiliations; b) 2007-2018, total 328 document affiliations.

Before 2007, Sumimoto Heavy Industries reported primarily the system integration studies for Stirling cryocoolers for aerospace instruments [36], some studies discussed hybrid systems [37] a two-stage [38] and a single stage [39] mechanical cryocoolers. Sumimoto co-authored most of their published papers with Japan Aerospace Exploration Agency and University of Tsukuba with the focus on applying Stirling refrigerators for cooling aerospace instruments. Nagoya University covered several research questions, including the design of thermoacoustic coolers [40] and regenerator materials [41]. Mitsubishi Electric worked on the development of a reliable Stirling cryocooler [42], tested a free-piston design of the Vuilleurmier cooler [43] and reported results of operating Stirling cryocoolers as elements of space instrument electronics [44].

After 2006, JAXA reported abundant results on the operation of cryosystems for cooling aerospace instruments [45], [46] a few studies focused on the reliability of cryocoolers [47], and some on conceptual designs of new cryogenic systems for space applications [48]. It is worth noting that the presented results focus almost exclusively on space Stirling refrigerators and do not discuss on-Earth applications. Univeristy of Tsukuba primarily studied the application of Stirling coolers for CO2 capturing [49]. A few studies focused on food processing [50] and cooling space instruments [51]. Sumimoto co-authored many papers with JAXA, but the company also studied the application of Stirling coolers to high temperature superconductors [52].

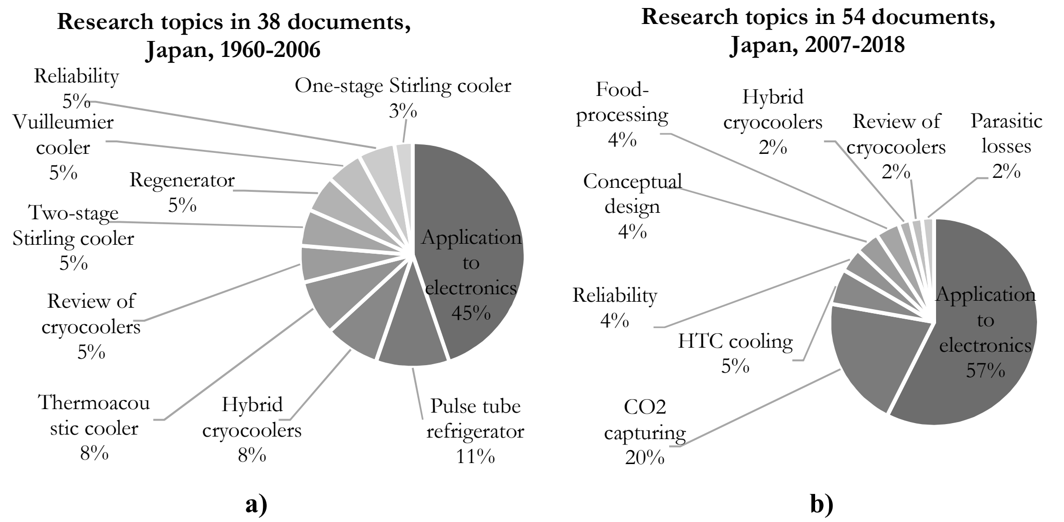


Fig. 11. Distribution of research topics in scientific documents by top publishing organizations in Japan: a) 1966-2006, 38 analyzed documents from 70 total b) 2007-2018, 54 analyzed documents from 90 total.

Figure 12 analyzes the ownership structure of active granted patent families in 2018 and indicates organizations that defined the growth in granted patent families over the last 18 years.

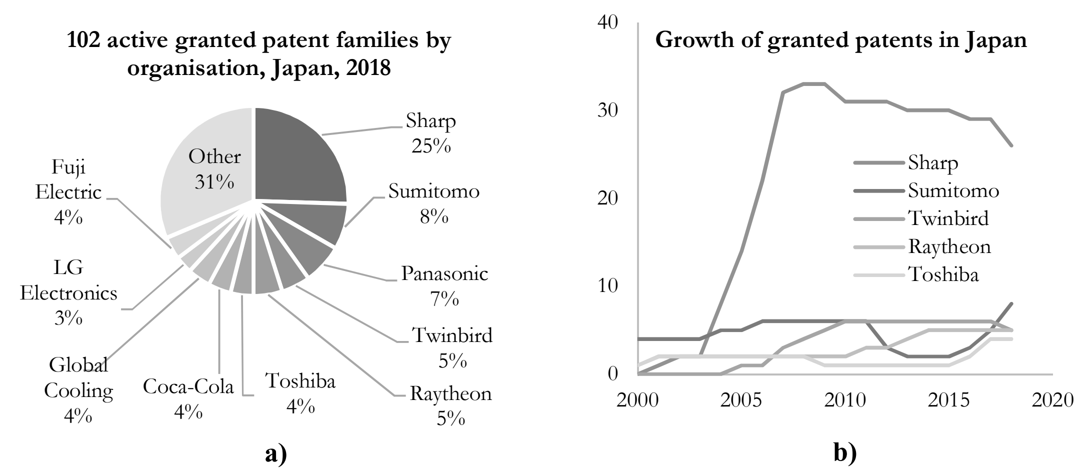


Fig. 12. Granted patent families in Japan: a) distribution of 102 active granted patent families by organization, active in 2018 b) Organizations that defined growth in granted patent families in Japan over the last decade.

Sharp has patented a cabinet refrigerator [53], original free-piston Stirling engine [54] and its control methods with a group of patents. Apart from patent families granted in United State, Sumitomo patented a split-system Stirling cryocooler in Japan [55]. Toshiba protected several configuration of its proprietary free-piston Stirling refrigerator [56]. Twinbird has the same patent portfolio in Japan as in the US. Raytheon has protected different inventions related to the split-Stirling system for aerospace electronics, such as the system with moving piston and cylinder [57], the combination of the Stirling refrigerator and the pulse tube cooler [58], and other improvements related to the motor, suspension system, and thermal interface.

II.3 CHINA

Between 1960 and 2006 in China were published 53 scientific documents with identifiable country of origin about Stirling refrigerators, which is 4.1 times less than United States and 1.3 times – than Japan. In the period between 2007 and 2018 the same number was 209 documents, or 1.3 times more than in US and 2.3 times – than in Japan, indicating a noticeable growth in scientific activities about Stirling refrigerators in China over the last 12 years. Figure 13 depicts the publishing trends of scientific and patent documents over time in China.

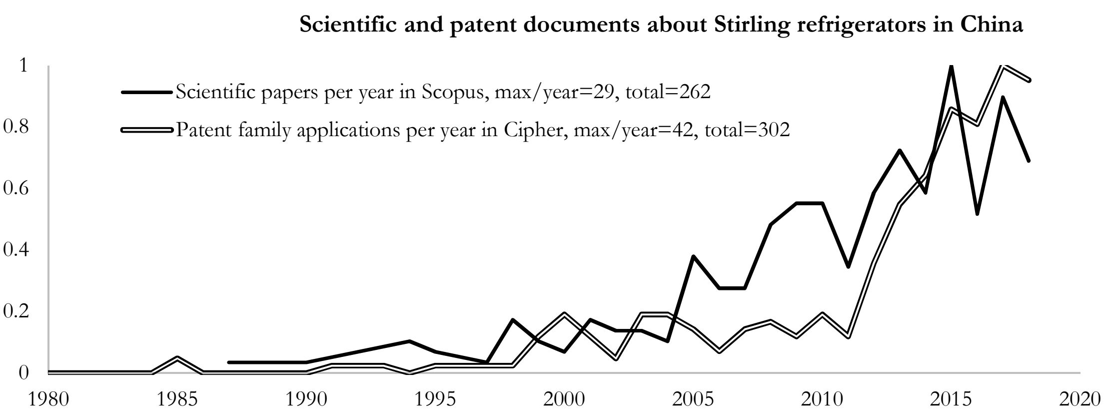


Fig. 13. Publishing activity for scientific papers and patent documents for the search term "Stirling refrigerator [etc.]" based on data from Scopus and Cipher for a period between 1980 and 2018 in China. Absolute numbers are normalized by the maximum number of documents per year.

Figure 14 shows the document affiliation for the two analyzed time periods. Before 2007, Chinese Academy of Science studied several topics, including the application of cryocoolers to high-temperature superconductor devices [59], thermoacoustic coolers [60] and pulse tube refrigerators [61]. Technical Institute of Physics and Chemistry and Shanghai Institute of

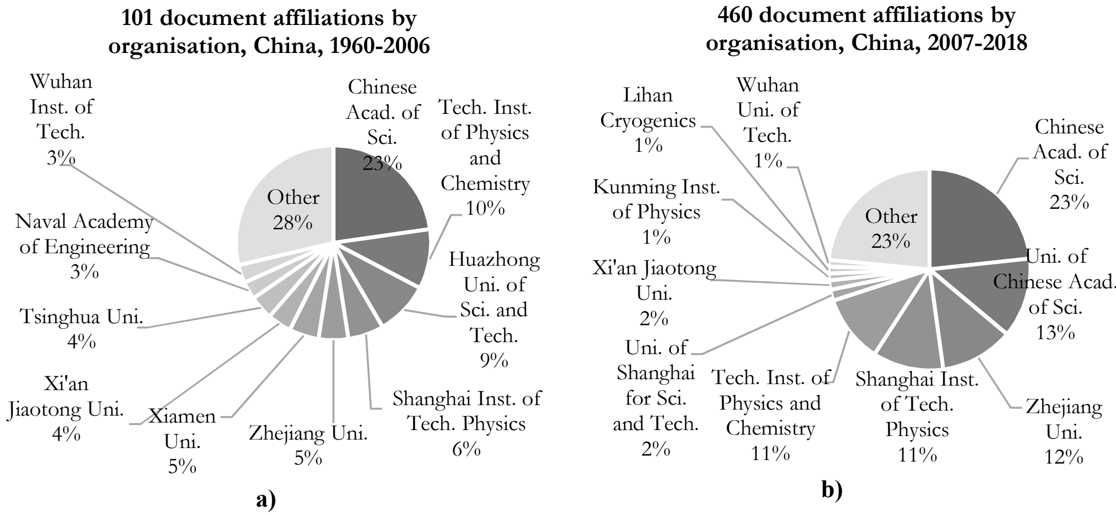
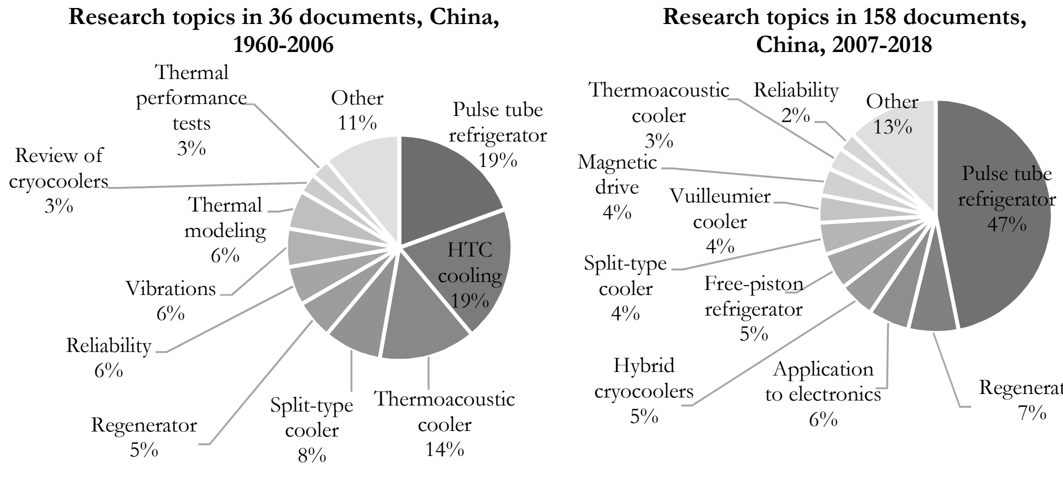


Fig. 14. Document affiliations in China by organization in Scopus: a) 1960-2006, total 101 document affiliations; b) 2007-2018, total 460 document affiliations.

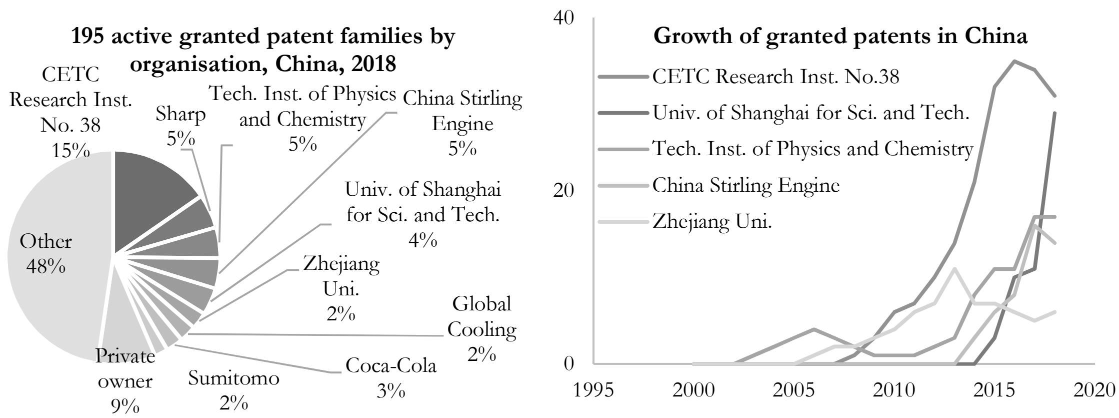
Technical Physics co-authored all referenced works with the Chinese Academy of Science. Huazhong University of Science and Technology studied different topics, including the regenerator [62] and thermal optimization of Stirling coolers [63]. Zhejiang University focused on the analysis of pulse tube refrigerators [64].

After 2006, Chinese Academy of Science reported many studies focused mainly on the investigation of pulse tube coolers [65]. To a lesser degree, the focus was on the application studies for space electronics [66], free-piston refrigerators [67], split-types coolers [68], Vuilleumier coolers [69], and hybrid coolers [70]. Occasionally the research activities reported on the studies of regenerators [71], magnetic drives [72] and reliability questions [73]. The University of Chinese Academy of Science co-authored affiliated works, focusing primarily on pulse tube coolers. Zhejiang University worked independently form the Chinese Academy of Science, but also focused primarily on the study of pulse tube refrigerators [74], with some focus on the regenerator properties at 4 K temperature levels [75] and with a few studies related to high-cooling capacity crank-rod Stirling refrigerators [76]. Shanghai Institute of Technical Physics and Technical Institute of Physics and Chemistry coauthored most of referenced studies with the Chinese Academy of Science.



a) **b)**

Fig. 15. Distribution of research topics in scientific documents by top publishing organizations in China: a) 1966-2006, 36 analyzed documents from 53 total b) 2007-2018, 158 analyzed documents from 209 total.



a) **b)**

Fig. 16. Granted patent families in Japan: a) distribution of 195 active granted patent families by organization, active in 2018 b) Organizations that defined growth in granted patent families in Japan over the last decade.

CETC Research Institute No. 38 protected different configurations of Stirling coolers (Fig.16b). The patents protect Stirling split-system [77], rotary alpha arrangement [78], rotary beta arrangement integrated with motor [79], and a hybrid cryocooler with a free-piston and pulse tube sub-coolers [80]. And other patents related to heat exchangers, moving magnets and control systems. The protected configurations indicate that the prime application area is the aerospace electronics. University of Shanghai for Science and Technology protected a pulse tube refrigerator [81] and several inventions for different application areas, such as natural gas liquefaction [82], storage cabinets [83], and water generation unit [84]. Technical Institute of

Physics and Chemistry protected a variety of duplex Stirling systems (coupled engine and refrigerator) [85], some hybrid cyocoolers [86] and rotary alpha arrangement coolers [87]. China Stirling Engine Mfg. Ltd has patented the cold container with Stirling free-piston cooler [88], the configuration for cooling truck containers [89], and for the moving belt food freezing [90]. Zhejiang University focused on protecting a proprietary pulse tube refrigerator [91] with related components and the hybrid cryocooling system [92].

III DISCUSSION

III.1 MATURITY OF THE DOMAIN

The results show that the domain of Stirling refrigerators is not well studied and have a limited number of design alternatives. This is supported by the fact that the total number of scientific documents published to date is 1391 (Fig. 1), which is 2.4 times less than for the domain of Stirling engines. The total number of patent applications to date is 1031 (Fig. 1), 3.1 times less than for Stirling engines. In addition, among ten top publishing sources 40% of scientific research over the last twelve years was still published in conference proceedings (Fig. 3b), a startling difference comparing to 5% for the Stirling engines. Another evidence of a young status of Stirling refrigerators comes from statistics about active patent families in 2018 and total patent expenditure. Figure 4 shows that currently there are 399 granted patent families or 2.5 less than for Stirling engines, and since there has been $13 million spent on patent protection since 2000 or 2.6 times less than for Stirling engines. In addition, Figure 1 indicates that although scientific publishing and patenting increased over time, the trends do not follow similar patterns and do not clearly lead one another. This behavior also shows that the knowledge base for Stirling refrigerators is developing non-systematically. Another evidence comes from the analysis of the ownership structure of patent family applications by organization over time. Figure 17 shows that there were periods when more than 50% of patent applications were defined by a small group of organizations (Fig. 17a and 17c) and periods where the patenting activities were performed by many different organizations and private individuals (Fig. 17b and 17d). It is interesting to note that those periods were consecutively following each other and that each lasted approximately ten years. Last two years in Fig. 17d even suggest that a new cycle of patenting activities is about to begin where more than 50% of patenting activities would be defined by a small number of top-publishing Chinese organizations. This cycle behavior in patenting, unlike for Stirling engines, shows that Stirling refrigeration technology evolves and have more participation of private individuals and organizations over time, yet this participation is not enough for the field to start systematic development of the technology distributed geographically and among different organizations and private individuals.

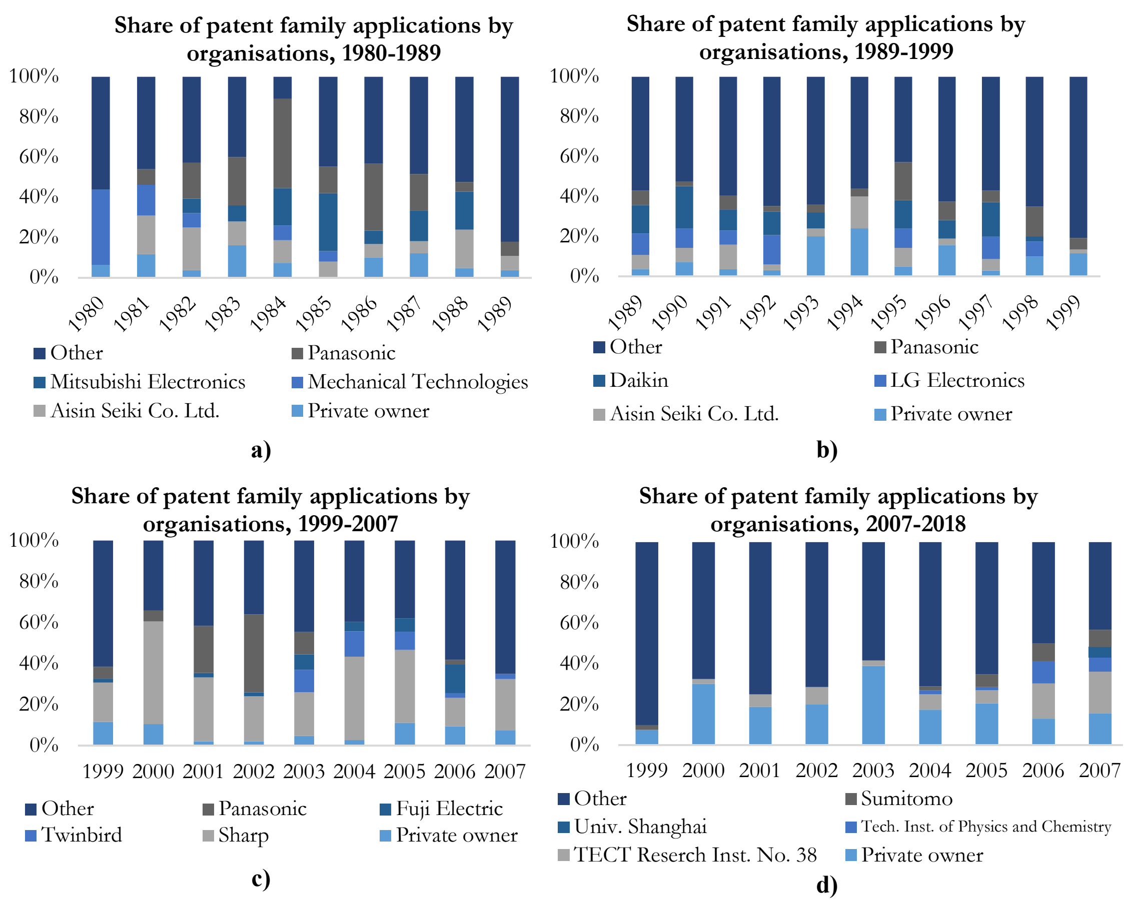


Fig. 17. Share of patent family applications by organisations for four time periods based on data from Cipher: a) 1980-1989, b)1989-1999, c) 1999-2007, d) 2007-2018.

Another interesting observation can be noted from the distribution of research topics in countries that published more than 50% of scientific research (Fig. 2) – US, Japan and China. US and Japan almost exclusively published results of perfecting aerospace electronics cryocoolers. China mainly focused on perfecting pulse tube coolers with its application to high-temperature superconductors and space. It is clear that this sub-fields of the Stirling refrigerators domain are strong and well-researched. However, it is important to note where these research activities did not focus on, and this is on-Earth high cooling capacity applications. This gap was partially covered by patent literature with patents protecting Stirling refrigerators for vehicle air-conditioning and cooling of transported goods, natural gas liquefaction and cabinet refrigerators. However, these examples are spares, and research and patenting activities are still inadequate. This is one more evidence that the domain of Stirling refrigerators is relatively young and still needs much research and inventing for the civil sector and commercial applications.

How does this result affect the selection of design alternatives? It shows that the alternatives for commercial applications are not well researched and patented. On the one hand, this limits the design arsenal of the technology developer, but on the other hand it makes room for novel design configurations. The conclusion that the high cooling capacity systems (50 W to 5000 W) are sparsely studied in scientific and patent literature was translated into the requirement for the testing and refinement stage to maximize cooling power of the experimental Stirling refrigerator system. The area of high capacity Stirling refrigerators is commercially promising with applications in gas liquefaction, biomedical storage and high-performance electronics cooling.

III.2 Differences between scientific literature and patent literature

Among research institutions that conducted noticeable scientific research and patenting at the same time are Technical Institute of Physics and Chemistry and Zhejiang University (Fig. 5 and 6). Both organizations are from China. The former protected a variety of duplex Stirling systems, and the latter patented a proprietary pulse tube refrigerator and hybrid cryocooling systems. These organizations may have a strategy to license these general inventions. Besides these two organizations, University of Shanghai for Science and Technology was very active in patenting its inventions (Fig. 6), but has had a limited scientific publishing noticeable only at the country level (Fig. 14). The scope of patents is diverse – a pulse tube refrigerator and several inventions for different application areas – which may indicate an intent to license these inventions. Apart from these examples, other research institutions were not noticeable in both scientific and patenting activities. Similarly to the domain of Stirling engines, this result shows that patenting and science for Stirling refrigerators are two independent domains with original research and engineering knowledge. Both domains may contain valuable alternatives for design decisions.

III.3 Applicability of scientific research and inventions

Scientific activities in US and Japan were conducted and financed by the governmental agencies for aerospace applications (Fig. 8 and 11) with cryocoolers having from several Watts to several dozens of Watts for cooling temperatures from 4 K to 80 K. Since China started active research only in 2005 (Fig. 13), it took time before the key research organizations in the country could reach a critical level of technololgy readiness. Most research organizations in China focused on the pulse tube cryocooler. In the upcoming years China is also expected to focus on the application studies of this technology. In the scientific domain most likely it will be aerospace instruments because the emergence of this topic in the research activities is already evident (Fig. 15b). The analysis of patent literature shows that the increase of patent applications in China, which started about ten years after the increase in scientific activities (Fig 13), could be describe by gradual implementation of pulse tube refrigerators in inventions. These is evident from patents of CETC Research Institute No. 38, University of Shanghai for Science and Technology and Zhejiang University that include inventions related to pulse tube cryocoolers.

Chinese research was motivated by the knowledge created in US and China. Therefore, even the selection of the pulse tube cryocoolers as a core technology is the result of this influence. This type of cryocoolers demonstrates low vibration characteristics and high reliability due to absent moving parts – requirements that are critical for space equipment. However, the integration of this technology into on-Earth commercial applications in some instances [52] proved to be less advantageous due to lower efficiency and scalability – critical requirements for civil applications. Figures 8, 11, and 15 conclusively prove that there was inadequate focus on high cooling capacity Stirling refrigerators for on-Earth commercial applications. There are example of these studies and patenting [76], [93]. Yet they are still inadequate to make an informed decisions about alternative design configurations for commercial Stirling refrigerators.

III.4 FINAL REMARKS ON SELECTING DESIGN ALTERNATIVES

The main finding related to the state of art in the domain of Stirling refrigerators is that it is much younger than the domain of Stirling engines. This was conclusively shown based on the amount of scientific and patent activity, patent expenditure and the evolution of these two subfields of knowledge over time. Some topics in Stirling refrigerators are very mature and include cryocoolers for aerospace instruments and pulse tube coolers. Nevertheless, this knowledge is very specific and is inadequate for commercial development of Stirling refrigerators for the civil sector. Therefore, one should be aware of the limited condition of this domain when evaluating existing design configurations of Stirling refrigerators. Similar to Stirling engines, both scientific literature and patent documents for Stirling refrigerators may contain insightful and original information about design alternatives. However, it is important to remember that the existing research was focused on space cryocoolers and influenced design decisions in scientific and patent literature accordingly. The selection of design alternative needs to be done with caution and understanding of inherent requirements embedded in the original design configurations of Stirling refrigerators.

III.5 Design of energy conversion technologies

The results reported in this chapter demonstrate the possibility of using the big data bibliometric method for the analysis of other energy conversion technologies. Even though the focus of the design was changed to another application area, the technique helped to quickly analyze existing scientific and patent knowledge, providing informative results and focus on relevant systematic studies reported in the literature. The development process of other ECT is not protected from a similar change of application area due to complex interplay of engineering and commercial process components. As mentioned in the first paragraph of Chapter I, the technological pathway focuses on the "technology push" developments; as such, it is typical for this type of projects to explore different applications. Therefore, the proposed in this chapter methodology can serve a quick and informative tool for literature analysis at the conceptual stage of other ECT.

III.6 Novelty and Contribution

- Identified a historically accurate account of scientific and patenting activities for Stirling refrigerators over the last 58 years;

- Identified statistically significant distribution of research topics about Stirling refrigerators;
- Developed a theoretical framework that defines factors that need to be taken into account when making design choices at the concept stage for commercial development of Stirling refrigerators and other energy conversion technologies.

IV CONCLUSIONS

- Stirling refrigerators is a relatively young domain of knowledge and contains approximately 2.5 less scientific and patenting documents than the domain of Stirling engines. It is also much less protected in terms of granted patent families and patent expenditure;
- Most research activities focus on the application of Stirling refrigerators for cooling aerospace electronics. There is inadequate coverage of high cooling capacity Stirling refrigerators and qualification studies for its civil applications;
- US, Japan and China were very active in studying rotary, free-piston and pulse tube cryocoolers for aerospace applications and in much fewer instances for high-temperature superconductor devices. The studies related to kinematic Stirling refrigerators are very limited;
- Patenting activities focused on designs of free-piston and pulse tube Stirling refrigerators and system designs of refrigeration cabinets. There are rare examples of patents protecting the operation of Stirling refrigerators as components of vehicle air-conditioning systems, truck container refrigeration, food-processing belts, and natural gas liquefaction systems;

- Based on the analysis of literature sources, the focus of testing and refinement stage was put on developing a high cooling capacity Stirling refrigerator because several on-Earth commercial applications were identified that required high cooling capacity, such as natural gas liquefaction, biomedical storage, and cooling high-performance computers.

REFERENCES

[1] Ross, R.G., Johnson, D.L. and Kotsubo, V., 1992. Vibration characterization and control of miniature Stirling-cycle cryocoolers for space application. In *Advances in cryogenic engineering*, pp. 1019-1027. Springer, Boston, MA.

[2] Coulter, D.R., Ross, R.G., Boyle, R.F. and Key, R.W., 2003. NASA advanced cryocooler technology development program. *In IR Space Telescopes and Instruments*, Vol. 4850, pp. 1020-1029. International Society for Optics and Photonics.

[3] Dick, G.J., Wang, R.T., Tjoelker, R.L. and Basu, R., 2005. Design and progress report for compact cryocooled sapphire oscillator" VCSO". *In Proceedings of the 2005 IEEE International Frequency Control Symposium and Exposition*, 2005, pp. 363-367. IEEE.

[4] Durso, S.S., May, R.D., Tuchscherer, M.A. and Webster, C.R., 1989. Frequency stability of a tunable diode laser mounted in a compact Stirling cycle cooler. *Applied optics*, 28(18), pp.3791-3792.

[5] Aumann, H.H. and Pagano, R.J., 1994. Atmospheric infrared sounder on the Earth Observing System. *Optical Engineering*, 33(3), pp.776-785.

[6] Patel, A.B. and Brisson, J.G., 2000. Experimental evaluation of a single stage superfluid Stirling refrigerator using a large plastic recuperator. *Journal of low temperature physics*, 118(3-4), pp.189-206.

[7] Hannon, C., Krass, B., Gerstmann, J., Chaudhry, G., Brisson, J., Smith, J. and Davis, T., 2005. Development of a 4K-10K Collins-type cryocooler for space. In *Cryogenic Optical Systems and Instruments* XI , Vol. 5904, p. 59040X. International Society for Optics and Photonics.

[8] Brisson, J.G., Kotsubo, V. and Swift, G.W., 1994. The superfluid Stirling refrigerator, a new method for cooling below 0.5 K. *Physica B: Condensed Matter*, 194, pp.45-46.

[9] Swift, G.W., Gardner, D.L. and Backhaus, S., 1999. Acoustic recovery of lost power in pulse tube refrigerators. *The Journal of the Acoustical Society of America*, 105(2), pp.711-724.

[10] Breon, S.R., Hopkins, R.A. and Nieczkoski, S.J., 1992. The X-ray spectrometer-A cryogenic instrument on the Advanced X-ray Astrophysics Facility. *In Advances in Cryogenic Engineering*, pp. 1193-1200. Springer, Boston, MA.

[11] Boyle, R., Banks, S., Cleveland, P. and Turin, P., 1999. Design and Performance of the HESSI Cryostat. *Cryogenics*, 39(12), pp.969-973.

[12] Shirey, K., Banks, S., Boyle, R. and Unger, R., 2006. Design and qualification of the AMS-02 flight cryocoolers. *Cryogenics*, 46(2-3), pp.143-148.

[13] Gully, W.J., Hopkins, R.A., Cork, C.P., Luke, P.N., Madden, N.W., Malone, D.F. and Pehl, R.H., 1996. Germanium gamma-ray detector operation with a flight-like mechanical cooler. *Cryogenics*, 36(10), pp.843-848.

[14] Glaister, D.S., Gully, W., Ross, R., Hendershott, P., Marquardt, E. and Kotsubo, V., 2006. Ball Aerospace 4–6 K Space Cryocooler. *In AIP Conference Proceedings*, Vol. 823, No. 1, pp. 632-639. AIP.

[15] Kirkconnell, C.S., Zagarola, M.V. and Russo, J.T., 2006. Hybrid Stirling/Reverse Brayton and Multi-stage Brayton Cryocoolers for Space Applications. *In AIP Conference Proceedings*, Vol. 823, No. 1, pp. 1489-1496. AIP.

[16] Kirkconnell, C.S., Harvey, J.P. and Desai, P.V., 2004. Comparison of Entropy Generation Rates in Various Multi-Stage Stirling-Class Cryocooler Configurations. *In AIP Conference Proceedings*, Vol. 710, No. 1, pp. 1519-1526. AIP.

[17] Wu, Y.W.A. and Kieffer, M., 1995. Active temperature control of Stirling-cycle cryocoolers. *In Proceedings of International Conference on Control Applications*, pp. 1059-1064. IEEE.

[18] Schaefer, B.R., Bellis, L., Ellis, M.J. and Conrad, T.J., 2014. Advanced regenerator testing in the Raytheon dual-use cryocoolerr. *In AIP Conference Proceedings*, Vol. 1573, No. 1, pp. 570-575. AIP.

[19] Schaefer, B.R., Bellis, L., Ellis, M.J. and Conrad, T.J., 2014. Raytheon low temperature RSP2 cryocooler airborne testing. *In AIP Conference Proceedings*, Vol. 1573, No. 1, pp. 1318-1324. AIP.

[20] Mulcahey, T.I., Conrad, T.J., Ghiaasiaan, S.M. and Pathak, M.G., 2014. Investigation of gravitational effects in pulse tube cryocoolers using 3-D CFD. *In AIP Conference Proceedings*, Vol. 1573, No. 1, pp. 1002-1009. AIP.

[21] Gan, Z.H., Liu, G.J., Wu, Y.Z., Cao, Q., Qiu, L.M., Chen, G.B. and Pfotenhauer, J.M., 2008. Study on a 5.0 W/80 K single stage Stirling type pulse tube cryocooler. *Journal of Zhejiang University-SCIENCE A*, 9(9), pp.1277-1282.

[22] Sneiderman, G.A., Shirron, P.J., Fujimoto, R., Bialas, T.G., Boyce, K.R., Chiao, M.P., DiPirro, M.J., Eckart, M.E., Hartz, L., Ishisaki, Y. and Kelley, R.L., 2016. Cryogen-free operation of the soft x-ray spectrometer instrument. *In Space Telescopes and Instrumentation 2016: Ultraviolet to Gamma Ray*, Vol. 9905, p. 99053N. International Society for Optics and Photonics.

[23] Fujimoto, R., Mitsuda, K., Yamasaki, N., Takei, Y., Tsujimoto, M., Sugita, H., Sato, Y., Shinozaki, K., Okamoto, A., Ohashi, T. and Ishisaki, Y., 2010. Cooling system for the soft x-ray spectrometer (SXS) onboard ASTRO-H. In *Space Telescopes and Instrumentation 2010: Ultraviolet to Gamma Ray*, Vol. 7732, p. 77323H. International Society for Optics and Photonics.

[24] Boyle, R. and Marquardt, E., 2014. TIRS cryocooler: spacecraft integration and test and early flight data. *In AIP Conference Proceedings*, Vol. 1573, No. 1, pp. 352-356. AIP.

[25] Sun, X., Abshire, J., Krainak, M., Lu, W., Beck, J., Sullivan, W., Mitra, P., Rawlings, D., Fields, R., Hinkley, D. and Hirasuna, B., 2018. Single photon HgCdTe avalanche photodiode and integrated detector cooler assemblies for space lidar applications. *In Advanced Photon Counting Techniques XII*, Vol. 10659, p. 106590C. International Society for Optics and Photonics.

[26] Pathak, M.G., Patel, V.C., Ghiaasiaan, S.M., Mulcahey, T.I., Helvensteijn, B.P., Kashani, A. and Feller, J.R., 2013. Hydrodynamic parameters for ErPr cryocooler regenerator fillers under steady and periodic flow conditions. *Cryogenics*, 58, pp.68-77.

[27] Kirkconnell, C.S., Hon, R.C., Perella, M.D., Crittenden, T.M. and Ghiaasiaan, S.M., 2017. Development of a miniature Stirling cryocooler for LWIR small satellite applications. In *Tri-Technology Device Refrigeration* (TTDR) II, Vol. 10180, p. 1018002. International Society for Optics and Photonics.

[28] Fang, T., Spoor, P.S. and Ghiaasiaan, S.M., 2017. A computational approach for coupled 1D and 2D/3D CFD modelling of pulse Tube cryocoolers. *In IOP Conference Series: Materials Science and Engineering*, Vol. 278, No. 1, p. 012172. IOP Publishing.

[29] O'baid, A.H. and Kunimoto, W.Y., Superconductor Technologies Inc, 2005. *Stirling cycle cryocooler with improved magnet ring assembly and gas bearings*. U.S. Patent 6,880,335.

[30] Yoshida, K., Twinbird Corp, 2005. *Portable container*. U.S. Patent 6,964,179.

[31] Berchowitz, D.M. and Saito, M., Twinbird Corp and Global Cooling BV, 2010. *Stirling cycle engine*. U.S. Patent 7,692,339.

[32] Rudick, A.G. and Berchowitz, D.M., Global Cooling BV and Coca-Cola Co, 2003. *Stirling refrigeration system with a thermosiphon heat exchanger*. U.S. Patent 6,550,255.

[33] Berchowitz, D.M., Global Cooling Inc, 2014. *Free-piston Stirling machine in an opposed piston gamma configuration having improved stability, efficiency and control*. U.S. Patent 8,752,375.

[34] Nakano, K. and Hiratsuka, Y., Sumitomo Heavy Industries Ltd, 2014. *Stirling-type pulse tube refrigerator and flow smoother thereof*. U.S. Patent Application 14/281,367.

[35] Nakano, K., Hiratsuka, Y. and Yumoto, K., Sumitomo Heavy Industries Ltd, 2015. *Stirling refrigerator*. U.S. Patent Application 14/666,459.

[36] Inatani, J., Narasaki, K., Tsunematsu, S., Kyoya, M., Manabe, T., Seta, M., Iida, T. and Satoh, R., 2001. Mechanical cooler and cryostat for submillimeter SIS mixer receiver in space. In *Sensors, Systems, and Next-Generation Satellites V*, Vol. 4540, pp. 197-209. International Society for Optics and Photonics.

[37] Narasaki, K., Tsunematsu, S., Ootsuka, K., Watanabe, N., Matsumoto, T., Murakami, H., Nakagawa, T., Sugita, H., Murakami, M. and Awazu, H., 2004. Development of 1 K-class mechanical cooler for SPICA. *Cryogenics*, 44(6-8), pp.375-381.

[38] Kyoya, M., Narasaki, K., Ito, K., Nomi, K., Murakami, M., Okuda, H., Murakami, H., Matsumoto, T. and Matsubara, Y., 1994. Development of two-stage small Stirling cycle cooler for temperatures below 20 K. *Cryogenics*, 34(5), pp.431-434.

[39] Narasaki, K., Tsunematsu, S., Kanao, K., Otsuka, K., Hoshika, S., Fujioka, K., Tsurumi, K. and Hirabayashi, M., 2006. Development of single stage Stirling cooler for space use. In AIP Conference Proceedings, Vol. 823, No. 1, pp. 1505-1512. AIP.

[40] Ueda, Y., Biwa, T., Yazaki, T. and Mizutani, U., 2003. Construction of a thermoacoustic Stirling cooler. *Physica B: Condensed Matter*, 329, pp.1600-1601.

[41] Yagi, W., Iwata, H. and Mizutani, U., 1997. Studies of Low Temperature Specific Heats and Thermal Conductivities of the CsCl-type (Pr1-xNdx) Ag (0≤ x≤ 1) Intermetallic Compounds: Application to Regenerator Materials. *Japanese journal of applied physics*, 36(9R), p.5638.

[42] Kawada, M., Fujisada, H., Akao, H., Kobayashi, M., Kawajiri, K., Nishimoto, A. and Nishihara, O., 1997. Performance of long-life Stirling cycle cryocoolers for the ASTER instrument. *In Sensors, Systems, and Next-Generation Satellites*, Vol. 3221, pp. 220-230. International Society for Optics and Photonics.

[43] Kawajiri, K., Honda, T. and Sugimoto, T., 1997. Study of Free Piston Vuilleumier Heat Pump: Basic Performance Analysis. *JSME International Journal Series B Fluids and Thermal Engineering*, 40(4), pp.617-625.

[44] Iwata, Y., Otsuka, M. and Akao, H., 1997. Preflight test results of the ASTER SWIR flight model. *In Sensors, Systems, and Next-Generation Satellites*, Vol. 3221, pp. 202-210. International Society for Optics and Photonics.

[45] Sato, Y., Sawada, K., Shinozaki, K., Sugita, H., Nishibori, T., Sato, R., Mitsuda, K., Yamasaki, N.Y., Takei, Y., Goto, K. and Nakagawa, T., 2014. Development status of the mechanical cryocoolers for the soft x-ray spectrometer on board Astro-H. *Cryogenics*, 64, pp.182-188.

[46] Satoh, T., Sato, T.M., Nakamura, M., Kasaba, Y., Ueno, M., Suzuki, M., Hashimoto, G.L., Horinouchi, T., Imamura, T., Yamazaki, A. and Enomoto, T., 2017. Performance of Akatsuki/IR2 in Venus orbit: the first year. *Earth, Planets and Space*, 69(1), p.154.

[47] Narasaki, K., Tsunematsu, S., Ootsuka, K., Kanao, K., Okabayashi, A., Mitsuda, K., Murakami, H., Nakagawa, T., Kikuchi, K., Sato, R. and Sugita, H., 2012. Lifetime test and heritage on orbit of coolers for space use. *Cryogenics*, 52(4-6), pp.188-195.

[48] Sato, Y., Sugita, H., Shinozaki, K., Okamoto, A., Yamawaki, T., Komatsu, K., Nakagawa, T., Murakami, H., Matsuhara, H., Murakami, M. and Takada, M., 2010, August. Conceptual design of a cryogenic system for the next-generation infrared space telescope SPICA. *In Space Telescopes and Instrumentation 2010: Optical, Infrared, and Millimeter Wave,* Vol. 7731, p. 77314G. International Society for Optics and Photonics.

[49] Song, C., Kitamura, Y., Li, S. and Lu, J., 2013. Deposition CO2 capture process using a free piston stirling cooler. *Industrial & Engineering Chemistry Research*, 52(42), pp.14936-14943.

[50] Sun, J.F., Kitamura, Y. and Satake, T., 2010. Improvement of the cooling performance of a food processing Stirling cooler. *Journal of food process engineering*, 33(6), pp.1052-1064.

[51] Nakagawa, T., Enya, K., Hirabayashi, M., Kaneda, H., Kii, T., Kimura, Y., Matsumoto, T., Murakami, H., Murakami, M., Narasaki, K. and Narita, M., 2007. Flight performance of the AKARI cryogenic system. *Publications of the Astronomical Society of Japan*, 59(sp2), pp.S377-S387.

[52] Nakano, K., Yumoto, K. and Hiratsuka, Y., 2015. Development of high-efficiency Stirling cryocoolers for high temperature superconducting motors. *In IOP Conference Series: Materials Science and Engineering*, Vol. 101, No. 1, p. 012003. IOP Publishing.

[53] Zhang, H. and Masuda, M., Sharp Corp, 2008. *Refrigerator.* U.S. Patent 7,386,984.

[54] Ogura, Y., Sharp Corp, 2005. *Stirling refrigerating machine*. U.S. Patent 6,886,348.

[55] Yoshitaka Ikuta et al., Sumitomo Heavy Ind. Ltd., *Refrigerator*. Japanese patent JP2000314570A.

[56] Otani Yasumi et al. Toshiba Corp., *Stirling refrigerator*. Japanese patent JP6054079B2.

[57] Hon, R.C., Bellis, L.A., Yoneshige, C.H., Kirkconnell, C.S. and Barr, M.C., Raytheon Co, 2013. *Cryocooler with moving piston and moving cylinder*. U.S. Patent 8,490,414.

[58] Price, K.D. and Kirkconnell, C.S., Raytheon Co, 2001. *Apparatus and method for achieving temperature stability in a two-stage cryocooler*. U.S. Patent 6,330,800.

[59] Ju, Y.L., Yuan, K., Hou, Y.K., Jing, W., Liang, J.T. and Zhou, Y., 2003. System design of 60 K Stirling-type coaxial pulse tube coolers for HTS RF filters. *Physica C: Superconductivity*, 386, pp.540-543.

[60] Liu, Y., Wu, C., Wu, F., Guo, F. and Li, Q., 2006. Investigation on the characteristic frequency of the micro-miniature thermoacoustic-Stirling refrigerator. *International journal of ambient energy*, 27(3), pp.137-140.

[61] Dang, H.Z., Ju, Y.L., Liang, J.T., Cai, J.H., Zhao, M.G. and Zhou, Y., 2005. Performance of Stirling-type non-magnetic and non-metallic co-axial pulse tube cryocoolers for high-Tc SQUIDs operation. *Cryogenics*, 45(3), pp.213-223.

[62] Xiang, Y., Kuang, B. and Guo, F.Z., 1995. Parity simulation of thermoacoustic effect in regenerator of Stirling cryocooler. *Cryogenics*, 35(8), pp.489-494.

[63] Wu, F., Chen, L., Wu, C., Sun, F. and Li, Q., 1999. Optimum performance of an irreversible Stirling cryocooler. *International journal of ambient energy*, 20(1), pp.39-44.

[64] Qiu, L.M., Sun, D.M., Yan, W.L., Chen, P., Gan, Z.H., Zhang, X.J. and Chen, G.B., 2005. Investigation on a thermoacoustically driven pulse tube cooler working at 80 K. *Cryogenics*, 45(5), pp.380-385.

[65] Liubiao, C., Xianlin, W., Xuming, L., Changzhao, P., Yuan, Z. and Junjie, W., 2018. Numerical and experimental study on the characteristics of 4 K gas-coupled Stirling-type pulse tube cryocooler. *International Journal of Refrigeration*, 88, pp.204-210.

[66] Dang, H., Tan, J., Zha, R., Li, J., Zhang, L., Zhao, Y., Gao, Z., Bao, D., Li, N., Zhang, T. and Zhao, Y., 2017. Advances in single-and multi-stage Stirling-type pulse tube cryocoolers for space applications in NLIP/SITP/CAS. *In IOP Conference Series: Materials Science and Engineering*, Vol. 278, No. 1, p. 012008. IOP Publishing.

[67] Yu, G.Y., Li, K., Dai, W., Luo, E.C. and Wu, Z.H., 2014. Investigation on a free piston Stirling cryocooler with large cooling capacity. *In AIP Conference Proceedings*, Vol. 1573, No. 1, pp. 1626-1632. AIP.

[68] Wang, L.Y., Yuan, Y., Ge, Z.G., Song, Y.J., Gan, Z.H. and Wu, Y.N., 2014. Study on a pneumatically driven split Stirling cryocooler operating above 100 Hz. *In AIP Conference Proceedings*, Vol. 1573, No. 1, pp. 1633-1637. AIP.

[69] Wang, Y., Wang, X., Dai, W. and Luo, E., 2018. A cryogen-free Vuilleumier type pulse tube cryocooler operating below 10 K. *Cryogenics*, 90, pp.1-6.

[70] Pan, C., Zhang, T., Wang, J., Chen, L., Guo, J., Zhou, Y. and Wang, J., 2017. Experimental progress of a 4K VM/PT hybrid cryocooler for pre-cooling 1K sorption cooler. *In IOP Conference Series: Materials Science and Engineering*, Vol. 278, No. 1, p. 012044. IOP Publishing.

[71] Zhou, Q., Chen, L., Pan, C., Zhou, Y. and Wang, J., 2015. Experimental investigation on regenerator materials of Stirling-type pulse-tube refrigerator working at 20 K. *Physics Procedia*, 67, pp.530-535.

[72] Huo, Y., Wang, T., Yan, C., Dong, L. and Chen, H., 2013. Single phase, single slot tubular permanent magnet linear motor for Stirling cryocooler—design and operation. *In 2013 IEEE International Conference on Information and Automation (ICIA)*, pp. 747-750. IEEE.

[73] Wan, F., Tan, Y., Jiang, Z., Chen, X., Wu, Y. and Zhao, P., 2017. Lifetime prediction and reliability estimation methodology for Stirling-type pulse tube refrigerators by gaseous contamination accelerated degradation testing. *Cryogenics*, 88, pp.116-124.

[74] Sun, J.C., Dietrich, M., Qiu, L.M. and Thummes, G., 2015. Operating characteristics of a single-stage Stirling-type pulse tube cryocooler with high cooling power at liquid nitrogen temperatures. *Journal of Zhejiang University-SCIENCE A*, 16(7), pp.577-585.

[75] Li, Z., Jiang, Y., Gan, Z., Qiu, L. and Chen, J., 2015. Influence of regenerator void volume on performance of a precooled 4 K stirling type pulse tube cryocooler. *Cryogenics*, 70, pp.34-40.

[76] Xu, Y., Sun, D., Qiao, X., Yan, S.W., Zhang, N., Zhang, J. and Cai, Y., 2017. Operating characteristics of a single-stage Stirling cryocooler capable of providing 700 W cooling power at 77 K. *Cryogenics*, 83, pp.78-84.

[77] Zhouyi fan et al. *Measuring system of linear Stirling refrigerator dynamic displacement*. CETC Research Institute No. 38. Chinese patent CN202511758U, 2012.

[78] Wu Youmin et al. *Stirling refrigerator supported by oil-free lubrication liner bearing*. CETC Research Institute No. 38. Chinese patent CN201992902U, 2010.

[79] Chi Guochun et al. *Rotary integrated Stirling cryocooler*. CETC Research Institute No. 38. Chinese patent CN106679216A, 2016.

[80] Wang Bo. *Stirling cycle helium refrigerator up to two temperature zones*. CETC Research Institute No. 38. Chinese patent CN104819593B, 2015.

[81] Chen Wei, Wu Yuan. *Pulse tube type free piston Stirling refrigerator*. University of Shanghai for Science and Technology. Chinese patent CN108626903A, 2018.

[82] Chen Wei, Ling Fei. *Cascade regenerative natural gas liquefying system using pulse-tube type Stirling cryocoolers*. University of Shanghai for Science and Technology. Chinese patent CN108413705A, 2018.

[83] Chen Yu et al. *Novel Stirling air cooling frost-free refrigerator and temperature control method. University of Shanghai for Science and Technology*. Chinese patent CN109028702A, 2018.

[84] Chen Wei, Zhang Lijin. *Solar energy air water obtaining device*. University of Shanghai for Science and Technology. Chinese patent CN108797702A, 2018.

[85] Yu Guoyao et al. *A kind of combined cooling and power system*. Institute of Engineering Thermophysics , Chinese Academy of Science. Chinese patent CN106884735B, 2017.

[86] Wang Xiaotao et al. *Mixed cycle cryocooler*. Institute of Engineering Thermophysics , Chinese Academy of Science. Chinese patent CN107940790A, 2017.

[87] Li Qing et al. *Stirling refrigerator with linear slide rail bearing support*. Institute of Engineering Thermophysics , Chinese Academy of Science, Chinese Academy of Sciences. Chinese patent CN1683845A, 2004.

[88] Cheng Lu. *Stirling refrigerator*. China Stirling Engine Mfg. Ltd. Chinese patent CN204027143U, 2014.

[89] Cheng Lu. *Stirling motor commodity circulation refrigerator car*. China Stirling Engine Mfg. Ltd. Chinese patent CN206664413U, 2017.

[90] Cheng Lu. *Mesh belt refrigerator*. China Stirling Engine Mfg. Ltd. Chinese patent CN106839598A, 2017.

[91] Gan Zhihua et al.. *Pulse tube/Stirling gas coupling composite multi-stage refrigerator*. Zhejiang University. Chinese patent CN103062952B, 2013.

[92] Wang Bo et al. *Stirling/pulse tube composite type cryocooler precooled low-temperature J-T cryocooler.* Zhejiang University. Chinese patent CN108954890A, 2018.

[93] Kim, T. and Kwangweon, A.H.N., Hyundai Motor Co, 2014. *Stirling refrigerator for vehicle*. U.S. Patent Application 13/710,262.

CHAPTER V
SEALING IN STIRLING REFRIGERATORS

The existing scientific and engineering literature, which I analyzed in Chapter IV, define the state of art for technical characteristics of Stirling refrigerators. This knowledge helped to set the strategy for desired operation of the developed Stirling machine, which is the maximization of the cooling capacity. The piston-cylinder seal for the Stirling refrigerator is one of the most essential components to improve performance due to two reasons: it provides the required compression in the cycle and sustains high level of cooling capacity, and it defines reliability of the commercial system. This element operates under challenging conditions. The piston-cylinder seal experiences pressure gradients over the gap and has high surface speeds. These parameters change more than nine times per second due to typical shaft frequencies from 9 Hz to 25 Hz. The analytical modelling of this component is difficult because the operation of the seal or other critical system components is a complex combination of factors from mechanics, fluid dynamics and tribology. An alternative approach to analyze different design alternatives for the seal is to perform experimental optimization. This design problem at the stage of testing and refinement can be generalized for other energy conversion technologies (ECT). Different ECT have design components critical for their performance. The components that operate under most stressful and difficult to model conditions require experimental optimization because the fidelity of analytical or numerical optimization is likely to be low. This chapter demonstrates how the experimental optimization of a critical component in an ECT can significantly improve its performance. More specifically, this chapter shows the application of this design approach using the example of the piston-cylinder sealing optimization in a real Stirling refrigerator.

I INTRODUCTION

Cascade vapor-compression refrigeration, cascade VCR or CVCR [1] has been widely used in chemical [2], biological [3], and pharmaceutical [4] industries to produce many value-added products. The cooling temperatures are found between −30ºC and −150ºC with cooling loads of several hundred watts. Empirical studies recently showed significant improvement of coefficient of performance or COP (Fig. 1), as well as cost and environmental benefits of Stirling refrigerators (SR) [3], [5]-[7].

Nomenclature

p_a	Pa	Absolute atmospheric pressure
R	$J \cdot kg^{-1} \cdot K^{-1}$	Specific gas constant
T_r	°C	Room temperature
p_c	Pa	Absolute air compressor temperature
q	$m^3 \cdot s^{-1}$	Seal volumetric leakage rate
u_p	$m \cdot s^{-1}$	Piston speed
V_{cv}	m^3	Leakage control volume
τ	s	Time to empty the leakage control volume
q_{cv}	$ml \cdot s^{-1}$	Control volume leakage rate
ρ_{cv}	$kg \cdot m^{-3}$	Air density in control volume
$\bar{\rho}$	$kg \cdot m^{-3}$	Average density in the seal gap
$\Delta p_{a,h}$	Pa	Seal gauge pressure for air/helium
δ	m	Radial seal gap
$\mu_{a,h}$	$Pa \cdot s$	Gas dynamic viscosity for air/helium
D	m	Piston diameter
L	m	Piston length
ε	m	Piston eccentricity
$u_{a,h}$	$m \cdot s^{-1}$	Average flow speed in the sealing gap for air/helium
d_e	m	Equivalent diameter for the sealing gap
$\mu_{s,k}$	$N \cdot m$	Mean value of the static/kinetic friction moment
σ	$N \cdot m$	Standard deviation of the static friction moment
$CV = \sigma / \mu_s$	-	Coefficient of variation for the static friction moment
l	m	Length of friction lever
$\bar{F}_{PCP}$	N	Average break-away force of the piston-cylinder pair (PCP)
F_{tot}	N	Total break-away force both for the mechanical drive and mounted PCP
F_m	N	Break-away force both for the mechanical drive with dismounted PCP
W_f	W	Friction losses in PCP during operation
T_c	°C	Stirling refrigerator (cooling) temperature
T_h	°C	Stirling refrigerator hot side (radiator) temperature
$T_{a,e,b}$	°C	Measured temperatures: ambient, electrical heater plate, shaft bearing

p	Pa	Charge pressure of the Stirling refrigerator
m_c	$1 \cdot \mathrm{min}^{-1}$	Radiator cooling water flow rate
Q_c	W	Stirling refrigerator cooling capacity
$W^{(\mathrm{ideal})}$	W	Ideal power input to the Stirling refrigerator
$p_{1,2}$	Pa	Refrigerant gas pressure at the beginning/end of compression
f	Hz	Shaft frequency
$\overline{m}_{3-4}$	kg	Average mass of refrigerant gas during the expansion
m_2'	kg	Gas mass after the compression with gas leakage
$m_{3,4}'$	kg	Gas mass before/after the expansion with gas leakage
$p_{3,4}'$	Pa	Gas pressure before/after the expansion with gas leakage
ρ_3'	$\mathrm{kg} \cdot \mathrm{m}^{-3}$	Gas density before the expansion with the leakage
γ	s	Duration of the compression and the expansion processes
$\Delta m_{\mathrm{c,e}}$	kg	Average mass of leaked gas in the compression/expansion
$q_{\mathrm{c,e}}$	$\mathrm{m}^3 \cdot \mathrm{s}^{-1}$	Average volumetric leakage rate in the compression/expansion
$\rho_{\mathrm{c,e}}^{\mathrm{g}}$	$\mathrm{kg} \cdot \mathrm{m}^{-3}$	Leakage density in the compression/expansion
$Q_{\mathrm{h}}^{(\mathrm{eq})}$	W	Heating load
U_{dc}	V	DC voltage of electrical heater
I_{dc}	A	DC current of electrical heater
t_{eq}	min	Thermal equilibrium time in thermal chamber
t	min	Duration of experiment with the Stirling refrigerator

SRs are 35% to 95% more energy efficient than VCR for cooling temperatures below 0 °C and with cooling loads from 20 W to 200 W. The energy efficiency increase for low temperature refrigerators is relevant, because their absolute energy consumption is about 17 times higher than for typical domestic refrigerators [5]. Furthermore, in her analysis, Legett [3] demonstrates how the 30% replacement of conventional low temperature refrigerators in U.S. with 25% more efficient systems could result in potential cost savings of $12.5 millions a year.

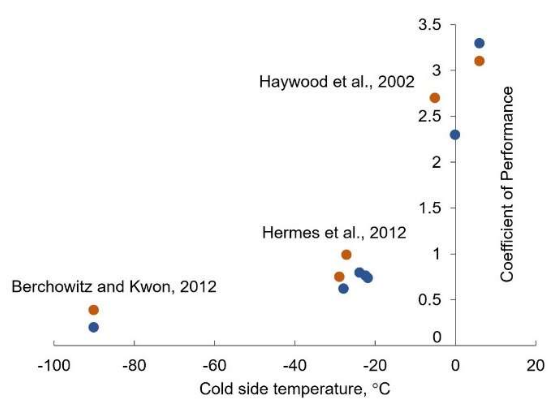


Fig. 1. Empirical comparison of COP values for vapor-compression refrigerators (blue) and Stirling refrigerators (orange).

Analytical and empirical studies [8]-[10] provide additional evidence that SR is more efficient than CVCR for temperatures between −30ºC and −150ºC at cooling loads of several hundred watts. This is possible due to the heat regeneration process which is absent in VCR. Simplicity, energy efficiency, and environmental friendliness make the experimental study of SRs timely from the scientific and technological viewpoints. Previous studies [11]-[16] have inadequately addressed the experimental analysis of separate SR elements and their influence on reliability, cooling capacity and COP values. We aim to partially bridge that gap in this chapter. We concentrate on the experimental evaluation of the sealing mechanism in piston-cylinder pair (PCP) of SRs, because this element carries complex functionality that affects reliability, efficiency, cooling capacity and cost of SR, thus influencing commercial dissemination of this emerging refrigeration technology.

The chapter is organized as follows. Sections I.1 and I.2 provide a detailed discussion of four different sealing designs in Stirling machines [11], [17]. In Section II, I present the methodology to study the PCP in a SR experimentally, addressing in particular the problems caused by leakage. We analyze and discuss our experimental data in Section III, and end the chapter with concluding remarks.

I.1 Sealing in Stirling machines

The sealing between piston and cylinder is a universal and crucial element in SMs. There are currently four technical solutions (see Fig. 2) implemented separately or in combination for the SM PCP: Lubricated piston rings (a) [18]-[20], dry-friction piston rings (b) [21]-[23], gas bearings (c) [24]-[26], and labyrinth seals (d) [27]-[29].

I.2 High-tolerance seal

There is much uncertainty about optimal design choice for PCP seals. Oil-based piston rings, labyrinth sealing and dry friction piston rings have been designed for seals of internal combustion engines and air compressors working in conditions of high-pressure gradients, up to 33 MPa. Common designs of SMs do not have large pressure gradients over the sealing because high pressures are created by the gas pressure charge within the SM housing. Therefore, the condition to have a developed ring or labyrinth sealing structure may not be relevant. The gas bearing seal has a complicated gas channel structure that makes it a relatively expensive bearing and sealing solution, while little is discussed in scientific literature about its sealing performance. Searching for an optimal seal design for SM, we evaluate an alternative approach to piston sealing, which we called “high-tolerance seal”, and includes high-pressure resistance gap with increased sealing length and small gap between mating surfaces. Fig. 2 depicts key elements of reviewed (a–d) and proposed for analysis (e) seals, and Table 1 summarizes the various types of piston sealing with the advantages and disadvantages that we identify from the literature.

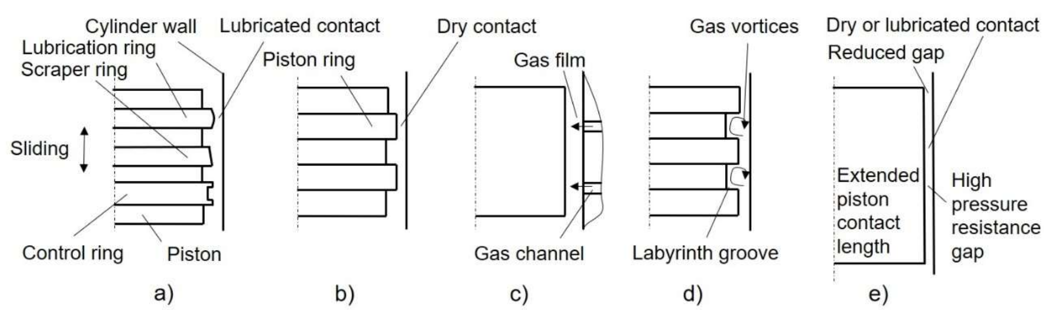


Fig. 2. Illustrative representation of piston-cylinder seal types in Stirling machines: a) Lubricated piston rings, b) dry-friction piston rings c) gas bearing, d) labyrinth seals, and e) analyzed in this work high-tolerance seal.

II METHODOLOGY

The object of this study, a high-tolerance PCP (Fig. 2 (e)), is analyzed with three measurement methods to characterize its performance. They include: variation of pressure difference over the seal to evaluate the change of the leakage rate, measurement of the static friction force that is required to overcome the cohesion in the pair, and actual operation of the pair within the SR [30] (Fig. 3) to measure the change of refrigeration temperature. This multifaceted approach helps to characterize PCP leakage sealing, contact efficiency, and sustaining low operational temperatures. Earlier findings [11]-[16] used alternative methods relevant for the performance evaluation of the entire SR. The proposed methodology permits the evaluation of one specific element of the refrigerator – the PCP.

Table 1 summarizes types of piston sealing with implied from the literature advantages and disadvantages.

Table 1. Types of piston-cylinder sealing solutions with advantages and disadvantages.

Seal type	Sealing effect	Advantages	Disadvantages
Oil-based piston rings (Fig. 2, a)	Tight contact between rings and cylinder with hydrodynamic lubrication	– Known and cheap	– High friction losses – Wear in 100 to 6000 hours
Dry friction piston rings (Fig. 2, b)	Tight self-lubricated contact between rings and cylinder	– Cheap – No oil lubricants – Operating life from 4000 to 24000 hours	– Friction losses – Intensified wear due to contacts with condensed water and oil films
Gas bearing (Fig. 2, c)	Pressurized gas create the gas film in gap that blocks the leakage	– No friction losses – No lubrication – Sealing by internal gas – No wear limit	– Expensive design – Precise centering – Pumping losses
Labyrinth (Fig. 2, d)	Generation of gas vortices in piston or cylinder grooves with increased pressure resistance	– No friction losses – No lubrication – Sealing by internal gas – No wear limit	– Precise centering – Internal leakages – Low volumetric efficiency
High-tolerance (Fig. 2, e)	Increased pressure resistance through reduced gap and extended piston contact length	– Simple design and manufacturing – Reduced friction losses – Little or no lubricants – No wear limit	– Larger dimensions – Precise centering and gap manufacturing

II.1 Piston-cylinder pair in Stirling refrigerators

A PCP is the combination of the piston (2) and the cylinder (11) that are separated by the sealing gap (10). PCPs for the compression volume (3) and for the expansion volume (7) are similar in design in this work; therefore, the term PCP is used for both elements. The piston in the PCP has diameter of 48 mm, length of 60 mm, and is manufactured of gray cast iron. The cylinder is manufactured of steel AISI 420S. The machining of PCP was conducted by an external manufacturer using conventional turning and grinding methods under human control with no computer numerical control. The average sealing gap between piston and cylinder claimed by the manufacturer is 10 μm. The PCP has reduced side forces because the pistons in the improved Ross-yoke mechanism (1) move in parallel to the cylinder walls [31]. The Ross-yoke mechanism type is a kinematic configuration of the refrigerator mechanical drive that transforms shaft rotation into the reciprocal movement of pistons without extorting significant force on cylinder walls. Such parallel piston movements allows generalizing results of this analysis both for kinematic and free-piston SMs with minimal side forces. The maximum misalignment between piston and cylinder of the designed Ross-yoke kinematic mechanism is 1 mm.

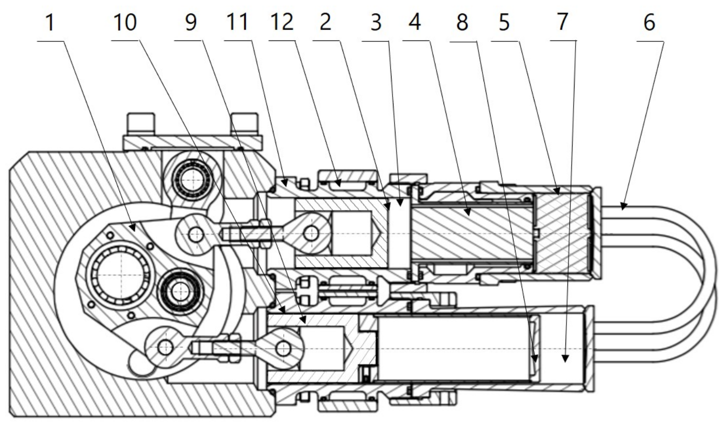


Fig. 3. Internal structure of the in-house Stirling refrigerator [29]: 1 kinematic drive of Ross-yoke type; 2 compression piston; 3 compression space; 4 water-cooled radiator; 5 regenerator; 6 heat exchanger; 7 expansion space; 8 displacer; 9 expansion piston; 10 piston-cylinder sealing gap; 11 compression cylinder; 12 cooling jacket.

II.2 Sealing types

To protect the piston surface from the direct contact with the cylinder surface and ensure gap sealing at a relatively low shaft frequency (9.1 Hz), we studied the PCP performance with the application of two different types of seals, namely the high-tolerance PCP with the lubricant NLGI 3 consistency classification number (hereafter Seal 1) and the high-tolerance PCP with the lubricant NLGI 2 (Seal 2). The application of lubricants in this work is justified by two factors: first, comparing to other NLGI numbers, lubricants with NLGI 2 and NLGI 3, after covering piston and cylinder surfaces, remained on the surfaces without dripping while, second, provided surface protection with relatively lower friction due to lower viscosity.

II.3 VARYING PRESSURE CONDITIONS

During the operation of the SR (Fig. 3), the PCP experiences varying gauge pressure over the sealing gap with estimated values from 0 to 1 MPa. The higher the gauge pressure, the higher gas leakage rate. To model this process, the PCP was dismounted from the SR (Fig. 3) and tested in a separate test bench (Fig. 4). No lubricant was applied to the PCP in this treatment to validate the 10 μm claimed by the manufacturer gap and to model a worst-case scenario for the leakage rate, characterized by a significant lubricant oil bleed [32] due to high operational dynamic and static loading or by the gradual loss of the lubricant due to dynamic forces and pressure gradients in the gap. In Fig. 4, the compressor (1) generated different pressures, controlled by the manometer (2). The cylinder with plugs (4) hermetically encompassed the piston (3). The pressure was applied from one side of the cylinder in the direction (7)-(4). The other side was connected to the control volume (5), which was filled with water under atmospheric pressure. The gas leakage pressed the water out of the controlled volume until it was emptied. We controlled the air pressure on one end with the manometer (2) and kept it constant in the control volume (5) on the other end at the atmospheric pressure.

II.3.1 GAS LEAKAGE MEASUREMENT

The time τ to empty the control volume (5) (Fig. 4) was measured for different compressor pressures p_c. The required pressure level was measured with a manometer (2) (accuracy class 1.5). After opening the valve (7), the time was measured with a digital chronometer to empty the control volume $V_{cv} = 5.2\times10^{-3}\,\mathrm{m}^3$. The measurements were conducted for compressor pressures of 0.19 MPa, 0.28 MPa, 0.38 MPa, 0.47 MPa, and 0.58 MPa. The air temperature in the gap was assumed to be equal to room temperature $T_r = 20°\mathrm{C}$

after thermal equilibrium between the test bench and environment. During measurements the pair was not in operation. In the long term, the wear of pair would lead to increased leakage loss [33].

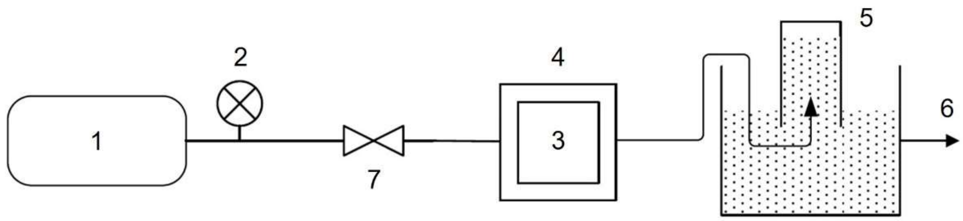


Fig. 4. Schematic diagram for measurements of gas leakage: 1 compressor receiver; 2 manometer; 3 piston; 4 cylinder with plugs; 5 control volume; 6 hydraulic locking; 7 valve.

II.3.2 SEALING GAP LEAKAGE CALCULATION

The volumetric leakage rate for the control volume $q_{\mathrm{cv}} = V_{\mathrm{cv}}/\tau$ was not equal to the seal volumetric leakage rate q due to difference in gas densities. Using the continuity equation, we find:

$$q = q_{\mathrm{cv}} \cdot \frac{\rho_{\mathrm{cv}}}{\overline{\rho}} \tag{1}$$

where the average air density in the sealing gap $\overline{\rho}$ was calculated using the ideal gas model $\overline{\rho} = \overline{p}/(RT_{\mathrm{r}})$ and the average seal pressure $\overline{p} = (p_{\mathrm{c}} + p_{\mathrm{a}})/2$. The compressor pressure p_{c} was acting on the seal from the front side, and the atmospheric pressure p_{a} – from the opposite side and was assumed to be 101.325 kPa. Similarly, we calculated the average air density in the control volume ρ_{cv} , using the ideal gas model and the absolute atmospheric pressure. In a general case, the seal volumetric leakage rate q depends on the piston eccentricity ε relative to the cylinder [34]. During the measurements of air leakage rate and the refrigerator operation, the piston inside the cylinder could be found between two limit positions, concentric and eccentric to the cylinder axis. Ensuring a uniform gap is a complex problem due to small imperfections of manufactured mating surfaces, the hydrodynamic forces created by a lubricant or a gas in the sealing gap, and the side forces generated by the mechanical drive. Therefore, we assumed that the gap during the operation could be at times either uniform (co-axial PCP) or non-uniform (eccentric PCP). For a laminar flow, the volumetric leakage rate q in a concentric cylindrical gap δ is related to the gap pressure gradient Δp for both limit positions [35], [36]:

$$q = \pi D\left(\frac{\Delta p\delta^3}{12\mu L} - \frac{u_p\delta}{2}\right)\cdot\left(1+\frac{3}{2}\frac{\varepsilon^2}{\delta^2}\right) \tag{2}$$

and we see that compared to the concentric state (eccentricity $\varepsilon = 0$), the volumetric leakage rate q with $\varepsilon = \delta$ may increase 2.5 times when the piston is placed in contact with the cylinder. Eq. (2) also indicates that the reduction of leakage mass flow rate when piston is moving is proportional $0.5\pi D u_{\mathrm{p}}\delta$. For each measured compressor pressure, the piston speed u_{p} was 0 (see Section II.3) and the gap calculated for two limit states. The average was used to validate the value of the radial sealing gap of 10 μm claimed by the manufacturer.

II.3.3 SIMILARITY CONDITION

The similarity condition for the air and helium flow in the gap was calculated because it was necessary to ensure that the experimental data obtained from an air-based pair (see Section II.3.1) could be used to characterize the operation of a helium-based pair. The thermodynamic and thermoelastic properties of air and helium under similar temperatures and pressures could be quite different. Furthermore, although the pressure gradient over the seal is similar for the air pair and the operational helium pair; the latter operates under the average ambient helium pressure 2.5 MPa, while the former was tested with ambient pressures several times lower. Different ambient pressures result in different gas densities in the gap. To be confident that the gas leakage measured with air pair would be similar to the helium leakage, we employed the similarity condition, which is represented by dimensionless groups of Reynolds and Euler numbers, Eqs. (3) and (4) respectively [37]:

$$\frac{\bar{\rho}_{\mathrm{a}} \cdot u_{\mathrm{a}} \cdot d_{\mathrm{e}}}{\mu_{\mathrm{a}}} = \frac{\bar{\rho}_{\mathrm{h}} \cdot u_{\mathrm{h}} \cdot d_{\mathrm{e}}}{\mu_{\mathrm{h}}} \tag{3}$$

$$\frac{\Delta p_{\mathrm{a}}}{\bar{\rho}_{\mathrm{a}} \cdot u_{\mathrm{a}}^2} = \frac{\Delta p_{\mathrm{h}}}{\bar{\rho}_{\mathrm{h}} \cdot u_{\mathrm{h}}^2} \tag{4}$$

where the dynamic viscosity values are those at room temperature: $\mu_{\mathrm{a}} = 1.82 \times 10^{-5}\ \mathrm{Pa \cdot s}$ for air, and $\mu_{\mathrm{h}} = 1.96 \times 10^{-5}\ \mathrm{Pa \cdot s}$ for helium. The operational helium PCP was cooled using tap water at temperature 17°C. Average densities were calculated for the atmospheric temperature after the gap for the air PCP and for 2.5 MPa after the gap for the helium PCP. The velocities u_{a} and u_{h} were calculated using Eq. (2) and the value of the gap cross-section area. The characteristic size d e was taken to be equal to two average seal gaps δ derived from Eq. (2). The dependence of Euler number and Reynolds number on gauge pressure is discussed below in the Results section. The range of gauge pressures was identified where the difference between Re and Eu numbers for air and helium was less than 25%. We assumed the similarity for this pressure range. The similarity gauge pressure range was used to approximate the dependence of the leakage rate from the gauge pressure in the entire range from 0 to 1 MPa. The maximum gauge pressure of 1 MPa was selected based on the ideal compression ratio of 1.4 for the SR with the charge pressure 2.5 MPa. Under the assumption that the pressure after the gap remains constant 2.5 MPa, the maximum gas pressure in the end of the compression cycle was 3.5 MPa.

II.4 Effects of gas leakage on ideal cooling capacity

The ideal cooling capacity $Q_{\mathrm{c}}^{(\mathrm{ideal})}$ for the SR (Fig. 3) was used as a reference value to evaluate the effect of measured leakages [38]:

$$Q_{\mathrm{c}}^{(\mathrm{ideal})} = mfRT_{\mathrm{c}} \ln\left(\frac{p_3}{p_4}\right) = mfRT_{\mathrm{c}} \ln\left(\frac{V_4}{V_3}\right) \tag{5}$$

The operational parameters of the SR are: $m = 0.571 \times 10^{-3}\ \mathrm{kg}$, $f = 9.1\ \mathrm{Hz}$, $R = 2077\ \mathrm{J \cdot kg^{-1} \cdot K^{-1}}$, $T_\mathrm{c} = -96°\mathrm{C}\left(177\ \mathrm{K}\right)$, and $V_4/V_3 = 1.4$, so we obtain $Q_\mathrm{c}^\mathrm{ideal} = 647\ \mathrm{W}$ for the ideal refrigerator cooling capacity. However, it should be reduced, because the refrigerant gas leaks from the compression space (Fig. 3, 3) to the kinematic drive space (1). Leaks cause the reduction of effective gas mass m in Eq. (5) that carries heat from a cold bath and the limitation of the expansion effect defined by leakage-reduced pressure p_3 Further, the leakage of gas from the drive space to the expansion space (Figs. 3 and 7) limits the expansion effect by increasing p_4 . The effect of these negative factors on T_c is more complicated and could be precisely evaluated in future work. To account for the change in gas mass and pressure due to leaks, we use:

$$\triangle Q_\mathrm{c}^\mathrm{ideal} = mfRT_c \cdot \ln\left(\frac{V_4}{V_3}\right) - \bar{m}_{3-4} fRT_c \ln\left(\frac{p_3'}{p_4'}\right) \tag{6}$$

where $\bar{m}_{3-4} = \left(m_3' + m_4'\right)/2$ is the mass of the refrigerant gas during the expansion and for simplicity we assumed it to be equal to the arithmetic average of the gas masses before and after the expansion. Using Eqs. (5) and (6) and the ideal gas model we obtain::

$$\frac{\triangle Q_\mathrm{c}^\mathrm{ideal}}{Q_\mathrm{c}^\mathrm{ideal}} = 1 - \frac{\bar{m}_{3-4}}{m} \cdot \frac{\ln\left(\dfrac{m_3'}{m_4'} \cdot \dfrac{V_4}{V_3}\right)}{\ln\left(\dfrac{V_4}{V_3}\right)} \tag{7}$$

where $V_3 = 0.1 \times 10^{-3}\ \mathrm{m}^3$ being the total refrigerator volume before the expansion and $V_3 = 0.14 \times 10^{-3}\ \mathrm{m}^3$ being the total refrigerator volume after the expansion. Expressions for m_3' and m_4' read:

$$m_3' = m - \triangle m_c \tag{8}$$

$$m_4' = m_3' + \triangle m_e \tag{9}$$

with Δm_{c} and Δm_{e} being the average masses of leaked gas during the compression and expansion correspondingly, each with duration $\gamma = 1/(2f) = 0.055\ \mathrm{s}$ for the shaft frequency $f = 9.1\ \mathrm{Hz}$. Our approaches to calculate Δm_{c} and Δm_{e} are similar and based on the estimation of the average volumetric leakage for the measured (see Fig. 5) dependence of q from Δp_{a}. During the operation the gauge pressure over the seal changes from 0 MPa to 1 MPa. We use area averaging of the extrapolated quadratic function $q(\Delta p_{\mathrm{a}})$ to estimate the average volumetric gas leakage. We consecutively estimate the average masses of leaked gas by knowing γ.

II.5 Break-away force

The break-away force is the force required to overcome the adhesion of the PCP and to start piston sliding [39]. We measured the break-away force for the non-running refrigerator due to technical difficulties associated with measuring friction losses during the operation of the refrigerator. The break-away force was obtained for Seal 1. We selected Seal 1 with NLGI 3 because it has higher viscosity and, as a result, higher break-away force than Seal 2 with NLGI 2. In other words, Seal 1 characterizes a worst-case scenario. First, we measured the total break-away force F_{tot} for both the mechanical drive and the mounted PCP and second − the break-away force F_{m} for the mechanical drive without the PCP. The difference between the two values characterized the PCP break-away force $F_{\mathrm{PCP}} = F_{\mathrm{tot}} - F_{\mathrm{m}}$. For each type of measurements, the break-away force was measured at different shaft angles, every 60°, using a digital weight scale with accuracy class 1.5, calibrated before the measurements. The lever was connected to the shaft flywheel. The weight scale was connected to the lever and pulled smoothly by a hand at the angle of 90°. The force applied to the lever through the weight scale overcame the static friction to start the flywheel rotation. The maximum force value was recorded as a break-away force. The distribution of this force was assumed to be normal [40]. For each angle the total number of measurements of the break-away force $N = 50$ was conducted to obtain a coefficient of variation below $CV = \sigma/\mu = 0.3$, which indicated the measurement outcome with acceptable accuracy [41]. The arithmetic average of the values for the break-away force $\bar{F}_{\mathrm{PCP}}$ at each angle was used to calculate friction losses.

II.6 Static friction moment and friction losses

To obtain the friction losses, the link between static and kinetic friction should be described first. Static friction coefficient μ_{s} for the analyzed dry surfaces ranges from 0.3 [42] to 0.4 [43] and for lubricated surfaces is found around 0.21 [42]. Kinetic friction coefficient μ_{k} for dry surfaces ranges from 0.15 to 0.19 [42]-[44] and for lubricated surfaces ranges from 0.05 to 0.15 [42], [44]. Therefore, the average ratio of kinetic and static coefficients for dry surfaces is $\left(\mu_{\mathrm{k}}/\mu_{\mathrm{s}}\right)_{\mathrm{d}} = 0.17/0.35 = 0.49$ and for lubricated surfaces is $\left(\mu_{\mathrm{k}}/\mu_{\mathrm{s}}\right)_{\mathrm{l}} = 0.1/0.21 = 0.48$. Using the ratio between the two friction coefficients, the first-order estimation of friction losses during the operation reads:

$$W_{\mathrm{f}} = 2\pi l f \frac{\mu_{\mathrm{k}}}{\mu_{\mathrm{s}}} \bar{F}_{\mathrm{PCP}} \tag{10}$$

where l represents the length of the lever. To relate the friction losses to the refrigerator ideal power input $W^{(\mathrm{ideal})}$, we calculated the ratio $W_{\mathrm{f}}/W^{(\mathrm{ideal})}$ with:

$$W^{(\mathrm{ideal})} = mfR\left(T_h - T_c\right)\ln\left(\frac{p_2}{p_1}\right) \tag{11}$$

We find $W^{(\mathrm{ideal})} = 415\ \mathrm{W}$. It is important to mention that under some conditions, the friction loss may reduce with the operating time because of the break-in and self-lapping wear processes between mating surfaces [33].

II.7 Stirling refrigerator operation

The third type of the PCP treatment was the operation of the PCP with Seals 1 and 2 within an actual SR, which we designed and built. This procedure involved the operation of the SR test rig.

II.7.1 Test rig principal diagram

The principal diagram and the external view of the SR test rig was depicted in Fig. 8, Chapter I.

II.7.2 Test methodology

To evaluate the performance of two lubricated PCP seals, the measurement of the cooling temperature was conducted under three different conditions. Condition 1 was with newly applied lubricants during three test runs, where each test was separated by at least four calendar days. This type of measurement demonstrated the lowest refrigeration temperature achievable with different seals, and indicated how different lubricants in seals resist dynamic and static loading. Condition 2 included the measurement of the cooling temperature for the PCP with Seal 1 for two different levels of the heating load inside of the thermally isolated thermal chamber, 0 and 30 W. With this approach, I evaluated the impact of increased heating load on the sealing performance. For Condition 3, the refrigerator temperature was measured during prolonged time periods for Seal 1 and Seal 2 to observe the temperature fluctuations caused by the performance of the seals. A small clearance and the friction between piston and cylinder could cause the production of metal powder, which could pollute the working fluid and affect the results of the measurement. However, the presence of the lubricant between the mating surfaces causes the absorption of possible powder, which minimizes the contamination of the working fluid by the metal powder.

The refrigerator's cooling temperature T_c was measured directly using a Chromel-Alumel thermocouple at the surface of the expansion cylinder. Additionally, three other temperatures were measured and monitored during the experiment in the refrigeration chamber. Fig. 8, Chapter I depicts temperature measurement thermocouples: T1 measured the cooling temperature T_c on the surface of the expansion cylinder; T2 and T3 measured the ambient temperatures T a,t and T a,b in the thermal chamber at the top and at the bottom of the chamber correspondingly; T4 was located to measure the surface temperature of the electrical heater plate T_e. Temperature of the cooling water T_h was measured before and after each experiment with T5. Other operational parameters of the SR were measured to be maintained at a constant level. They included, the charge pressure p , the shaft frequency f , and the cooling water flow rate m_c .

II.8 Heating load

Eq. (12) describes how the heating load was measured during the experiment. The approach was similar to the one presented in [15]. Here I determine the heating load $Q_h^{(eq)}$ generated by the electrical heater in the thermal chamber at equilibrium time t_{eq} , from the values of direct current and voltage in the electrical heating circuit, Fig. 8, Chapter I, positions (1)–(3), (7), and (8). I selected two types of heating loads 0 W and 30 W because they best represent the operational cooling conditions in the chamber with a reduced effect of radiative heat transfer from the heater surface. Equation for heating load reads:

$$Q_h^{(eq)} = U_{dc} \cdot I_{dc} \qquad 12$$

Table 2, Chapter I summarizes measured parameters in the test rig with information about their accuracy. The measured refrigerator temperature was plotted against the times axis to evaluate the performance of seals.

III RESULTS

III.1 GAS LEAKAGE AND SEALING GAP VALIDATION

Table 2 shows raw measurement data for the leakage rate in the control volume. Using Eq. (1), the volumetric leakage rate is obtained in the sealing gap of the PCP depicted in Fig. 5. These results were used to validate the real size of the sealing gap that is required to sustain the measured leakage rates. The average leakage rate for the ideal gauge pressure range, from 0 to 1 MPa, was found to be $q_{\mathrm{c}} = 18.6\times10^{-6}\ \mathrm{m}^3\cdot\mathrm{s}^{-1}$ and corresponds to the gauge pressure 0.63 MPa. For the reduced compression pressure $p_2' = 3.47\ \mathrm{MPa}$, $q_{\mathrm{c}} = 16.8\times10^{-6}\ \mathrm{m}^3\cdot\mathrm{s}^{-1}$. Table 3 shows the values of gas parameters in order to calculate the ideal cooling capacity loss. Using Eq. (7) we find that $\Delta Q_{\mathrm{c}}^{(\mathrm{ideal})} = 2.2\%$. The average calculated gap using Eq. (2) for the concentric and eccentric PCP positions among ten calculated values was equal 11.6 μm. Fig. 6 depict Reynolds (a) and Euler (b) numbers for the air and helium PCPs. The dimensionless numbers were calculated for two limit PCP positions − concentric and eccentric − with the validated 11.6 μm gap. The results indicate that experimental measurements of the air PCP could model the operation of the helium PCP in the gauge pressure range of 0.3 MPa and 0.6 MPa, where the geometric and the dynamic similarity is satisfied; see Eqs. (3) and (4) . Based on the analysis of similarity conditions, Fig. 5 depicts linear and quadratic extrapolations of the leakage rate for measurements obtained between 0.3 and 0.6 MPa gauge pressure. Eq. (2) is the adapted Darcy–Weisbach equation for the flow in a concentric gap, where the leakage rate is linearly dependent from the gauge pressure. Therefore, a linear interpolation was applied. On the other hand, the empirical measurements of the leakage rate suggest that the dependence is closer to a quadratic form, thus the corresponding quadratic interpolation was also used. The resulting expected range of leakage rate for the helium PCP with the gauge pressures reaching 1 MPa can be expected between $27\times10^{-6}\ \mathrm{m}^3\cdot\mathrm{s}^{-1}$ and $46\times10^{-6}\ \mathrm{m}^3\cdot\mathrm{s}^{-1}$.

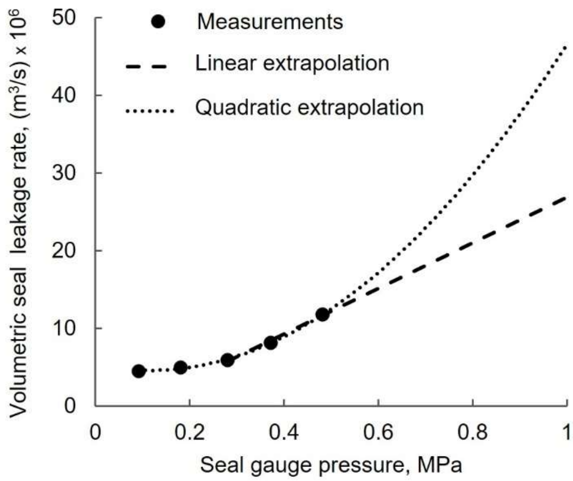


Fig. 5. Measured leakage rate for dry (no lubricant) PCP from varying pressure gradient. Dotted and dashed lines are the quadratic and linear extrapolations correspondingly for experimental points.

Table 2. Measurements of leakage rate in the control volume.

Δp_a (MPa)	τ (s)	q_{cv} $(\mathrm{ml\cdot s^{-1}})$
0.09	795	6.54
0.18	552	9.42
0.28	374	13.9
0.37	228	22.8
0.48	132	39.4

Table 3. Results of calculating gas leakage parameters based on the measurements of the leakage rate.

Parameter	**Value**
m (kg)	$0.583\cdot 10^{-3}$
m_c $(\mathrm{m^3\cdot s^{-1}})$	$16.8\cdot 10^{-6}$
m_e $(\mathrm{m^3\cdot s^{-1}})$	$16.8\cdot 10^{-6}$
ρ_c^g $(\mathrm{kg\cdot m^{-3}})$	4.65
ρ_e^g $(\mathrm{kg\cdot m^{-3}})$	3.63
γ (s)	0.055
Δm_c (kg)	$4.3\cdot 10^{-6}$
Δm_e (kg)	$3.4\cdot 10^{-6}$
m_3' (kg)	$5.79\cdot 10^{-4}$
m_4' (kg)	$5.82\cdot 10^{-4}$
$\overline{m}_{3-4}$ (kg)	$5.80\cdot 10^{-4}$
p_2' (Pa)	$3.47\cdot 10^{6}$
p_3' (Pa)	$2.11\cdot 10^{6}$
p_4' (Pa)	$1.53\cdot 10^{6}$

III.2 Friction losses

The measured and adjusted after calibration average static friction force of the shaft with dismounted PCP was 2.92 N with a standard deviation of 0.39 N. The value for the mechanical drive with the shaft and the mounted NLGI 3 lubricated PCP was 6.92 N with a standard deviation of 0.22 N. Subtracting these values, we found that the static friction force of the lubricated PCP, $\overline{F}_{\mathrm{PCP}}$, without the shaft friction, was 3.99 N with a standard deviation of 0.29 N. Using Eq. (11), the value of ratio for dynamic and kinetic friction coefficients, a lever length $l = 0.31\,\mathrm{m}$, and an operational frequency of the lubricated PCP $f = 9.1\,\mathrm{Hz}$ the friction losses for the lubricated PCP only, is equal to 33.9 ±2.5 W. The estimated friction losses for the shaft and lubricated PCP was 58.9 ±1.9 W and for the shaft only, was 25 ±3.1 W. Given that the ideal power input to the cycle is 415 W, the percentage of friction losses in the PCP only is equal to 8.1%, and of total friction losses is 14.1%.

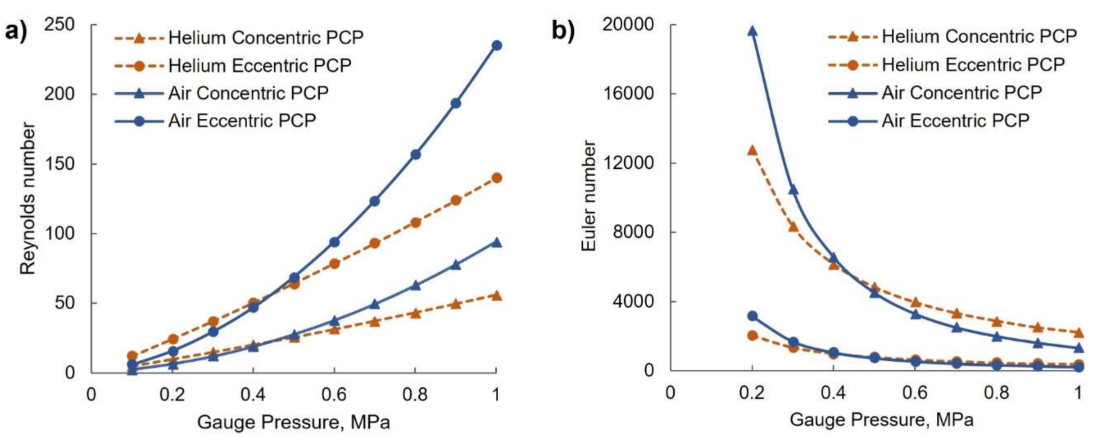


Fig. 6. Calculated Reynolds (a) and Euler (b) Numbers for concentric and eccentric PCP with air and helium as working medium. Absolute pressure after the gap for air PCP is 101.3 kPa and for helium PCP is 2.5 MPa.

III.3 Refrigerator operational temperature

For Condition 1, the NLGI 2 lubricant demonstrated better performance in two categories: the lowest refrigerator's temperature, and the highest resistance to the dynamic loading during operation and static loading during between-tests calendar time. The implementation of NLGI 2 resulted in the lowest refrigerator temperature −125ºC and the increase of the refrigerator temperature 0.5°C per day, comparing to the NLGI 3 with the similar values of −105ºC and 3.6°C per day. The results of the SR operation at Condition 2 also show that an increased heating load insignificantly affects the performance of the high-tolerance NLGI 3 lubricated PCP. The fitting curves for the heating load 0 W and 30 W are almost parallel, which implies that the increase of heating load in a given range does not play a significant effect on the performance of the PCP. Fig. 7 depicts results for Condition 3. The results show how the operational temperature of the refrigerator changes during operation for: the PCP with the NLGI 2 lubricant and 10 cumulative hours in operation; the PCP with NLGI 3 and 15 cumulative hours of operation; and the PCP with the NLGI 2 and 19 cumulative hours of refrigerator operation. The refrigerator temperature changed on two levels: micro-changes within the time scale below 15 min and macro-changes – above 15 min. Micro-changes of the refrigerator temperature vary from 1 °C to 14 °C. These temperature fluctuations indicated immediate changes in the lubricated PCP that most likely happened in the structure of the lubricant under dynamic loading of moving mating surfaces and changing gauge and static pressures. Macro-changes are related to the temperature change over a longer period of time, for example for NLGI 2, 19 refrigerator hours, where the refrigerator temperature has increased from −96°C to −52°C within 135 min with a fresh lubricant. Although Fig. 8 alone is not enough to characterize performance of seals, it indicates a possible reason for the macro-changes of the refrigerator temperature. Both the expansion (a) and compression (b) pistons wore out significant areas of their surfaces (highlighted with white) under the side forces from the Ross-yoke mechanism. As a result, the gap between mating surfaces increased and enabled higher leakage rates and reduced compression. Further, the circumferential increase of the gap at the upper and the lower areas of the piston surface likely could cause a pressure gradient inside the gap that had squeezed the lubricant out during the operation (Fig. 9), thus further reducing the sealing capability of the lubricant.

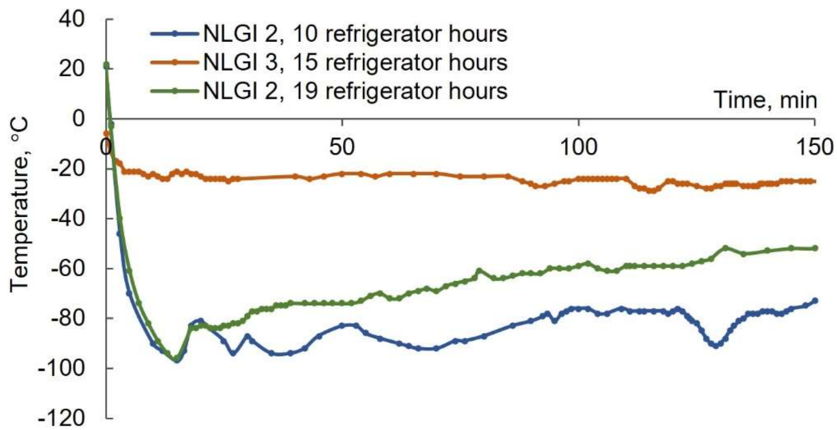


Fig. 7. Refrigerator temperature during operation for different lubricant and after different cumulative operation time of the refrigerator: "10 refrigerator hours" means that before this test the total cumulative operation of the PCP was 10 h.

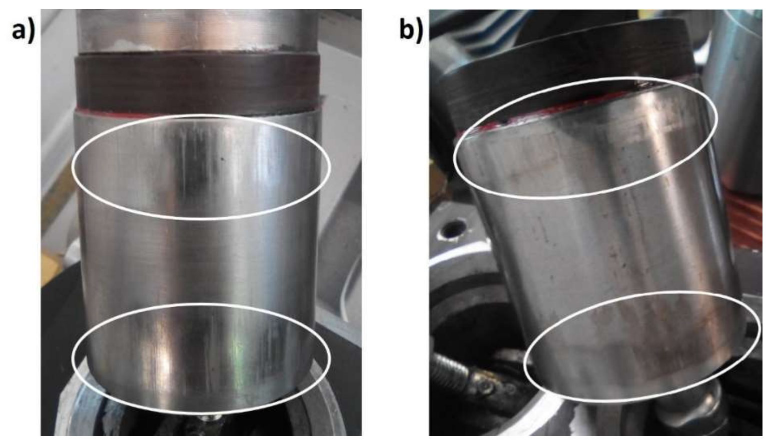


Fig. 8. Expansion (a) and compression (b) piston with worn-out areas on the surface (white).

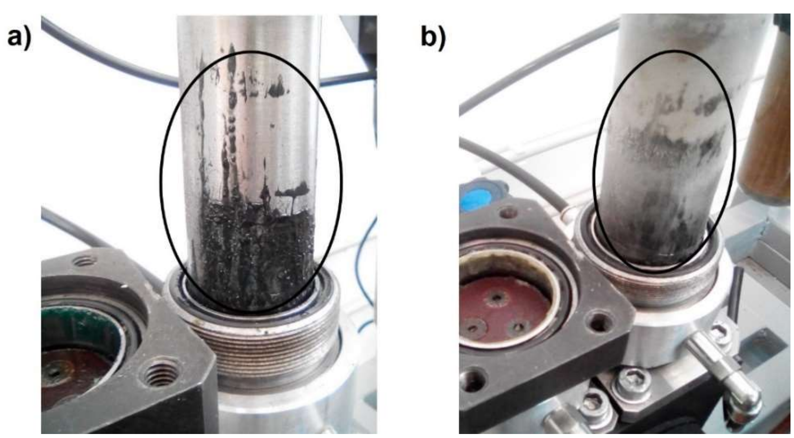


Fig. 9. Displacer surface covered with NLGI 2 (a) and NLGI 3 (b) lubricant (black), which was squeezed out from the gap during operation.

IV DISCUSSION

IV.1 PISTON-CYLINDER SEALING IN STIRLING REFRIGERATORS

This study sought to understand experimentally fundamental processes that affect the high-tolerance configuration of the sealing mechanism. The results show that a high-tolerance lubricated (grease, NLGI 2) piston-cylinder pair made of gray cast iron (piston) and AISI 420s steel (cylinder) has an operating life of 15 working hours when operated at 9.1 Hz frequency in an alpha Stirling refrigerator with the Ross-yoke drive with 1 mm maximum misalignment between piston and cylinder axes. Our results also show that in order to implement a high-tolerance PCP successfully in SRs, several practical questions need to be addressed: the selection or development of advanced lubricant materials that can withstand dynamic and static loading during the refrigerator operation; the selection of materials for piston and cylinder mating surfaces that can withstand lateral forces of the Ross-yoke mechanical drive; the manufacturing of Ross-yoke mechanical drive with piston and cylinder axis misalignment less than 1 mm, the precise manufacturing to insure a minimal gap δ between mating surfaces; and the optimal D/L ratio and shaft frequency f to minimize gas leakages in the gap. Study of higher shaft frequencies and dry contact between mating surfaces is the subject of future work.

This chapter focused on the performance of the high-tolerance PCP. However, during the experimental optimization of the piston-cylinder pair, a different type of the seal was tested – a dry friction seal with piston rings containing 75% PTFE and 25% coke (Fig. 2b). Key performance characteristics of the Stirling refrigerator with the PTFE piston rings were reported in Chapter I, Table 1 and Fig. 6. In Table 4, I compare two tested PCP types under charge pressure of 2.5 MPa and shaft frequency 550 rpm.

High-tolerance seal has 7.6 times lower friction losses and 1.7 times lower cost, compared with the tested dry-friction seal, which shows significant efficiency advantage and considerable cost benefit for the high-tolerance PCP. However, dry-friction seal demonstrated outstanding stability of the refrigerator temperature over separate and long test runs, as well as considerably lower refrigeration temperature after 19 refrigerator hours (Fig. 11). Two peaks

Table 4. Comparison of tested high-tolerance seal and dry-friction seal.

Parameter	High-tolerance seal	Dry-friction seal
Seal type	"Cast iron – polished stainless steel with 0.32 μm roughness", 10 μm clearance gap with 60 mm length, Fig. 2e	"PTFE piston rings – polished stainless steel with 0.32 μm roughness", two rider and two expander rings made of 75% PTFE and 25% Coke, dry contact, Fig. 2b
Average friction losses W_f , W	34	260
Temperature increase between separate test runs (Condition 1), °C per day	0.5	–2
Micro-changes of the refrigerator temperature during a 150 min test run after 19 refrigerator hours, °C	14	<1
Refrigerator steady-state temperature after 19 refrigerator hours, °C	–52	–165
Cost of materials and manufacturing, $	150	250

in temperature measurements at 30 min and 50 min are attributed to the malfunction of the air fan, which was operating in the thermal chamber, and not to the instability on the dry-friction PCP. The temperature increase between separate tests of –2°C per day indicates that the refrigerator temperature was not increasing but instead decreasing with each new test: there was three independent test runs over 24 hours each with consequently achieved temperatures –143°C, –146°C and –147°C. This reduction of temperature could happen because the surface of the cylinder wall was covered during the operation with the layer of carbon, which was present in the structure of piston rings. This layer could improve the sealing performance of the PCP. An alternative explanation is the effect of uncontrollable external environment

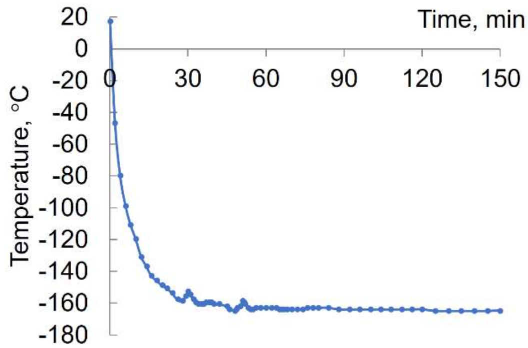


Fig. 11. Refrigerator temperature during operation with the dry-friction seal after the cumulative operation of 19 refrigerator hours.

factors, such as room and cooling temperatures. However, before each test, the room temperature was 24°C and the temperatures of the cooling water were 21°C, 20°C, and 21°C correspondingly – variations that are unlikely to cause the reduction of refrigerator temperature by 4°C between the first and the third tests. The relative improvements in the operation of the dry-friction seal indicate a higher compression rate and lower leakage. The exact value of the leakage for the dry-friction seal was not measured. The operating life of the dry-friction seal, unlike for the high-tolerance seal, extends well beyond 19 refrigerator hours, given the observed stability of the refrigerator temperature. The absent lubrication for the dry-friction seal prevents possible evaporation of a lubricant and the contamination of internal refrigerator channels. Also, the design of flexible expander rings did not pose high manufacturing tolerances to the mechanical drive. Despite high friction losses and higher cost, dry-friction seal is a reliable sealing design alternative as an intermediate step before a more reliable high-tolerance seal could be developed. The latter demonstrates exceptional efficiency and lower cost and therefore deserves further research studies. This result shows that experimental optimization of critical components in the Stirling refrigerator can significantly improve the operational performance of the system and use it in larger number of market applications. The operational data for the dry-friction PCP collected during the work reported in this chapter was used in the next chapter to validate the physical model of the Stirling refrigerator system for scaling for different commercial applications.

IV.2 Design of energy conversion technologies

The example of operating a real Stirling refrigerator and experimentally optimizing its piston-cylinder seal with successful outcomes shows that this methodology can be applicable to other types of ECT. All ECT have a material form, in which some elements affect the performance of the system more than other components. Some of these high-priority elements also may operate at stressful conditions, where mathematical prediction has low accuracy. For such elements, it is important to perform experimental optimization. This approach would allow saving time and resources on inaccurate modeling and quickly increase the performance of the ECT system to satisfy commercial requirements for more market segments.

IV.3 Novelty and Contribution

- Systematized the knowledge about existing designs of piston-cylinder seals with advantages and disadvantages;
- Studied experimentally the performance of a high-tolerance and dry-friction PTFE seals for the piston-cylinder pair in a real Stirling refrigerator with the alpha configuration of the Ross-yoke mechanical drive;
- Obtained the performance of the high-tolerance piston cylinder seal with the 10 μm gap;
- Demonstrated how the performance of the Stirling refrigerator can be improved through the experimental optimization of the sealing mechanism.

V CONCLUSIONS

- The ideal cooling capacity losses due to gas leakage was estimated to be 2.2%. However, in the long term, the wear of seals is expected to increase the ideal cooling capacity loss;
- The ratio between measured friction losses in the piston-cylinder pair and the ideal power input of the Stirling refrigerator was estimated to be 8.1%. We can expect the reduction of friction losses with the operating time because of the break-in and self-lapping wear processes between mating surfaces;
- A NLGI 2 lubricant enabled 12% lower refrigerator temperature than a NLGI 3 lubricant (−125ºC and −105ºC) under similar operational conditions. The NLGI 2 lubricant also demonstrated seven times higher resistance to dynamic loading during operation and static loading during and between test intervals, which caused gradual increase of the refrigerator temperature;
- The increase of electrical heating load from 0 to 30 W has not significantly affected the operation of the lubricated PCP;
- During the operation, the lubricated PCP produced instantaneous temperature fluctuations with maximum temperature amplitude 14 °C that indicated unstable behavior for both types of studied lubricants in the PCP;
- The cumulative refrigerator operation with the lubricated cast iron-steel PCP exceeding 15 h has circumferentially worn out upper and lower regions of pistons and caused the 44°C increase of refrigerator temperature within 135 min of continuous operation;
- The operation of the dry friction piston rings instead of the high-tolerance sealing demonstrated the lowest cooling temperature down to −165°C without its increase after 19 hours of cumulative operation;

- The operational data for the dry-friction PCP shall be used in the next chapter to validate the physical model of the Stirling refrigerator system for scaling for different commercial applications.

REFERENCES

[1] Mishra, R.S., 2017. Thermal modelling of three-stage vapour compression cascade refrigeration system using entropy generation principle for reducing global warming and ozone depletion using ecofriendly refrigerants for semen preservation. *Int. J. Res. Eng. Innovation*, Vol. 1, pp. 22–28.

[2] Chen, G., Wang, W., 2007. Role of freeze drying in nanotechnology. *Drying Technol.*, Vol. 25, pp. 29–35.

[3] Legett, R., 2014. Field Demonstration of High-efficiency Ultra-low temperature Laboratory Freezers. Navigant Consulting, Inc . Better Building Alliance - *U.S. Department of Energy*.

[4] Gieseler, H., Gieseler, M., 2017. The importance of being small: miniaturisation of freeze drying equipment. *Eur. Pharm. Rev.*, Vol. 22, pp. 28–32.

[5] Berchowitz, D.M., Kwon, Y., 2012. Environmental profiles of Stirling-cooled and cascade-cooled ultra-low temperature freezers. *Sustainability*, Vol. 4, pp. 2838–2851.

[6] Haywood, D., Raine, J.K., Gschwendtner, M., 2002. Stirling-cycle heat pumps and refrigerators a realistic alternative? In: *Institute of refrigeration, heating & air conditioning engineers (IHRACE) conference*. Christchurch, New Zealand.

[7] Hermes, C.J., Barbosa Jr Jr, J.R., 2012. Thermodynamic comparison of peltier, Stirling, and vapor compression portable coolers. *Appl. Energy*, Vol. 91, pp. 51–58.

[8] Chaudhari, P., D'Souza, D., Borkar, S., Haribhakta, V., Trimbake, S., 2016. Mathematical formulation of free piston Stirlingcooler for domestic refrigeration using loss factor. *Int. J. Scientif. Eng. Res.*, Vol. 7, pp. 1505–1510.

[9] Qian, S., Nasuta, D., Rhoads, A., Wang, Y., Geng, Y., Hwang, Y., Radermacher, R., Takeuchi, I., 2016. Not-in-kind cooling technologies: a quantitative comparison of refrigerants and system performance. *Int. J. Refrig.*, Vol. 62, pp. 177–192.

[10] Song, C., Lu, J., Kitamura, Y., 2015. Study on the COP of free piston Stirling cooler (FPSC) in the anti-sublimation CO 2 capture process. *Renewable Energy*, Vol. 74, pp. 948–954.

[11] Hachem, H., Gheith, R., Aloui, F., Nasrallah, S.B., 2017. Optimization of an air-filled beta type Stirlingrefrigerator. *Int. J. Refrig.*, Vol. 76, pp. 296–312.

[12] Huang, Y., Wang, B., Cheng, L., Wu, J., Wang, R., 2016. Cooling performance measurement of the reverse application of a coaxial free-piston Stirling engine. *Sci. Technol. Built Environ.*, Vol. 22, pp. 556–564.

[13] Janssen, M., Beks, P., 2002. Measurement and application of performance characteristics of a free piston Stirling cooler. In: Paper 539. *Ninth international refrigeration conference at Purdue University*, West Lafayette, IN, USA - July, pp. 16–19. 2002.

[14] Le'an, S., Yuanyang, Z., Liansheng, L., Pengcheng, S., 2009. Performance of a prototype Stirling domestic refrigerator. *Appl. Thermal Eng.*, Vol. 29, pp. 210–215.

[15] Oguz, E., Ozkadi, F., 2000. An experimental study on the refrigeration capacity and thermal performance of free piston Stirling coolers. In: Paper 520. *Eighth international refrigeration conference at Purdue university*, West Lafayette, IN, USA - July, pp. 25–28

[16] Otaka, T., Ota, M., Murakami, K., Sakamoto, M., 2002. Study of performance characteristics of a small Stirling refrigerator. *Heat Transf. - Asian Res.*, Vol. 31, pp. 344–361.

[17] Smirnov, D., Golkar, A., 2015. Stirling engine systems tradespace exploration framework. *Procedia Comput. Sci.*, Vol. 44, pp. 558–567.

[18] Baumüller, A., Laug, A., 2002. *Kolben und zylinder für einen Stirling-motor*. In: Europaisches patent am EP 1239137b1.

[19] Nilsson, H., 1983. The united Stirling 4–95 and 4–275 engines in underwater use. In: *Proceedings of the intersociety energy conversion engineering conference*, pp. 850–855. Orlando, FL, USA. 21 August 1983.

[20] Smedley, G., 2004. Piston ring design for reduced friction in modern internal combustion engines (doctoral dissertation, *Massachusetts institute of technology*).

[21] Carlsen, H., 1994. Operating experience with 10 kw hermetic Stirling engine. In: *Proceedings of the intersociety energy conversion engineering conference*, Monterey, pp. 1911–1916., CA, USA, 7-12 August 1994.

[22] Carlsen, H., Ammundsen, N., Traerup, J., 1996. 40 kW Stirling engine for solid fuel. In: *Proceedings of 31st Intersociety Energy Conversion Engineering Conference*, Vol. 2, pp. 1301–1306

[23] Matsuguchi, A., Kagawa, N., 2014. Study on Performance Improvement of a Double-acting Stirling Engine (performance evaluation of a new-type matrix geometric pattern), *In Memoirs of the National Defense Academy*, Japan, Vol. 54, pp. 1–11.

[24] Beale, W. T., 1989. *Sliding surface lubrication particularly advantageous for a free piston Stirling engine*. U.S. Patent 4802332.

[25] Berchowitz, D. M., & Kwon, Y. R., 2013. *Bearing support system for free-piston Stirling machines*. U.S. Patent 8615993.

[26] Unger, R. Z., 2001. *Gas bearing and method of making a gas bearing for a free piston machine*. U.S. Patent 6293184.

[27] Bartolini, C. M., Naso, V., & Suraci, F., 1988. *Stirling engine*. U.S. Patent No. 4760698.

[28] Dhar, M., 1992. *Stabilizing gas bearing in free piston machines*. U.S. Patent 5140905.

[29] Stetson, N. B., 1991. *Miniature integral Stirling cryocooler*. U.S. Patent No. 5056317. Washington, DC: U.S. Patent and Trademark Office.

[30] Dvortsov, V. S., Tkachenko, M. M., Smirnov, D. S., Saichenko, A. S., & Kukolev, M. I., 2017. *Thermal motor*. Russian Patent No.RU 175182 U1. Federal Institute of Industrial Property 33, (in Russian).

[31] Ross, M. A., & Beale, W. T., 1988. Compact crank drive mechanism with guided pistons.

[32] Baart, P., van der Vorst, B., Lugt, P.M., van Ostayen, R.A. , 2010. Oil-bleeding model for lubricating grease based on viscous ow through a porous microstructure. *Tribol. Trans.*, Vol. 53, pp. 340–348.

[33] Barber, G.C., Ludema, K.C., 1987. The break-in stage of cylinder-ring wear: a correlation between fired engines and a laboratory simulator. *Wear*, Vol. 118, pp. 57–75.

[34] Acherkan, N.S., 1954. *Reference Book of Engineer in six Volumes, 2(2). Masgiz*, Moscow, (in Russian).

[35] Bashta, T.M., 1971. Machine Building Hydraulics. *In: Reference book. Mashinostroenie*, pp. 95–98, (in Russian).

[36] Braga, V.M., Deschamps, C.J., 2017. Numerical analysis of gas leakage in the piston–cylinder clearance of reciprocating compressors considering compressibility effects. *In: IOP conference series: Materials science and engineering,* Vol. 232. *IOP* Publishing.

[37] Zohuri, B., 2015. Dimensional analysis and self-similarity methods for engineers and scientists. Chapter 2, 109. Springer.

[38] Barilovich, V.A., Smirnov, U.A., 2010. Fundamentals of Technical Thermodynamics and the Theory of Thermal and Mass Exchange. *SPbSTU,* (in Russian).

[39] Olsson, H., Strm, K.J., De Wit, C.C., Gfvert, M., Lischinsky, P., 1998. Friction models and friction compensation. *Eur. J. Control*, Vol. 4, pp. 176–195.

[40] Lyon, A., 2013. Why are normal distributions normal? *British J. Philos. Sci.*, Vol. 65, pp. 621–649.

[41] Brown, C.E., 1998. Coefficient of variation. In: *Applied multivariate statistics in geohydrology and related sciences*. Springer, Berlin, Heidelberg, pp. 155–157.

[42] Klimenko, A.V., Zorina, V.M., 1999. Thermal energetics and thermal techniques: General questions. In: Reference textbook, pp. 222–222, *MEI,* (in Russian).

[43] Babichev, A.P., Babushkina, N.A., Bratkovsky, A.M., 1991. Physical values: reference textbook. *Energomashizdat*, pp. 128–129, (in Russian).

[44] Panov, V.A., Kruger, M.Y., Kulagin, V.V.,1980. Engineer reference textbook in optical and mechanical devices. *Mashinostroenie*, Leningrdad, 742, (in Russian).

CHAPTER VI
MODELING HEATING LOAD

The stage of testing and refinement is followed by the phase of production and scale-up. This stage was reached for the developed Stirling machine in the first half of 2019 with the lowest refrigeration temperature of −165°C (108 K) and the cooling capacity of 75 W. A number of requests from the industry created the need to evaluate the dimensions and performance characteristics of the system for different operational conditions and heating loads. The scaling of technology for different market applications became an important factor that motivated the development of the system's physical model. This chapter reports findings from the development of the scaling method for the Stirling refrigerator using the analytical modeling and experimental results. The operational data collected during the test and refinement stage (Chapter V) was used to validate the objective physical model.

I INTRODUCTION

The scaling of the system "Stirling refrigerator – thermal chamber" for different commercial applications, such as liquefaction of gasses, refrigeration cabinets, and cryocooling is the next important step after the construction of the experimental system. The analysis in Chapter IV showed that existing literature is still lacking the understanding of the joint operation between the Stirling refrigerator and various heating loads. Therefore, there is a need in an analytical model validated by the experimental results that would model this coupling and help to save costs in the process of scaling Stirling refrigerator for different commercial applications. This model helps to study the effects of different alternative design configurations of the Stirling refrigerator and the thermal chamber and optimize the design specifications. This chapter aims to build and validate such model for the in-house Stirling refrigerator.

II Methodology

There are two objectives for this study:

1) The development of a combined numerical model for the system "Stirling refrigerator - Thermal chamber" to predict steady state temperature of gas in the thermal chamber of a Stirling refrigerator.

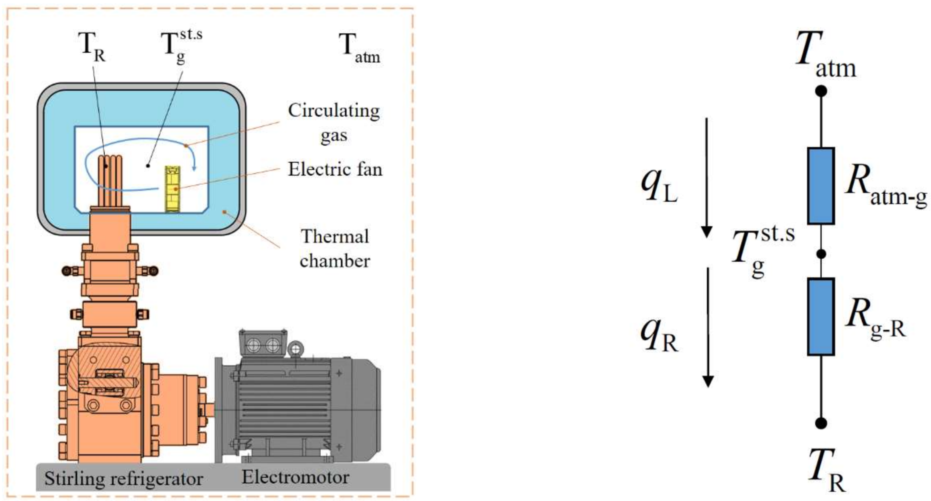
2) Carrying out real experimental tests for the objective system to collect temperature data and validate the numerical model.

The following sections discuss in details the proposed methodology to achieve these two goals.

II.1 Object of analysis

The object of analysis in this chapter is the thermo-mechanical system "Stirling refrigerator – Thermal chamber" (hereafter – System), which is shown in Fig. 1a. In the figure, T_{atm} is atmospheric temperature (K), T_{R} is refrigerator temperature (K), $T_{\mathrm{g}}^{\mathrm{st.s}}$ is steady state gas temperature in the thermal chamber (hereafter – TC) (K). An electromotor drives the Stirling refrigerator (hereafter – SR). The latter converts input mechanical work into cooling capacity and cools the gas in the TC. The heat transfer occurs due to forced convection initiated by the gas and the air, which is circulated in the chamber by the electrical fan.

Fig. 1b depicts the thermo-electrical analogy of System. I used this analogy to model thermal balance in the chamber. In Fig. 1b, q_{R} is cooling capacity of the SR (W), q_{L} is the sum of all parasite heat leaks from the atmospheric environment (W), $R_{\mathrm{atm\text{-}g}}$ is total thermal resistance of the system "atmosphere-gas" ($\mathrm{K \cdot W^{-1}}$) and $R_{\mathrm{g\text{-}R}}$ is thermal resistance of the system "gas-SR" ($\mathrm{K \cdot W^{-1}}$). Table 1 lists key operational, geometrical and thermal parameters of System used in the analysis.



a) b)

Fig. 1. Thermal system "Stirling refrigerator – Thermal chamber": a) principal diagram with key elements b) Thermo-electrical analogy.

Table 1. Characteristics of the Stirling refrigerator and the thermal chamber used in the analysis.

Stirling refrigerator	
Cooling capacity, q_R	50 W
Refrigeration temperature, T_R	123 K
Radiator temperature, T_r	287 K
Working fluid	Helium
Width × height × depth	262 × 474 × 205 mm
Charge pressure, p	2.5 MPa
Shaft frequency	9 Hz
Surface area of the cold heat exchanger	0.024 m^2
Thermal chamber	
Circulating gas	Air
External width × height ×depth	330 × 250 × 425 mm
Internal width × height ×depth	225 × 135 × 275 mm
External surface area	0.724 m^2
Internal surface area	0.259 m^2
Heat conductance of the wall	0.040 $W \cdot m^{-1} \cdot K^{-1}$

The objective was to find an expression for the steady state temperature of the gas in the chamber as a function of refrigerator control parameters and environment conditions.

II.2 Steady state gas temperature in the thermal chamber

II.2.1 Cooling capacity of the Stirling refrigerator

$$q_{\mathrm{R}} = S_{\mathrm{R}} \cdot \Theta \cdot f \cdot p \cdot \frac{T_{\mathrm{R}}}{T_{\mathrm{r}}} \tag{1}$$

In equation (1), which was adapted from the experimental work of Otaka *et al.*[1], S_{R} is design configuration constant (cm^3) defined below (2), Θ is a constant number introduced in [1], which according to the authors contains "the effects of phase-difference temperature ratio, volume ratio, and dead volume ratio" and hereafter called the Otaka number, f is frequency of the cycle, p is charge pressure (MPa), T_{R} is refrigeration temperature (K) and T_{r} is temperature of the refrigerator's radiator (K).

$$S_{\mathrm{R}} = V_{\mathrm{e}} \frac{\kappa \cdot \sin\alpha}{\xi} \cdot (\kappa + 1) \tag{2}$$

In equation (2), V_{e} is expansion volume (cm^3), κ ratio between compression and expansion volumes, α is phase difference between the moving compression and expansion pistons, ξ is dead volume ration between the total dead volume and the expansion volume. For the purpose of this study the parameter S_{R} is a constant number (Table 1) due to a fixed design of the experimental SR.

Table 2. Design parameters of the Stirling refrigerator.

Parameter	Value
Expansion volume, V_e, cm^3	60
Volume ratio, κ	1
Phase difference, α	90
Dead volume ration, ξ	1.33
Design configuration constant, S_{R}, cm^3	90.2 (calculated)

Equation (1) and (2) define the relation between the cooling capacity of the Stirling refrigerator and its design and control parameters. The advantages of this method is simplicity. The drawback is the need to experimentally obtain the constant Θ. Numerical calculation of this parameter is possible, but difficult, because the Otaka number is an aggregate characteristic of the refrigerator performance. It depends on a specific design of the refrigeration system and to the author's knowledge, there are no research studies that offered a method to calculate it numerically. The experimental results may however suggest a more general approach to find Θ for any design of the SR using the relation between the temperature ratio in (1) and the ratio $\Theta_s = \Theta / (S_R \cdot f \cdot p)$, which is called the specific Otaka number (W^{-1}). This parameter is independent from design configurations of the Stirling refrigerator characterized by S_R, p and f. Therefore, Θ_s describes more fundamental factors in the operation of the SR, which could be relevant to different design configurations and not only to the design described in Table 1 and 2. In the results section we find the relation between Θ_s and T_R / T_r.

II.2.2 STEADY STATE FOR THE REFRIGERATOR AND THE GAS IN THE THERMAL CHAMBER

$$q_R = \frac{T_g - T_R}{R_{gR}} = \frac{T_g - T_R}{1/(h_{gR} \cdot A_R)} \tag{3}$$

In equation (3), T_g is gas average temperature in the thermal chamber (K), $h_{g\text{-}R}$ is heat transfer coefficient between gas and the refrigerator, $W \cdot m^{-2} \cdot K^{-1}$, A_R is area of the refrigerator heat exchanger surface (m^2).

II.2.3 STEADY STATE FOR THE ATMOSPHERE AND THE GAS IN THE THERMAL CHAMBER

$$q_L = \frac{T_{atm} - T_g}{R_{atm\text{-}g}} = \frac{T_{atm} - T_g}{\dfrac{L_w}{k_w A_{w1}} + \dfrac{1}{h_{w2} A_{w2}}} \tag{4}$$

In equation (4), total thermal resistance of the system "atmosphere-gas" $R_{atm\text{-}g}$ includes: 1) thermal resistance of the thermal chamber wall defined by thickness L_w (m), heat conductance k_w ($W \cdot m^{-1} \cdot K^{-1}$), and external freezing chamber surface area A_{w1} (m^2) and 2) thermal resistance between the internal freezing chamber wall and the circulating gas in the chamber defined by the heat transfer coefficient between the wall and gas h_{w2} ($\mathrm{W \cdot m^{-2} \cdot K^{-1}}$) and the surface area of the internal freezing chamber wall A_{w2} ($\mathrm{m^2}$).

II.2.4 STEADY STATE GAS TEMPERATURE IN THE THERMAL CHAMBER

At the steady state, the gas temperature inside the thermal chamber stops changing because the heat flow rate created by heat pumping from the thermal chamber is compensated by the parasite heat flow rate from the atmospheric environment.

$$q_{\mathrm{R}} = q_L \tag{5}$$

Using equations (3), (4) and (5) we find T_{g}:

$$T_g = \frac{T_{\mathrm{atm}} \cdot R_{\text{g-R}} + T_{\mathrm{R}} \cdot R_{\text{atm-g}}}{R_{\text{g-R}} + R_{\text{atm-g}}} \tag{6}$$

Using equations (1) and (3) we find T_{R}:

$$T_{\mathrm{R}} = \frac{T_{\mathrm{g}}}{R_{\text{g-R}} \cdot S_{\mathrm{R}} \cdot \Theta \cdot f \cdot p \cdot \frac{1}{T_{\mathrm{r}}} + 1} = \frac{T_{\mathrm{g}}}{R_{\text{g-R}} \cdot \mathrm{C} + 1} \tag{7}$$

Combining (6) and (7) we obtain steady state gas temperature in the thermal chamber $T_{\mathrm{g}}^{\mathrm{st.s}}$:

$$T_{\mathrm{g}}^{\mathrm{st.s}} = \frac{T_{\mathrm{atm}} \cdot R_{\text{g-R}}}{R_{\text{atm-g}} + R_{\mathrm{gR}} - \dfrac{R_{\text{atm-g}}}{R_{\text{g-R}} \cdot \mathrm{C} + 1}} \tag{8}$$

II.3 CALCULATION OF THERMAL RESISTANCES

To calculate the steady state temperature in Expression (8), one needs to calculate the thermal resistances between the gas and the Stirling refrigerator $R_{\text{g-R}}$ and between the atmosphere and the gas in the cooling chamber $R_{\text{atm-g}}$.

II.3.1 THERMAL RESISTANCE BETWEEN THE GAS AND THE STIRLING REFRIGERATOR

I first calculate the heat transfer coefficient between gas and the refrigerator $h_{\text{g-R}}$. The cold surface of the Stirling refrigerator is a bank of plain tubes. Therefore, I used a classical approach to calculate $h_{\text{g-R}}$ from Gnielinski *et al.* [2]. For this calculation, one needs to know the geometry of heat exchangers, the table properties of the gas in the thermal chamber and its main flow velocity. I measured geometry directly, used the table properties for air, and obtained the gas velocity in the chamber experimentally.

To find the main flow velocity, I conducted three measurements in six different locations of the thermal chamber using the CEM DT-318 Flexible thermo-anemometer with the accuracy ±3%. The measurements were conducted with an opened thermal chamber to access the internal volume with the anemometer. During the operation, the chamber is closed and the actual velocity values may differ. However, the measurements were conducted at the bottom of the chamber in the main flow of the air flow; therefore, the expected difference should not affect the results significantly.

II.3.2 THERMAL RESISTANCE BETWEEN THE ATMOSPHERE AND THE GAS

The thermal resistance of the chamber wall has had a varying thickness, and I used the average value $L_{\text{w}} = 62\ \text{mm}$. The wall consisted of different types of isolation materials; therefore, the heat conductance of the wall material was averaged by thickness and was equal $k_{\text{w}} = 0.04\ \text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$. The external thermal chamber surface area was $A_{\text{w1}} = 0.724\ \text{m}^2$.

To calculate thermal resistance between the internal chamber wall with $A_{\text{w2}} = 0.259\ \text{m}^2$ and the gas in the chamber, one needs to calculate the heat transfer coefficient h_{w2}. To do so, I used a classical relation to find the Nusselt number:

$$\text{Nu}_{\text{w2-g}} = 0.037\cdot\text{Re}_{\text{w2-g}}^{4/5}\cdot\text{Pr}_{\text{w2-g}}^{1/3} \tag{9}$$

II.4 Evaluation of model accuracy

To evaluate how accurately the developed model predicts temperatures comparing to experimental measurements, I selected three accuracy metrics. These metrics are the coefficient of variance, mean biased error, and the coefficient of determination.

Coefficient of variance (CV) in (10) is the average magnitude of error. It shows the average difference between what the model predicted and what was measured in a real experiment. This parameter does not account for the sign of those differences (square). It penalizes larger errors because of the square. The smaller the CV, the better the prediction.

$$\mathrm{CV}=\frac{\sqrt{\frac{1}{n-1}\sum_{i=1}^{n}\left(x_i-\hat{x}_i\right)^2}}{\overline{x}_i}\cdot 100 \tag{10}$$

In Expression (10), CV – the coefficient of variance, n – the number of experimental points corrected with the Bessel's correction ($n-1$ instead of n), x_i – measured parameter value, $\hat{x}_i$ – predicted parameter value, and $\overline{x}_i$ – mean value of the measured parameter. In this work, x_i takes values of $T_{\mathrm{g}}^{\mathrm{st.s}}$ and T_{R}.

Mean biased error (MBE) in (11) characterizes the average magnitude of a systematic error. Unlike for CV, similar positive and negative errors in MBE would cancel each other out. The smaller the MBE, the lesser systematic errors in the model.

$$\mathrm{MBE}=\frac{1}{n-1}\cdot\frac{\sum_{i=1}^{n}\left(x_i-\hat{x}_i\right)}{\overline{x}_i}\cdot 100 \tag{11}$$

Coefficient of determination (R^2) in (12) shows how the squared prediction error of the proposed model (nominator) is related to the squared prediction error of the model that always predicts the mean value of measurements (denominator). It ranges from 0 to 1, where 0 means the worst fit and 1 means the perfect fit.

$$R^2 = 1 - \frac{\sum_{i=1}^{n} \left(x_i - \hat{x}_i \right)^2}{\sum_{i=1}^{n} \left(x_i - \overline{x}_i \right)^2} \tag{12}$$

For acceptable accuracy, CV and MBE should be less than 10% and R^2 is more than 0.9.

II.5 Experimental measurements of the steady-state temperature

The goal of real experiments was to measure the steady state temperature in the thermal chamber of a real Stirling refrigerator for different charge pressures and shaft frequencies. The results of real experiments were used to validate the numerical model for a wide range of operational parameters.

II.5.1 Stirling refrigerator test rig

The principal diagram and the external view of our SR test rig is depicted on Fig. 8, Chapter I. The measurement instruments and accuracy are described in Table 2, Chapter I.

II.5.2 Preliminary test

Three independent tests were carried out at nominal parameters of the Stirling refrigerator — $p = 2.5\ \mathrm{MPa}$, $f = 550\ \mathrm{rpm}$ and $m_c = 5\,\mathrm{l} \cdot \mathrm{min}^{-1}$ — to obtain experimentally the Otaka number Θ as the arithmetic average. The constant was used in the Expression (7) to predict the steady-state gas temperature at different operational parameters of the Stirling engine.

II.5.3 Test methodology

The measurements comprised of 13 separate tests at different non-nominal operational parameters of the Stirling refrigerator. During each test the SR was operated at different charge pressures and shaft frequencies. For charge pressures 0.5 MPa and 1.5 MPa the SR was operated at 110 rpm, 220 rpm, 330 rpm, 440 rpm and 550 rpm. For pressure 2.5 MPa the shaft frequencies were 110 rpm, 440 rpm and 550 rpm. At frequencies 220 rpm and 330 rpm the electrical motor was overheating due to non-optimal operational points and high compression forces required. The objective of each test was to reach the condition in the thermal chamber, in which the change of gas temperature was less than 0.5 degree/minute. Although this condition does not represent an ideal steady state, it helped to reduce the experiment time and find a state where temperature does not change significantly over time. This temperature represented a quasi-steady state. Preliminary tests demonstrated that after the quasi-steady state is reached, the further temperature change was less than 5 K, an acceptable discrepancy in experimental results.

The quasi-steady state temperature $T_{g}^{st.s}$ was measured using two thermocouples. One was installed in the upper part of the thermal chamber and the other in the lower part (Fig. 8, Ch. I, T3 and T4). The resulting value was the average between two measurements. The temperature of the Stirling refrigerator T_{R} was also measured with two thermocouples (Fig. 8, Ch. I, T1 and T2) at the cold heat exchanger (Fig. 1a) and the resulting temperature was an arithmetic average.

III RESULTS

III.1 PRELIMINARY TESTS

The gas that moved around the bank of tubes had the average velocity of 1 $m \cdot s^{-1}$. Around the walls gas had the average velocity of 0.7 $m \cdot s^{-1}$. Table 3 shows measurement results for the three preliminary tests and calculated thermal resistance between the gas and the refrigerator, cooling capacity and the Otaka number.

Table 3. Measurement and calculation results for three preliminary tests.

Test number	p, MPa measured	f, Hz measured	T_{r}, K measured	$T_{g}^{st.s}$, K measured	T_{R}, K measured	$R_{g\text{-}R}$, $K \cdot W^{-1}$ calculated	q_{R}, W calculated	Θ calculated
1	2.5	8.9	289	168	123	0.795	56.6	0.059
2	2.5	8.9	287	169	129	0.797	50.2	0.052
3	2.5	8.9	287	162	128	0.785	43.3	0.048

Although the maximum variations of values for $T_g^{st.s}$ and T_R is 7 K or 5%, the resulting maximum variation in cooling capacity is significant, 13.3W or 30%. Different measured temperatures are the result of different initial operating conditions. In Test 1 and 2, the SR was in the condition of thermalization with the atmosphere before the launch and it took 40 minutes before reaching the quasi-steady state in both tests. However, before Test 3, the SR had worked for 45 minutes and the system was in a precooled condition; therefore, the time required to reach the quasi-steady state was less than in Test 1 and 2, 15 min. All tests were conducted in different days, which also affected the temperature of the cooling water and the thermal state of the system. This differences in initial conditions influenced the temperature measurements. On the other hand, this variation better represents the operation in real environment and helped to calculate more realistic average Otaka number, which from Table 3 was equal to 0.053.

III.2 EXPERIMENTAL MEASUREMENTS

Fig. 2 shows how the temperature of gas in the chamber depends from the temperature of the Stirling refrigerator.

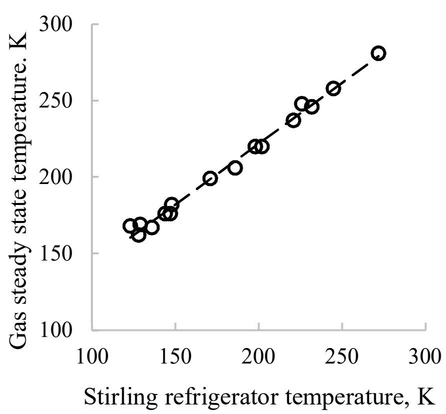


Fig. 2 Relation between gas and refrigerator temperatures measured in the experiment.

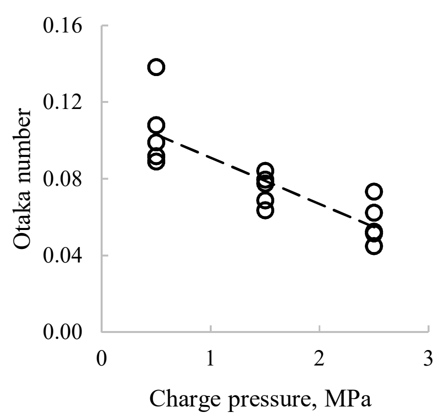


Fig. 3. Otaka number from the charge pressure.

The same calculation approach was applied as in Table 3 to find Θ for all the rest measured points and plotted it against the charge pressure (Fig. 3). I chose the charge pressure as the independent variable for the Otaka number because it relates to the input energy, which is necessary to run the refrigerator, and should be connected with its performance, characterized by Θ. Fig. 3 shows an interesting result: the Otaka number is not constant as it was believed previously in [1] and originally assumed in this work. The results indicate that with higher charge pressure the performance of the Stirling refrigerator, which is characterized in equation (1) by Θ, declines. Higher charge pressure allows obtaining lower refrigeration temperatures, and at lower temperatures, the performance of the SR typically declines. This can be seen in Fig. 4 with the dependence of the specific Otaka number Θ_s from the temperature ratio. The results from Fig. 3 were used to correct the model input Otaka number with values defined by the linear fitting. A more accurate study of Θ as a function of Stirling refrigerator control parameters is a subject of future work.

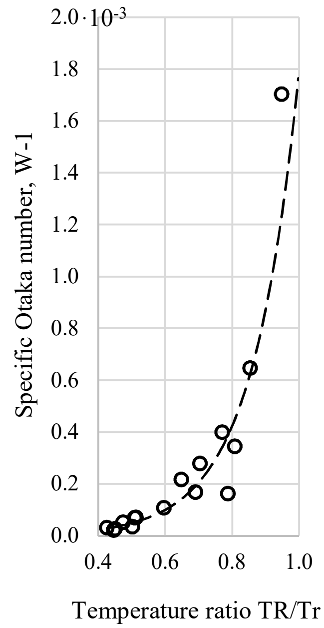


Fig. 4. Specific Otaka number from temperature ratio.

III.3 Model accuracy

Fig. 5 shows the comparison between experimental and calculated temperatures $T_g^{st.s}$ and T_R with the assumption of a constant Θ. For this results, the coefficients of variation are 12% and 16%, and the coefficients of determination are 0.6 and 0.62 for $T_g^{st.s}$ and T_R correspondingly. It is also evident that the model systematically and in some instances quite significantly undervalued the objective temperatures. This is also confirmed by the corresponding mean bias errors of −10% and −14%. The assumption about constant Θ leads to inacceptable accuracy of the prediction results. The model showed good accuracy for the measurement points, for which we estimated the Otaka numbers in Table 3. Other than this, the model calculated temperatures systematically lower than the measured values, on average 10% and 14% lower for $T_g^{st.s}$ and T_R correspondingly. Since the only unknown input parameter in the model was the Otaka number, these observations indicate that the calculated $\Theta = 0.053$ is lower than the actual value, except for the measured points in Table 3. If the number was higher, the systematic negative error would decrease.

For results in Fig. 4, the Otaka number was a variable and corrected using linear fitting line in Fig. 3. The coefficients of variation are 4% and 3%, the mean bias errors of −2% and -2%, and the coefficients of determination are 0.96 and 0.99 for $T_g^{st.s}$ and T_R correspondingly. These values represent a significant and satisfactory improvement of the model accuracy.

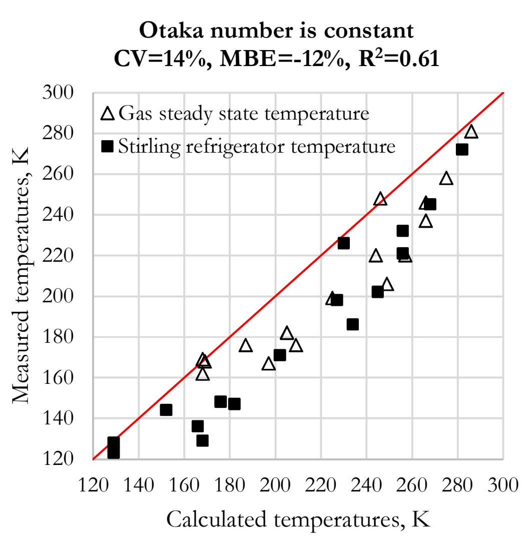


Fig. 5. Measured and calculated temperatures in the thermal chamber with a constant Otaka number.

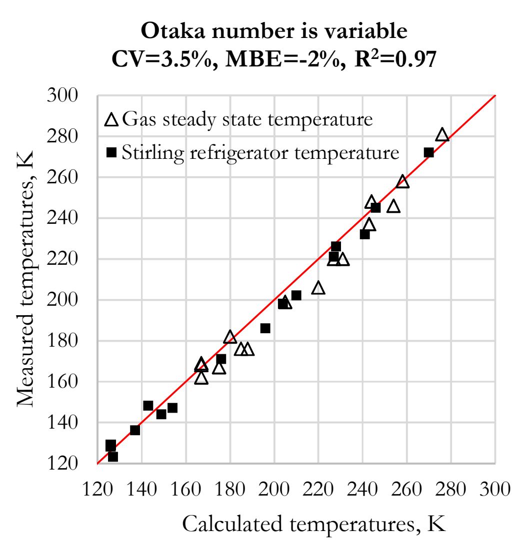


Fig. 6. Measured and calculated temperatures in the thermal chamber, with a variable Otaka number.

IV DISCUSSION

IV.1 LIMITATIONS

Although, the results do offer a numerical model validated at different operational parameters of the SR, this modeling approach has several limitations. The validation data was obtained for the Stirling refrigerator and the thermal chamber with fixed designs and for a specific range of operational parameters. Future research work could apply this methodology and conduct similar tests for different SRs and chambers with various operational parameters to extend the applicability of the proposed model. The thermo-electrical analogy assumes the equilibrium conditions in the chamber, but during the real operation of the Stirling refrigerator within a finite time the system is always in the non-equilibrium condition, and the notion of thermodynamic equilibrium in the thermal chamber is only acceptable with some reservations. In addition, some applications require the prediction of the cooling time, and the current model does not provide yet the solution to this problem. For future work, an alternative approach to model heat flows and temperatures in the studied system for non-equilibrium settings is the Onsager formalism [3]. The thermal model at this stage does not account for the presence of the cooled objects in the chamber, yet this problem is relevant in order to accurately predict the steady state condition and the change of temperature in the chamber over time. In addition, the expressions for the resistance between the atmosphere and gas assume an infinite flat wall. This is not the case for the thermal chamber with a finite wall of a specific geometrical form. To improve the accuracy of the model, the expressions that account for the geometry of heat exchanger surfaces could be introduced in future.

A more careful study of the Otaka number is necessary. At this point, the relation of the Otaka number to the operational and design characteristics of the refrigerator at different temperatures limits the applicability of the proposed model for system scaling. Nevertheless, it is possible to experimentally obtain some dependence of Otaka number from operational parameters and apply the model based on this number for scaling with accurate predictions. Such an application was demonstrated by using the results in Fig. 3 to obtain predictions in Fig. 6 with sufficient accuracy. In other words, although the Otaka number is not a constant parameter as previously expected, it is still possible to obtain accurate performance predictions (Fig. 6) and scaling using its dependence from the charge pressure (Fig. 3). Therefore, the model in Eq. 1 can be applied for system scaling, however, with some limitations discussed above.

Literature offers other approaches to model the cooling capacity of the Stirling refrigerator, which are problematic to apply. One method is to calculate cooling capacity using the equation for the ideal isothermal expansion derived from the first principles: $q_{\mathrm{R}} = f\, m_{\mathrm{He}} R_{\mathrm{He}} T_{\mathrm{R}} \ln\left(V_4/V_3\right)$, where f is the cycle frequency, m_{He} mass of working gas (helium), R_{He} is the specific gas constant for helium, V_3 and V_4 – the volumes of gas before and after the expansion. Zhu *et al.* [4] apply this method with some modifications to the volume parameters. The advantage of this equation is its relative simplicity and reliability. Similar to Eq. 1, it includes shaft frequency and refrigerator temperature. However, the application of this equation may be problematic because of several reasons. It does not include the refrigerator charge pressure, and its relation to available parameters is not straightforward; the equation is for the ideal isothermal expansion; and real values of m_{He} and gas volumes before and after the expansion are not known. The introduction of an experimental correction parameter can solve this problem. The parameter would account for the differences between ideal and real expansion, but this approach would be similar to the Otaka number. Several studies [5] assume a polytrophic process to model the Stirling refrigerator for more realistic analysis. Although this parametric approach can be informative for numerical studies, obtaining polytropic exponents for a real refrigerator would also require experimental measurements. Hachem *et al.* [6] propose a method to calculate cooling capacity using time-dependent differential equations.

This method requires complex numerical modeling. Le'an *et al.* [7] offer an approach to model cooling capacity, which includes shaft frequency, average expansion pressure, and design parametric relations. This method is the closest to the one offered by Otaka *et al.* However, the parametric equation is more complex and requires knowledge of the average expansion pressure. Another approach to estimate cooling capacity is to employ the logarithmic mean temperature difference [8]. This method is suitable for a numerical optimization but does not offer the relation between cooling capacity and operational parameters of the refrigerator. In addition, the accuracy of the discussed methods was sparsely studied using experimental data, unlike to the model in Eq. 1. It is difficult to obtain a model that would accurately predict cooling capacity without introducing an experimental correction parameter. It might be possible to reduce some uncertainty imposed by real system operation, but the complexity of such modeling is likely to increase. Besides, providing a model that includes shaft frequency, charge pressure, and known design parameters is likely to include experimental correction variables. Constrained by these conditions, the model offered by Otaka *et al.* is a compromise between complexity and accuracy.

IV.2 Specific Otaka number

An interesting relation between the specific Otaka number and the temperature ratio was found in the experiment. The specific Otaka number declines exponentially with reducing temperature of the refrigerator. This exponential reduction may be explained by the reduced performance of the regenerator. The same specific Otaka number $\Theta_{\mathrm{s}} = \Theta / \left(S_{\mathrm{R}} \cdot f \cdot p \right)$ for a given refrigeration temperature (Fig. 4) defines Θ for different configurations and operational characteristics of the Stirling refrigerator as long as the product in the denominator of Θ_{s} gives the same value. These configurations are defined by parameters S_{R}, p and f.

IV.3 Finite time operation of the Stirling refrigerator

Section IV.1 discussed challenges in adapting analytical modeling to predict the cooling capacity of the Stirling refrigerator. The main problem is to find an accurate model that also provides the relation between the operational parameters, such as shaft frequency and charge pressure, and cooling capacity. A solution adopted in this chapter is the employment of a model with an experimental coefficient (Otaka number). Another approach could be the increase of accuracy for models that are based on the first principles of classical thermodynamics. This accuracy improvement can be achieved by implementing the methods of finite-time thermodynamics (FTT).

FTT asks an interesting question that classical thermodynamics does not answer, specifically how constraints on time and operation rates affect a system's performance. Andresen *et al.* [9] provide a good demonstration of how these constraints can affect the efficiency of ECT. With some adaptation, I describe the essence of this example here. A watermill would be grinding the grain slowly if the speed of the water stream was low. In this case, the efficiency of the watermill would be relatively high. The friction losses would be lower due to low rotational speed. The conversion efficiency of the watermill blades would be higher. The slow and homogeneous flow has more water parts that produce useful work and have optimal inflow angles that minimize the number of flow vortices. When the watermill speed is lower, the irreversible losses will be lower as well, and the efficiency will be higher. On the other hand, it is desirable to have a watermill that can grind the grain in a shorter time. However, with an increased watermill speed, one should expect higher irreversible losses and lower watermill efficiency. The point of this example is that when shorter time or higher rates (e.g., rates of grinded grain) are required, the irreversible losses during the conversion process increase and the efficiency declines.

It would be interesting to analyze whether the constraints from time and rates affect the performance of the Stirling refrigerator. The design of efficient Stirling refrigerators with minimized irreversible energy losses is an essential objective because high efficiency is an indicator of design perfection and lower electricity bills. On the other hand, it is desirable to achieve lower refrigeration temperatures or higher cooling capacities for different small temperature applications. For the tested system, two strategies were available in the framework of finite time operation: the increase of refrigerator shaft speed and the increase of charge pressure. Higher shaft speed means more heat pumping cycles are performed in a given time. Higher charge pressure means more helium molecules could transport heat from the heating load, which increases the rate of pumping heat. In Fig. 9, Chapter I, it is evident that with the increase of charge pressure for a given shaft speed, the refrigeration temperature and cooling capacity increase. For example, at 550 rpm, at 0.7 MPa the cooling capacity is 49W at −111°C and at 2.5 MPa the cooling capacity is 75W at −165°C. However, the highest efficiency (for a refrigerator, it is the coefficient of performance) is reached at lower pressure, 0.7 MPa. This finding serves for Stirling refrigerators as an experimental confirmation for the FTT idea that at higher performance rates, the irreversible losses are higher, and the efficiency is lower. Fig. 7 shows experimental data for the cooling capacity with changing shaft frequency. This data can be useful for the evaluation of the first strategy to increase cooling capacity. To confirm a negative effect on performance from a shorter cycle time, the trend lines should have a maximum and at some shaft frequency begin to decline after this maximum. This effect is not observed in Fig. 7, most likely because the shaft frequency was not high enough. The maximum shaft frequency of 550 rpm is not a typical limit for Stirling refrigerators. Values around 1000 rpm are more common. The designed refrigerator can operate at higher frequencies but has not been yet tested. Nevertheless, there is an expectation that the operation at higher frequencies would cause an increase in friction losses, creating additional irreversible losses and reducing

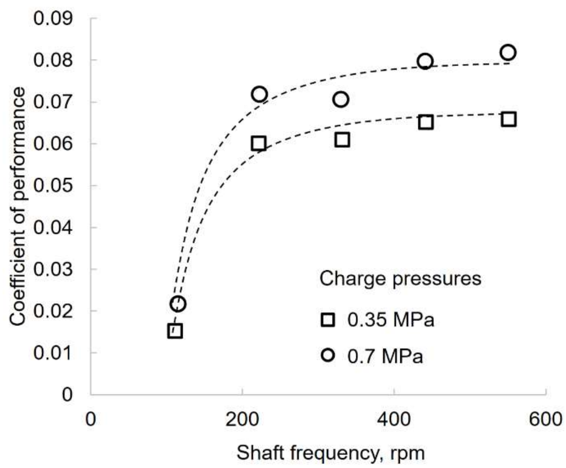


Fig. 7. Coefficient of performance as a function from shaft frequency and charge pressures.

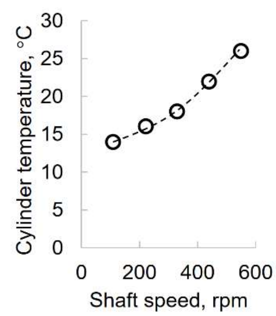


Fig. 8. Measured cylinder temperature with dismounted cold head and heat exchangers.

the COP value. Fig. 8 confirms this expectation and depicts the increase of cylinder temperature from friction losses at increasing frequencies. The cold head and heat exchangers were removed from the refrigerator, like shown in Fig. 6, Chapter I, to exclude the heating effect from gas compression. From Fig. 8, one may conclude that higher frequencies would cause higher irreversibility losses, and the coefficient of performance for the refrigerator is likely to reduce after 600 rpm. This reduction would also confirm the negative impact of shorter times on refrigerator efficiency. Overall, it is clear that the effect of shorter cycle times and higher cooling rates negatively affect the performance of the tested Stirling refrigerator. This effect could be integrated into the modeling equations for cooling capacity in the future.

Finite-time modeling of the Stirling refrigerator was previously studied in the literature [8]. The methods of this modeling approach were not used in the physical model for the studied Stirling refrigerator and can be addressed in future work. Some additional work would be necessary to conduct analytical integration between the primary heat transfer equations and operational parameters with experimental validation of results. Apart from numerical modeling, some elements can be optimized experimentally with the consideration of finite time operation.

The main design question when considering FTT is how to optimize the elements of the refrigerator and the whole system to yield a maximum heat lift per cycle? One example of the source for such optimization is the time-path of the pistons. The modern design was developed for the sinusoidal motion. The literature indicates that if the piston movement was optimized for a maximum cooling power per cycle, a 15% increase in efficiency could be achieved [9]. Other sources for minimization of irreversible losses include further optimization of the PCP pair, the performance improvement of regenerator, and the reduction of thermal resistance for coupling pairs "refrigerator - heat sink" and "refrigerator - heat load". The optimization of PCP was discussed in the previous chapter; the regenerator and the coupling between refrigerator and heat sink were not studied in this work and could be a topic of future investigations. The coupling between the refrigerator and the heat load was modeled and studied experimentally; therefore, several comments about the degrees of freedom to optimize the coupling are made below.

The thermal resistance $R_{\text{g-R}}$ of the coupling between the refrigerator and heating load (Fig. 1b) is the source of irreversible losses during the system operation. If the resistance $R_{\text{g-R}}$ was lower, a smaller value of $T_{\text{g}}^{\text{st.s}}$ could be reached with the same energy input. The physical model developed in this chapter helps to evaluate what technical aspects may help to reduce thermal resistance $R_{\text{g-R}}$. One option is to increase the speed of circulating air inside the thermal chamber, which would intensify forced convection between the gas and the surface of the cold head. Another alternative is the arrangement optimization of the cold head pipes. Different arrangements result in various heat transfer conditions and heat transfer coefficients. Also, the surface of the cold head pipes can be modified: more pipes with smaller diameter could be used or the structure of the surface can be adjusted with fins or cavities. The design of the thermal chamber significantly affects the coupling with the heat load. Hermetic sealing of the chamber from the environment would minimize parasite heat and air leaks. In the tested system in Fig.

1, complete sealing of the chamber was absent. As a result, warm room air was sucked in the chamber. Air moister and CO2 condensed on the surfaces of cold head pipes. This effect increased the resistance of the coupling. The intermediary environment between heat load and refrigerator is another source of reducing the resistance. A heat load could be cooled through the direct connection to the refrigerator. In this case, a primary mechanism of heat transfer is conduction. An alternative approach could be the introduction of an intermediary coolant loop. This approach may seem redundant and expensive at first. However, considering a complex shape of the cold head surface, the direct connection of the heat load may not always result in low thermal resistance. In this case, a second coolant loop can be considered with optimized heat exchangers. The analysis of discussed measures to reduce the coupling resistance can be the subject of future work.

IV.4 Design of energy conversion technologies

Energy conversion technology has the relation between operational parameters of the system and the performance characteristics. For the Stirling refrigerator, Eq. 8 describes this dependence. Also, there is the interaction of the system with the external environment, which can be modeled using, for example, thermal-electrical analogy, as for the case of the Stirling refrigerator, Eq. 3 and 4. The integration of the models for the external and internal interactions, Eq. 8, offers a relatively simple, yet powerful tool to scale the ECT. This model needs to be verified by experimental tests to validate assumption and increase accuracy. The example of the Otaka number for the Stirling refrigerator, previously assumed to be a constant value, shows the importance of the experimental testing and validation of the physical model. This design approach is a useful design tool to scale a selected ECT for desired applications.

IV.5 Novelty and contribution

The present work develops and experimentally validates a numerical model for the system “Stirling refrigerator – thermal chamber” to predict temperatures in the thermal chamber and refrigerator’s cooling capacity. The work also corrected previously believed assumption that the Otaka number is a constant parameter. In addition, the results showed an interesting relation between the specific Otaka number and the refrigeration temperature. The proposed model can now be applied to study the effect of different design configurations of the analyzed refrigeration system, and the main principles of the method can be used in scaling other ECT. The novelty of results can be summarized in the following points:

- Developed and validated experimentally a physical model of the system “Stirling-refrigerator – thermal chamber” by unifying the thermo-electrical analogy model of the thermal chamber and the model for the operation of the Stirling refrigerator.
- It was found that the Otaka number, previously believed to be a constant parameter, is dependent from the temperature of the refrigerator.
- Demonstrated simple and effective approach to scale ECT using experimentally validated physical model.

V Conclusions

- Developed a numerical model for the system “Stirling refrigerator – thermal chamber” based on the electro-thermal analogy to predict the steady state Stirling refrigerator temperature and gas temperature in the chamber.
- Conducted a series of 15 tests of the real Stirling refrigerator at mean pressures ranging from 0.5 MPa to 2.5 MPa and shaft frequencies ranging from 110 rpm to 550 rpm. Collected experimental data for the range of refrigerator temperatures from 272 K to

128 K and cooling capacities from 10 W to 53 W. The data was used to validate the model.

- Disproved the previously believed assumption that the Otaka number is a constant value. Using experimental results, the Otaka number was proven to be a variable parameter dependent on the refrigeration temperature.
- The coefficient of variance for the results of the validated model compared with the experimental results for the temperature of circulating air in the chamber is 4% for the refrigerator temperature is 3%. The mean bias error is –2% and –2% correspondingly, the coefficient of determination is 0.96 and 0.99 correspondingly for the gas and the refrigerator temperatures.

REFERENCES

[1] Otaka, T., Ota, M., Murakami, K. and Sakamoto, M., 2002. Study of performance characteristics of a small Stirling refrigerator. *Heat Transfer—Asian Research: Co-sponsored by the Society of Chemical Engineers of Japan and the Heat Transfer Division of ASME, 31(5)*, pp.344-361.

[2] V. Gnielinski, A. Zukauskas, and A. Skrinska. Heat exchanger design handbook, 1983. Fluid mechanics and heat transfer. Single-phase convective heat transfer. Banks of plain and finned tubes, Vol. 2, p.460. Hemisphere Publishing Corporation.

[3] Blundell, S.J. and Blundell, K.M., 2009. Concepts in thermal physics. Chapter 34, Non-equilibrium Thermodynamics, p. 408. Second edition. Oxford.

[4] Zhu, S., Yu, G., Jongmin, O., Xu, T., Wu, Z., Dai, W. and Luo, E., 2018. Modeling and experimental investigation of a free-piston Stirling engine-based micro-combined heat and power system. *Applied energy, 226*, pp.522-533.

[5] Erbay, L.B., Ozturk, M.M. and Doğan, B., 2017. Overall performance of the duplex Stirling refrigerator. *Energy conversion and management, 133*, pp.196-203.

[6] Hachem, H., Gheith, R., Aloui, F. and Nasrallah, S.B., 2017. Optimization of an air-filled Beta type Stirling refrigerator. *International Journal of Refrigeration, 76*, pp.296-312.

[7] Le'an, S., Yuanyang, Z., Liansheng, L. and Pengcheng, S., 2009. Performance of a prototype Stirling domestic refrigerator. *Applied Thermal Engineering, 29(2-3)*, pp.210-215.

[8] Ahmadi, M.H., Ahmadi, M.A., Mohammadi, A.H., Feidt, M. and Pourkiaei, S.M., 2014. Multi-objective optimization of an irreversible Stirling cryogenic refrigerator cycle. *Energy Conversion and Management, 82*, pp.351-360.

[9] Andresen, B., Salamon, P. and Berry, R.S., 1984. Thermodynamics in finite fille. *Physics today*, p.63.

CHAPTER VII
CONCLUDING DISCUSSION

This work has formulated different design methods for the commercial development of Stirling machines and combined them in an overarching design methodology. The methods for selecting alternative design decisions at different design stages were validated using the example of the development of a commercial Stirling machine. The objective of this chapter is to discuss the design methods further and analyze their applicability to the development of other energy conversion technologies (ECT).

I INTRODUCTION

I.1 BIBLIOMETRIC METHOD TO STUDY PAPERS AND PATENTS AT THE CONCEPT STAGE

The conceptual design stage always includes a literature analysis. The bibliometric study of Stirling engines (Chapter II) and refrigerators (Chapter IV) demonstrated a wide variety of design decisions embedded in Stirling machines from technical literature. It is interesting to note that design configurations from scientific and patent literature very often are different from each other. Depending on the geographical region, the time of the development, and the commercial or scientific focus of organizations, the choice of design configurations were not always alike. One should be aware of these factors when making design choices.

I.2 TRADESPACE EXPLORATION METHOD AT THE SYSTEM DESIGN STAGE

Trade studies may significantly influence how we make the design decisions (Chapter II, Section IV.9). Their results can clearly show the advantages and disadvantages of different machine configurations depending on the intended commercial application. This factor is essential because literature showed how many organizations with commercial on-Earth focus adapted the machine architectures that were originally developed for space applications. One argument for such line of action may be the availability of public knowledge. However, there are consequences of adapting configurations for non-intended applications, for example, limited scalability of space-intended free-piston systems.

I.3 APPLICATION OF GAME THEORY AT THE DETAIL DESIGN STAGE

A critical aspect of Stirling machine development is the influence of disciplinary designers and other stakeholders on the final design outcome (Chapter III). This influence was analyzed in detail using game theory. Engineering studies rarely focus on explaining how the organization of design teams and the design behavior of project participants would affect the design outcomes. This is a significant factor in the development process since commercial design projects typically constrained in time and budget.

I.4 EXPERIMENTAL OPTIMIZATION OF CRITICAL COMPONENTS AT THE TESTING STAGE

After selecting the design and managing the design process to the point where there is an experimental operating system, the next critical step is to prioritize performance characteristics and focus on improving the most important system elements (Chapter V). The piston-cylinder sealing is arguably the most vital element in Stirling machines. This work showed how gradual experimental improvement of this element allowed reaching satisfactory operational performance.

I.5 EXPERIMENT-BASED MODEL FOR SCALING AT THE SCALE-UP STAGE

The commercial development requires the scaling and testing of the developed experimental system for different customer requirements (Chapter VI). To do so most effectively, the development of a numerical model, which is validated by experimental results, is essential for evaluation of different commercial applications. The understanding of factors in section I.1-I.5 and implementing proposed design methods help to organize an effective research and development process of Stirling machines for commercial usage.

I.6 Analysis of the Thesis Statement

The results of each chapter showed that indeed the process of selection between alternative design decisions at different design stages during the commercial development of Stirling machines requires a unique set of design methods, which are crafted, with the consideration of the commercial nature of development. These unique design methods form a pathway through consequent design stages of the commercial development process, Fig. 1.

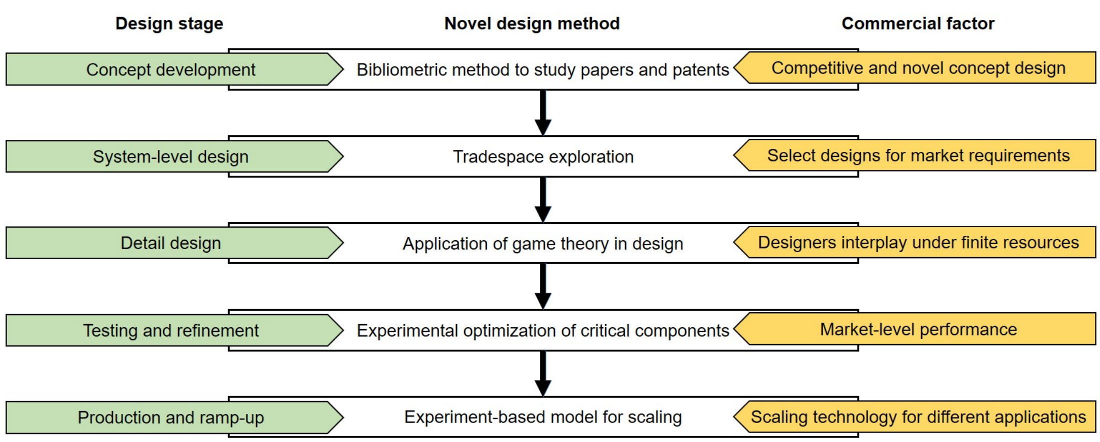


Fig. 1. The visual representation of the technological pathway to develop commercial Stirling machines

The chapters of this thesis informed the thesis statement for each development stage, and the results suggested that it could be possible to apply the same technological pathway to the development process of other energy conversion technologies. For the concluding discussion in this chapter, it would be interesting to evaluate further the commercial factors that inspired the proposed design methods. I will discuss the advantages and disadvantages of the methods and extend the discussion on their applicability to the development of other energy conversion technologies.

II Methodology

II.1 Commercialization factors in development

The analysis of commercialization factors that influenced the definition of proposed design methods will be assessed qualitatively using the framework of Barge-Gil and López [1]. The authors analyzed typical characteristics of commercial development projects in comparison to scientific projects for several categories, such as purpose, type of knowledge, and people management. Some of the factors are further validated by the results of the bibliometric analysis in Chapters II and IV.

II.2 Advantages and limitations of design methods

The analysis of advantages and limitations will be performed using a comparison table, where for each design method proposed in the thesis, I will list essential advantages and limitations. This qualitative comparison is a simple yet informative approach to assess critically the formulated design techniques.

II.3 Applicability of the technological pathway to other ECT

To discuss whether the proposed in this thesis overarching design methodology can be applied to the development of other energy conversion technologies, I use the table with a list of factors that justify and limit the application of methods to other ECTs. This approach is simple and instructive.

III Results

III.1 Commercialization factors in development

According to Fig. 1, several vital factors of commercialization influence the development of ECT at each design stage. The requirement for the competitive and novel design at the conceptual stage. The fulfilment of market requirements by the system-level design at the system design stage. The complex interplay of disciplinary designers under limited time and resources at the detail design stage. The requirement to achieve market-level performance at the test and refinement stage. And the need to scale technology at the production and scale-up stage.

The framework of Barge-Gil and López suggests that the purpose of the commercial development project is "introduction of new or improved process or product." This statement confirms the need for novel design. The concept of product also inherently assumes the competitiveness of the developed design. The other aspect of the development project is the "link with customers," which confirms the need to fulfill market requirements. The claim of limited time and resources during the detailed development is confirmed by the characteristic of "short term" with "pressure to market [that] usually constrains [the project] between six months and two years." Furthermore, the expectation of reaching the market-level performance of technology can be implied from the orientation towards customer and development of a new product. Results of bibliometric analysis in the patent field, Chapter IV, confirm the need for scaling technology for different applications of Stirling refrigerators: commercial organizations developed inventions for air-conditioning in cars and trucks, food processing, gas liquefaction, and container storage. In summary, the comparison of assumed in the thesis commercial factors with the constraints of real science-based product developments (Barge-Gil and López) confirms that assumed commercial factors are plausible. For future work, more references to the analysis of real product developments can be established for further confirmation of assumed commercial factors.

III.2 ADVANTAGES AND LIMITATIONS OF DESIGN METHODS

Table 1 lists the proposed design methods and enumerates their advantages and limitations for the ECT development.

Table 1. Advantages and limitations of proposed design methods.

Design method	Advantages	Limitations
Bibliometric method to study papers and patents using big data approach	• Results by leading organizations with consistent research record; • Representative analysis of all existing knowledge; • Understanding of historic factors; • Statistically significant conclusions.	• Expensive access to databases; • Analysis at the level of abstracts; • Possible to miss documents with significant contributions.
Tradespace exploration	• Analysis of large number of design alternatives; • Simple visualization and selection.	• Complex system model; • Accuracy of the system model may not be enough; • System metrics could be difficult to model mathematically.
Application of game theory in design optimization	• The only known methodology to account for the decision-making of decentralized disciplinary designers; • Understanding how the reallocation of design authority among designers affect the design outcome; • Understanding how designers affect the design outcome comparing to Pareto-optimal designs.	• Modelling a worst-case scenario that may represent the design process inaccurately; • At this moment, only modeling the design process with two disciplinary designers; • Difficult to model complex behavior; • Difficult to validate results of modeling.
Experimental optimization of critical components	• Can improve the performance of elements that otherwise would be too inaccurate to optimize analytically or numerically; • Quick success.	• Complex design of real experimental system; • May be labor-intensive and require strong management input.
Experiment-based model for scaling	• Quick evaluation of performance for alternative market applications.	• The need for experimental data at different operational conditions; • Inaccuracy in prediction if applied for operational conditions beyond the range of experimental data.

Advantages and disadvantages in Table 1 could be further confirmed by the analysis of the corresponding design literature. This analysis is a subject of future work.

III.3 APPLICABILITY OF THE TECHNOLOGICAL PATHWAY TO OTHER ECT

Table 2 lists the factors that enable and limit the application of the proposed technological pathway to the development process of other energy conversion technologies.

Table 2. Factors that enable and limit the application of the technological pathway to other ECT.

Design method	Enable	Limit
Bibliometric method to study papers and patents using big data approach	Any ECT has related body of scientific and patent publications that can be analyzed using literature data bases.	Novel ECT may be poorly referenced in scientific and patent databases because of native language of publishing in countries of invention.
Tradespace exploration	The modeling of relations between design parameters and metrics obey universal laws of energy conversion.	Relating the cost function to the operation parameters may not be always possible due to scares data.
Application of game theory in design optimization	Game theoretic algorithms are universal for any ECT.	The connection between game theoretic algorithms and the system model of ECT may be problematic.
Experimental optimization of critical components	ECT at the testing and refinement is a material device that has critical operational components.	May be too expensive to build experimental system, e.g. fusion reactor.
Experiment-based model for scaling	Any ECT could be modelled and experimentally tested.	The stability of ECT operation at different operational conditions can be low and the range of obtained results too limited.

For future work, a more detailed study of applicability can be performed. Several energy conversion technologies could be selected as case studies and the discussed in Table 2 factors can be verified against each case study.

IV DISCUSSION

IV.1 RELATION TO OTHER SCHOOL OF DESIGN THINKING

The elements of the proposed technological pathway can be related to other schools of thought to better ground the results of this thesis in existing design literature. For systematization, I created Table 3 that lists the elements of the technological pathway and related schools of design thinking with references. The table starts with the concept of the technological pathway as such. It can be characterized by the focus on science-intensive energy technologies, the development of which can be attributed best to the technology push concept. I commented on the application of this concept in the introductory paragraph of Chapter I. The linearity of the technological pathway can be related to different representations of the life-cycle models, and the approach by which design activities are conducted during the development process can be analyzed using the Waterfall-Vee-Agile models. The Bibliometric

Table 3. Elements of the novel technological pathway and related schools of design thinking.

Stages and names of novel design methods	**Literature**
Technological pathway	• Science-intensive technologies [2] • Technology push [3] • Life-cycle stages [4, p. 3.5] • Waterfall-Vee-Agile [5]
Concept *Bibliometric method to study papers and patents*	• Knowledge discovery from big data [6] • New product development using patents [7] • Brainstorming [8] • Life-cycle cost against time [4, p. 2.6]
System design *Tradespace exploration for Stirling machines*	• Tradespace exploration concept [9] • Requirements engineering [4, p. 7.6] • Visualization of ideas [10]
Detail design *Application of game theory in design*	• Bounded rationality [11] • Design process as a game [12] • Concurrent engineering [13]
Testing and refinement *Experimental optimization of critical components*	• Rapid prototyping [14] • Incremental and iterative development [4, p. 3.10] • Set-based design [15]
Scaling *Experiment-based model for scaling*	• Technology validation (for market applications) and scale up [16]

method to study papers and patents using big data technologies is related to the process of knowledge discovery from big data. The focus on patent literature can be related to studies about developing new products using patents. The objective of the proposed method was to maximize the number of alternative design concepts, which can be considered as a method in creative brainstorming. A strong focus on exhaustive analysis of technical literature helps to avoid making poor design decisions that would have high life-cycle costs at later design stages. The method of analyzing Stirling machines using tradespace exploration can be attributed to the field of tradespace exploration in systems engineering. The results also help to develop design requirements for later design stages and visualize the results of data-intensive trade studies. The application of game theory at the detail design stage helps to resolve the impact of human interaction on the design outcomes caused, for example, by bounded rationality of experts in real design settings. The method also relates to broader research literature that recommends considering a design process as a game. The design method that intends to improve designers' interaction during detail design could also be related to the field of concurrent engineering. The experimental optimization of critical components at the test and refinement stage relates to the field of rapid prototyping and incremental and iterative development. Experimental optimization with an open end for final application (engine or refrigerator) can be related to the concept of the set-based design. The selected approach for scaling the Stirling refrigerator using the experimentally-validate physical model relates to the field of technology validation and scale-up. By using referenced literature on design thinking, the proposed design methods could be further improved in future work.

IV.2 Pentagon model for the development of ECT

The results of this work helped to formulate a design process model for the commercial development of energy conversion technologies, which I named the Pentagon Model. This model provides a simple visual explanation of the formulated technological pathway in this thesis. The model is illustrated in Fig. 2.

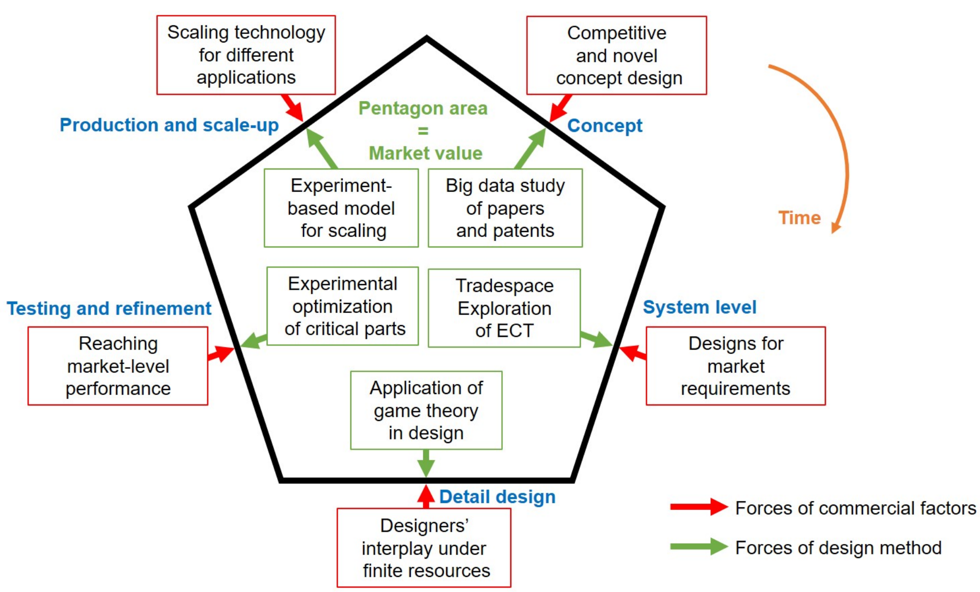


Fig. 2. Pentagon Model for commercial development of energy conversion technologies

The main idea behind the Pentagon Model is that during the development process, there are forces created by the commercial factors that influence the design process (red arrows). Each force acts at one of five design stages. The five stages form five sides of the pentagon. The area of the pentagon represents the market value of the system under development. The commercial forces act on the pentagon from the outside. There are also forces of design methods that counterbalance the commercial pressure from the inside of the pentagon. If there are no design methods that can counteract, the pentagon collapses, and there is no market value for the development project. The design methods proposed in this dissertation for each design stage maintain the integral structure of the pentagon. This simple representation helps to memorize the developed design methodology and apply it in future for the commercial development of other energy conversion technologies.

Two approaches for analyzing the interactions between different sides of the pentagon in Fig. 2 could be discussed. One approach is considering the interaction between stages as sequential. The other approach is assuming the concurrent design process between pentagon sides. The sequential interaction assumes that each previous stage pass on requirements and input design information to the following design stage. For example, the results of the concept study could include information collected using the big data bibliometric analysis about alternative design configurations of the ECT. For illustration, it could be alpha, beta, and gamma configurations of the Stirling refrigerator. These configurations have to be evaluated during the following system-level design stage using the tradespace exploration method to identify a preferable configuration, for example, an alpha-type, and optimal design characteristics – bore, stroke, refrigeration, and radiator temperatures. These system-level requirements are passed on to the following detail design stage. The detailed design should be conducted under the recommendations of the game-theoretic analysis to obtain the intended Pareto-optimal design. The developed design documentation and recommendations about most critical components – for example, the piston-cylinder sealing – are passed on to the testing and refinement stage for the production of the experimental system and to carry out experimental optimization. The results of experimental optimization provide data for the physical model of the ECT system for scaling. This examples show, how each stage of the sequential approach is connected – in the form of design requirements and information flows – with the previous and the following design stages. No cross-interaction between the pentagon sides was assumed. This sequential approach is well known in the systems engineering literature [4, p.3.4] and, more precisely, could be attributed to the Waterfall design process model [5]. The time arrow in Fig. 2 indicates that the design process follows sequentially around the pentagon from one side to the next.

The concurrent design process represents another approach to discuss the interactions between pentagon sides. The concurrent design process [13] assumes that a design expert represents each side of the pentagon. Several disciplinary designers could represent the detail design stage. The cross-stage feedback between the experts could result in the selection of design alternatives defined by constraints at different sides of the pentagon that would be different from those identified during the sequential design process. The concurrent design process inside the pentagon could be organized using different design process models, for example, the Agile model [5]. In this case, the timeline arrow would not be directed around the pentagon, as shown in Fig. 2, but would be facing upwards. This is because in each time moment of the design process there are interactions between experts inside the pentagon. With time dimension, the pentagon would form a three-dimensional object with changing shape, depending on the commercial and design forces acting on the pentagon sides.

V CONCLUSIONS

- This thesis developed a novel design methodology for the commercial development of energy-conversion technologies, which was tested on a real development process of the Stirling machine;
- A set of novel design methods was developed for each design stage from the concept design to the scale-up stage to help selecting alternative design decisions when developing a commercial energy conversion system;
- A visual model is proposed named “Pentagon Model” that combines main findings of this thesis and helps to organize the process of commercial development of any energy conversion technology.

REFERENCES

[1] Barge-Gil, A. and López, A., 2014. R versus D: estimating the differentiated effect of research and development on innovation results. *Industrial and Corporate Change*, *24*(1), pp.93-129.

[2] Meyer-Krahmer, F. and Schmoch, U., 1998. Science-based technologies: university–industry interactions in four fields. *Research policy*, *27*(8), pp.835-851.

[3] Herstatt, C. and Lettl, C., 2004. Management of" technology push" development projects. *International Journal of Technology Management*, *27*(2-3), pp.155-175.

[4] Haskins, C., Forsberg, K. and International Council on Systems Engineering, 2007. Systems engineering handbook:[SEH]; a guide for system life cycle processes and activities. *Incose.*

[5] Balaji, S. and Murugaiyan, M.S., 2012. Waterfall vs. V-Model vs. Agile: A comparative study on SDLC. *International Journal of Information Technology and Business Management*, 2(1), pp.26-30.

[6] Begoli, E. and Horey, J., 2012. Design principles for effective knowledge discovery from big data. *In 2012 Joint Working IEEE/IFIP Conference on Software Architecture and European Conference on Software Architecture*, pp. 215-218. IEEE.

[7] OuYang, K. and Weng, C.S., 2011. A new comprehensive patent analysis approach for new product design in mechanical engineering. *Technological Forecasting and Social Change*, 78(7), pp.1183-1199.

[8] Hargadon, A., 1996. Brainstorming Groups in Context: Effectiveness in a Product Design Firm Robert 1. Sutton. *Administrative science quarterly*, 41(4), pp.685-718.

[9] Ross, A.M. and Hastings, D.E., 2005. 11.4. 3 the tradespace exploration paradigm. *In INCOSE international Symposium*, Vol. 15, No. 1, pp. 1706-1718.

[10] Stump, Gary M., Mike Yukish, Timothy W. Simpson, and E. Nathan Harris., 2008 Design space visualization and its application to a design by shopping paradigm. *In ASME 2003 International Design Engineering Technical Conferences and Computers and Information in Engineering Conference*, pp. 795-804. American Society of Mechanical Engineers Digital Collection.

[11] Simon, H.A., 1990. Bounded rationality. In *Utility and probability*, pp. 15-18. Palgrave Macmillan, London.

[12] Vincent, T.L., 1983. Game theory as a design tool. *Journal of Mechanisms, Transmissions, and Automation in Design*, 105(2), pp.165-170.

[13] Prasad, B., 1996. Concurrent engineering fundamentals, Vol. 1. Upper Saddle River, NJ: Prentice Hall PTR.

[14] Chua, C.K., Leong, K.F. and Lim, C.S., 2010. Rapid prototyping: principles and applications. World Scientific Publishing Company.

[15] Ghosh, S. and Seering, W., 2014, August. Set-based thinking in the engineering design community and beyond. In *ASME 2014 International Design Engineering Technical Conferences and Computers and Information in Engineering Conference*, pp. V007T07A040-V007T07A040. American Society of Mechanical Engineers.

[16] Wieringa, R., 2014. Empirical research methods for technology validation: Scaling up to practice. *Journal of systems and software*, 95, pp.19-31.