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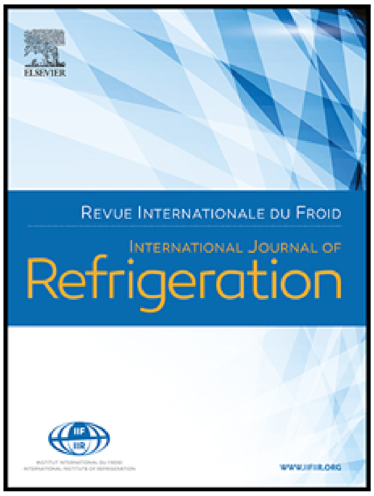

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Reversed regenerative Stirling cycle machine for refrigeration application: a Review

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Abstract

The Stirling cycle machines have many applications as prime mover and cooling purpose. A review is presented for the development of regenerative Stirling cycle machines in the refrigeration area. The Stirling cycle refrigerators of gas cycle machines, which are the counterparts of the Stirling engines. They can operate from low temperature to moderate temperature applications. The present paper describes an overview of the Stirling refrigerating machine and its associated researches carried out in this area.Initially, the review explains the general working principle, the configuration and the drive mechanisms as well as the research findings within the range from low-temperature cryocoolers to moderate temperature cooling applications. Furthermore, this review points out and discusses the various models and methods of optimization to improve the performance of the Stirling cycle refrigerator.

Keywords: Stirling refrigerator, moderate cooling, optimization, cooling capacity

Contents

1. Introduction

Refrigeration and Air-Conditioning systems are parts of day to day life in the world. Refrigeration is the process of removal of heat from an enclosed space so as to lower the temperature in the region. The purpose of any refrigerator is to extract heat as much as possible from a cold reservoir with an expenditure of as little power as it could be. This implies that the power requirement should be minimized for a given cooling load. Most of the refrigeration demand of the world is largely dependent on the vapor-compression refrigerator (VCR). VCR was first developed in the latter part of the 19th century based on the phase change process of Chlorofluorocarbon(CFC) or high Chlorofluorocarbon(HCFC) fluids. In a single stage, VCR produces a cold end temperature below 230 K, provided an appropriate refrigerant is used. The VCR currently exists technologically at the saturated level and is suited for household applications. Although, VCR is an efficient technology, especially for moderate temperature refrigeration applications, its environmental concern resulted in

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different scenarios. In the year 1987, a protocol was signed in U.K. with an objective to protect the ozone layer from further depletion by CFC emission. This protocol is called Montreal protocol and it states that CFC group of refrigerants, which is the main culprit for ozone layer depletion should be banned by the year 2010 [71]. The other related protocol signed in the year 1997 called Kyoto protocol, aimed to prevent global warming due to the use of HCFC/CFC refrigerants.

Recently, in addition to the effort to improve the working condition of VCR, two global challenges (global warming due to HCFC fluid consumption and continued increase in demand of refrigeration in almost all parts of the world) have caused the engineering community to search for alternatives to vapor-compression refrigeration. Inconsistent with this, it is emphasized that the utilization of environmentally safe fluids and improved energy efficiency leads to a number of new refrigerants that are under consideration and to the introduction of new technologies including Stirling cycle refrigerators [82]. In recent years, alternative cooling systems have gained significant importance. Stirling refrigerators for moderate temperature application is one of the technology that is brought from the concern about the use of CFC refrigerants and their effect on the environment. These machines theoretically have the highest efficiency possible for any practical thermodynamic system, and thus becomes a potential alternative to the existing vapor-compression refrigeration system.

Stirling cycle machine is a type of closed thermodynamic cycle machine invented in 1816 by Robert Stirling as a heat engine to convert thermal energy to mechanical energy. The air was used as a working fluid to replace the steam engine since they were prone to life-threatening explosions. The Stirling refrigerator, which is the counterpart of the Stirling engine was first recognized in 1832 [64]. The system was practically realized in 1862 when Alexander Kirk built and patented a closed cycle refrigerator based on Stirling cycle [60]. Until 1949, it was reported that very little development of Stirling refrigerators occurred, when the Philips Company in Holland ran a Stirling engine in the reverse direction by a motor and found that it liquefied air over the cold tip [63]. In the year 1971, Beale stated that Stirling cycle machines could be used for both work producing or refrigerating machines by reversing the cycle [15]. Therefore, Stirling cycle machine in this review is mostly meant Stirling cycle refrigeration machine.

Devices using the Stirling cycle are still expensive to construct due to limited production volume and the working condition (high temperature and high pressure especially in case of Stirling engine) that demands special materials and relatively high technology. Numerous researches have been conducted on Stirling machines especially about Stirling engine and Stirling cryocoolers for almost a century. The results reveal that the efficiency and performance improvements have been recorded and are still showing great improvement for Stirling engines and cryocoolers and wider applications are considered. Cooling by using Stirling cycle machine is rarely used at moderate/ambient temperatures for domestic or commercial refrigeration. On the other hand, for Stirling refrigerators with moderate temperature, where there is no need for high working temperature and pressure, the mass production cost can be reduced significantly as compared to a Stirling engine.

A major reason for the limited application of Stirling refrigerators is that vapor compression-type refrigerators which are already functioning in household refrigerators approached perfect conditions, and the necessity to develop other refrigeration systems is limited. For this reason, the number of researches conducted on the Stirling refrigeration for moderate temperature (near to ambient) applications are very few as compared to studies made about Stirling engine and Stirling cryocooler. However, currently increasing public and government concern at the impact of CFC/HCFC refrigerants on the ozone layer of the Earth has got attention to the search for alternatives to the present vapor-compression refrigerating systems. This social concern leads to unparalleled opportunities in the research and development on moderate temperature Stirling cycle refrigerating machine towards the application of household and commercial purpose.

The main aim of the present review is to briefly present Stirling cycle refrigerating machine. In the paper, the working principle of Stirling refrigerating machine, its configurations, driving mechanism of Stirling machine, low-temperature Stirling cycle coolers and moderate temperature Stirling refrigerating machine are reviewed. The review also tries to establish the main thermodynamic and parametric criteria that would be considered to design the refrigerating machine for different applications. Finally, research gaps were identified and conclusions are summarized.

2. Overview of Stirling refrigerating machine

A Stirling cycle refrigerating machine consists of two variable volumes (compression and expansion) spaces at different temperatures and physically separated by the regenerator. The presence of an economizer called the regenerator grouped Stirling cycle machine as a regenerative machine. The working gas is alternatively compressed and expanded by the power piston, while the displacer shuttles the gas back and forth between the cold end, where heat is absorbed, and the warm end, where heat is rejected. When the Stirling-cycle machine operates as a refrigerating machine, heat is lifted from the cold zone as shown in Fig. 1 during the expansion process. The work of expansion is lower than the work of compression. Therefore, the work input is needed in the refrigeration cycle to compensate for the compression work. The cooling effect is shown in the expansion space (see in Fig. 2).

An ideal Stirling refrigeration cycle consists of four separate thermodynamics processes which are explained as follows (Fig. 3):

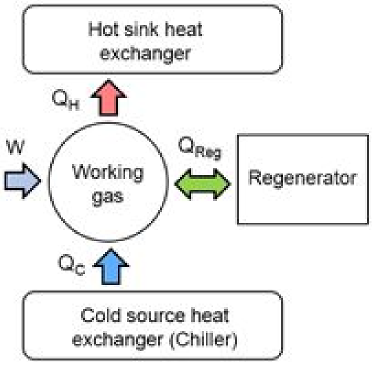

Fig. 1. Schematic diagram of a Stirling refrigerator

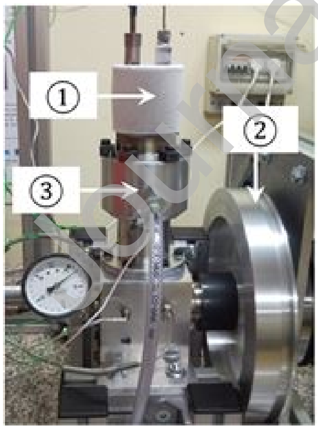

Fig. 2. Refrigeration Stirling machine (Beta type) at cooling stage with Nitrogen working gas (FEMTO-ST laboratory) where: 1. Cold source heat exchanger (chiller) 2. Flywheeel 3. Hot sink heat exchanger (water-cooling) [59]

1. *Isothermal compression process 1-2:* the working gas is compressed by the compression piston isothermally at the hot end, hence rejecting heat to the hot space (via the heat-exchanger called warm heat exchanger).
2. *Constant volume heat rejection process 2-3:* both pistons or a piston and displacer move together to transfer all the working gas isochorically through the Stirling regenerator to the cold end of the refrigerator (expansion space). Heat is rejected from the gas to the regenerator matrix as it passes through the regenerator, thus lowering the temperature of the gas to that of the cold space. Heat is stored in the regenerator matrix.
3. *Isothermal expansion process 3-4:* the low-pressure working gas expands isothermally at cold end temperature, hence absorbing heat from the cold space via the heat exchanger called chiller.
4. *Constant volume heat addition process 4-1:* both pistons or a piston and displacer move together to transfer all the working gas isochorically through the regenerator to the hot end of the Stirling refrigerator (compression space). Heat is restored to the gas as it passes through the regenerator from the regenerator matrix, thus raising the temperature as well as the pressure of the gas to that of the hot space and the cycle returns again to its initial position.

The surface delimited by the cycle represents the amount of net work input which must be provided to the refrigerating machine.

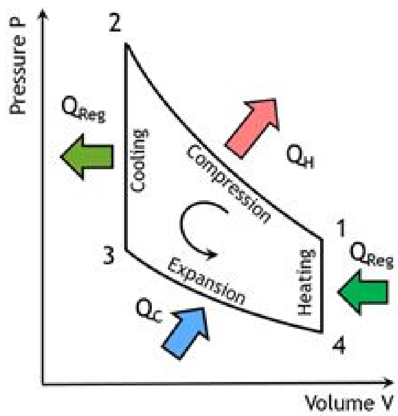

Fig. 3. P-V diagram of the ideal Stirling refrigeration cycle

The PV diagram of actual Stirling cooler is different from the ideal cycle (seen in Fig. 3). The working volume of most real Stirling machines follow sinusoidal variation based on the configuration and drive mechanisms. [75] designed, experimentally tested and showed the actual PV diagram of a 100 W capacity beta-type Stirling cycle cooler. The particular PV diagram for this Stirling cooler at a frequency of 13.3 Hz, cold head temperature of 233 K, and hot heat temperature of 303 K is shown in Fig. 4.

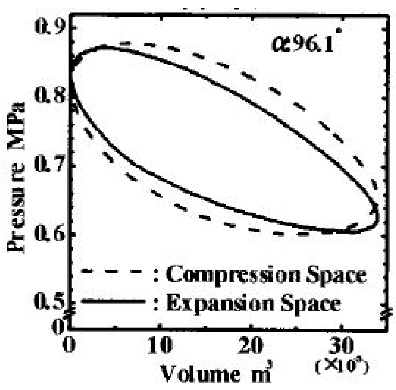

Fig. 4. P-V diagram of a particular real Stirling refrigeration cycle [75]

The motion of piston and piston/displacer with respect to time results in variation of the volume of the two working spaces(compression space and expansion space)as seen in the Fig. 5.

According to the classical theory of thermodynamics, the performance of a Stirling cycle machine is a function the following independent parameters [99, 85, 100]:

- Pressure of the working fluid,
- Speed of the machine,
- Phase angle between piston and displacer,
- Ratio of the temperatures (compression and expansion temperatures),
- Ratio of the swept volumes,
- Volume and efficiency of the regenerator and heat exchangers.

The parameters phase angle, ratio of temperature, ratio of swept volume and volume of the heat exchangers must be selected at the design stage of the Stirling cycle machine.

Heat is transferred in or out of the working fluid in the Stirling cycle refrigerators during all four phases of their operation, and their performance depends on the non-isothermal heat exchanges performed reversibly or irreversibly.

3. Configuration and driving mechanisms of Stirling cycle refrigerating machine

3.1. General

The systematic understanding of component configurations, common drive systems and their possible combination for Stirling cycle refrigeration is vital to develop appropriate selection criteria to use the machine for different applications. The Stirling cycle refrigerating machine can be classified based on their piston-cylinder configurations or based on the drive systems that ensure appropriate flow of working gas. Many different combinations of configuration and drives mechanisms are proposed for different applications [36]. Therefore, in this part of the review, we try to go through the various common arrangements of the Stirling cycle refrigeration systems and the associated researches done so far.

3.2. Configuration

Stirling machines have had a wide range of applications by varying their configuration. The mechanical configurations of Stirling cycle refrigeration machines are exactly similar to Stirling engine. Numerous designs of Stirling cycle machine exist but the following common elements shall be identified:

- A closed system containing a fixed mass of gas.
- Compression and expansion spaces with the volume of gas, which are controlled by a piston-cylinder arrangement. The enclosed working fluid flows in alternate directions between two interconnected gas spaces, at different temperatures, which is controlled by changing the succession volume.
- A displacement arrangement to shuttle the working fluid back and forth between the hot end and cold end gas spaces.
- A regenerator connecting the hot end and cold end gas spaces. The processes of heating and cooling are noticeably improved by the regenerator which generally consists of a matrix of fine wires or simply annular gaps made by winding foil.
- Hot heat exchanger and chiller for transferring heat to and from the cycle or they could act as heat reservoirs.

The basic parts of a Stirling cycle machine are the piston, the displacer, the cylinder volumes, the heat exchangers, and the crank mechanism [83]. The Stirling cycle machines are generally divided into three groups based on piston and piston/displacer-cylinder arrangement such as the Alpha, Beta, and Gamma configurations [62, 85, 99]. All configurations are working with the same thermodynamic cycle but with different mechanical design. The three configurations of the Stirling cycle machine could be described as:

1. *Alpha configuration :* two pistons in separate cylinders connected in series by a heater, regenerator and cooler [9, 65, 93, 67, 6]. The configuration of Alpha Stirling refrigerator is shown in Fig. 6. In this configuration, a displacer is not used. Both pistons need to transfer work and the piston-cylinder contact space is sealed to maintain the working gas at relatively higher pressure. The two pistons are arranged

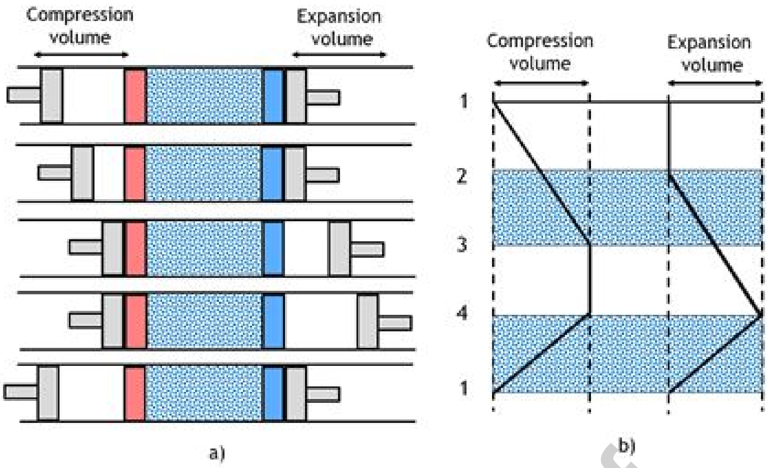

Fig. 5. (a) Full operation Stirling refrigerating cycle and (b) ideal time displacement diagram
1-2 : isothermal compression process 2-3 : constant volume heat rejection (regeneration) 3-4 : isothermal expansion process 4-1 : constant volume heat addition (regeneration)

to move uniformly in the same direction and provide constant-volume heating or cooling processes of the working fluid. Based on different drive mechanisms, Stirling machines with alpha configuration are found more suitable for medium and high-temperature difference applications [36]. Similarly, it is reported that Alpha type engine configurations are not suitable for low-temperature difference applications [30].

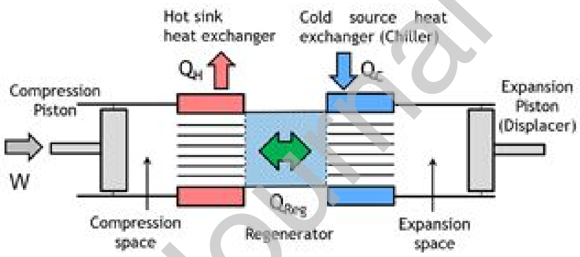

Fig. 6. Alpha configuration for Stirling refrigerator

2. *Beta configuration:* unlike the Alpha configuration, the Beta configuration Stirling machine has a single power piston and a displacer enclosed within a single cylinder [75, 47, 45]. A Beta configuration Stirling refrigerator is shown in Fig. 7. The main task of the displacer is to displace the working fluid and shuttle it between the compression space and the expansion space through the series of heat exchangers. From the patent drawing of 1816, it is shown that the Stirling machine first configuration was a Beta configuration. [36] reported that Beta configuration Stirling machines could be used conveniently from low to high-temperature difference applications according to different drive mechanisms with the crank drives more preferred from low to moderate temperature and rhombic drive mechanism more applicable from moderate to the high-temperature difference.

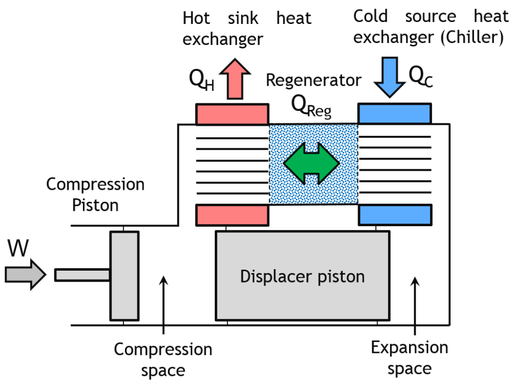

Fig. 7. Beta configuration for Stirling refrigerator

3. *Gamma configuration:* Gamma Stirling cycle machines have the same piston and displacer configuration [12] as Beta type Stirling machines. Unlike the Beta configuration, the power piston and the displacer are arranged in separate cylinders. A particular arrangement of Gamma type Stirling refrigerator is shown in Fig. 8. It is reported that only Gamma configurations are suitably applicable for low and very low-temperature difference applications [30, 36].

3.3. Driving mechanisms

To ensure the appropriate flow of working gas for the complete Stirling cycle, different types of drive systems are identified. According to the different configurations of

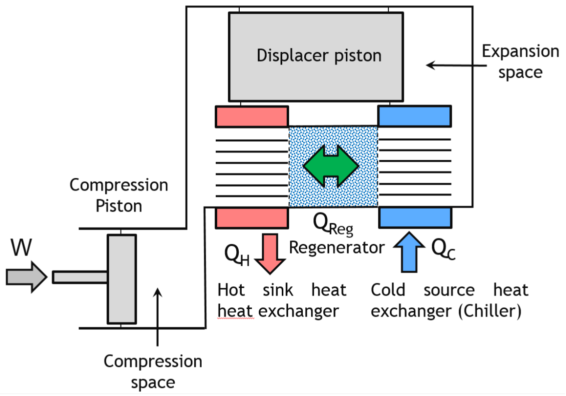

Fig. 8. Gamma configuration for Stirling refrigerator

the machine and to facilitate movement of the gas in the system, Stirling refrigerating machines are classified into kinetic, free-piston, pulse tube/thermoacoustic and liquid piston types. Among these drive systems, the first three types are the most commonly used at the different stages of the refrigeration system and discussed in the paper.

1. *Kinetic driven mechanism:* in kinetic drive mechanisms, the mechanical piston and piston/displacer are driven by kinetic drive mechanisms such as the simple crank-slider, Ross-yoke, rhombic drives, swash-plate and others as shown in the Fig. 9. The movement of the working gas in the Stirling machine is controlled by these mechanisms. The drive system is designed in such a way that the piston at the cold end should always move leading the piston in the hot end. All kinematic Stirling machines, no matter what the arrangements, are facing several significant engineering design challenges which may affect the effective application to any commercial success. Mechanical friction may be one challenge. The crank driven Stirling refrigerator with a Beta configuration is shown in Fig. 10.
2. *Free piston Stirling machine:* In response to the shortcomings of the conventional kinematic Stirling machine, in the year 1964 William Beale invented the free-piston Stirling engine (FPSE [14]). The mechanical pistons do not have any mechanical linkages therefore the crank mechanism is eliminated(see Fig. 11, 12, 13) and replaced by gas springs and metallic springs in the free-piston Stirling machine [98, 44]. The movements of the pistons are self-adapted to the required conditions by using appropriate piston-spring resonant mechanisms or pneumatic force. The thermodynamic process is strongly coupled to the piston-spring pneumatic force action. It is simple, compact and long life but it has some limitations as a less predictable motion of moving parts and performance is easily affected. Small free-piston Stirling refrigerators are widely used for the cooling of cryogenic infrared sensors.
3. *Pulse tube refrigerators (PTR):* The moving displacer in the kinetic and free piston Stirling refrigerators has limitations.The displacer creates vibration, has a short lifetime, and leads to axial heat conduction and to a shuttle heat loss. In a pulse tube refrigerator, the mechanical displacer is removed and oscillating gas flow in the thin tube produces cooling. This phenomenon is called pulse tube action. In the pulse tube refrigerator, the displacer is eliminated and replaced by a pulse tube as shown in the Fig. 14. The proper gas motion in Phase with the pressure is achieved by using phase shifters, such as orifice-reservoir, inertance tube, active displacer, etc. The reservoir is designed to be large enough that negligible pressure oscillation occurs in it during the oscillating flow. The oscillating flow through the orifice acts as displacer in separating the heating and cooling spaces. For large industrial systems, the mechanical compressor is replaced with thermoacoustic drivers to give a refrigerator with negligeble moving parts. The first pulse tube refrigerator was discovered accidentally at Syracuse University in the mid-1960s [41].

4. Reviews on Stirling cryocoolers (low temperature Stirling coolers)

4.1. General

In the previous years, the cryocoolers with Stirling cycle advanced rapidly because of their high efficiency, low power consumption, light weight, small size, fast cooldown, and high reliability etc. Because of the increasing demand for cryogenic temperatures below 120 K since the middle of the 20th century, lots of studies were conducted and resulted in a wide variety of application areas. The major applications of cryocoolers for several years were used for the military to cool infrared sensors to about 80 K and for tactical uses in tanks, airplanes and missiles [80]. The high capacity cryocooler working below 30 K cooling temperature finds lots of applications including superconducting motors, superconducting cables, and cryopumps. The cryocooler technology is extremely applied in energy, medical, and aerospace areas [70, 86, 50, 35]. This is mainly fostered with the advancement of high-temperature superconductor (HTS) materials and technological advancement. The requirements imposed in each of the above-listed applications have been difficult to meet and have been the motivation for considerable research and development in the area of cryocoolers since the past decades. Different kinds of cryocoolers are frequently experimented in laboratories and this resulted in continual improvements in the performance and reliability and implementation for commercial and space applications [81].

It is reported that the gas cycle refrigeration systems, including the Stirling refrigerator, has the highest cycle efficiency at lower source temperatures [40]. This argument

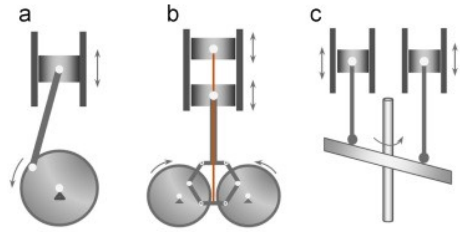

Fig. 9. Kinetic drive mechanisms used for Stirling machines: (a) crank-slider drive; (b) rhombic drive; (c) swash-plate drive; and (d) Ross-yoke drive [105]

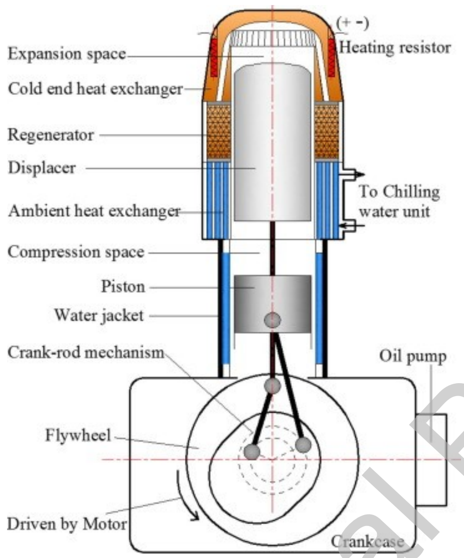

Fig. 10. Stirling cooler with kinetic drive mechanism [107]

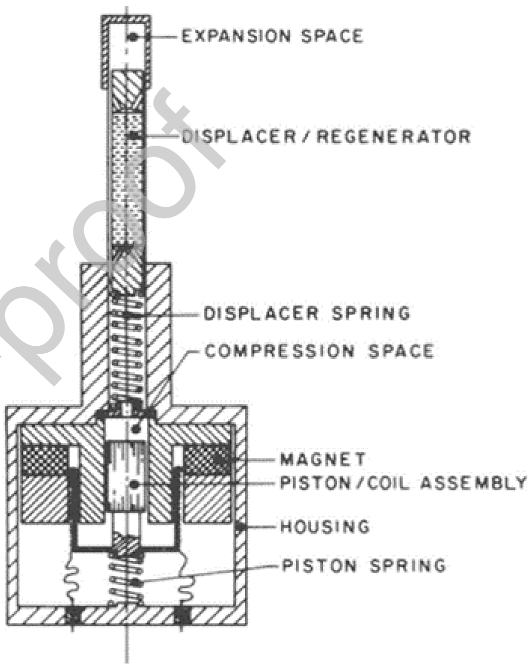

Fig. 12. Free piston configuration [44]

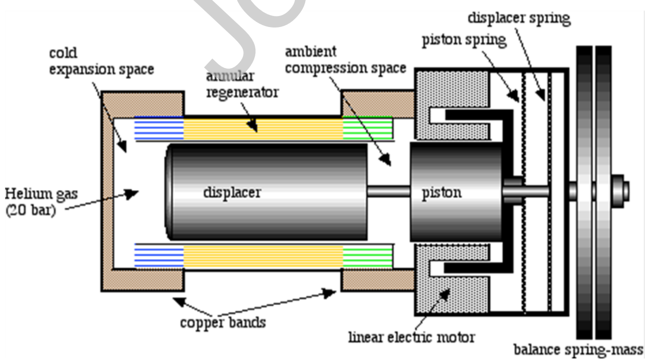

Fig. 11. Free piston configuration [98]

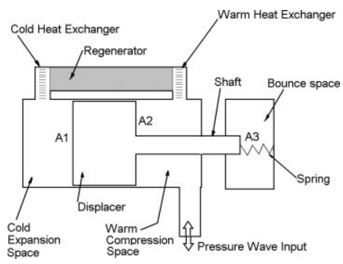

Fig. 13. Split free piston configuration [20]

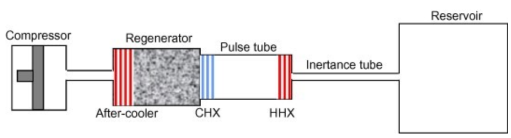

Fig. 14. stirling type pulse tube refrigerator [57]

leads to the fact that Stirling cycle refrigeration is more effective for cryogenic applications. This is because in Stirling coolers low temperatures could be obtained rapidly, which increases its effectiveness as compared to other alternative refrigerating technologies. Stirling refrigerators are well established in the cryogenic temperature range and found to be the system of choice in the miniature closed cycle. Stirling cycle machines are used for cryogenic applications with the cooling capacities ranging from fractions of a watt to 30 kW and in further research their performance and cooling range have been improved for the last decades. The working fluid is invariably gaseous with hydrogen and helium gases as the more preferred fluids because of their favorable thermodynamic and transport properties.

4.2. Kinetic driven Stirling Cryocoolers

The kinematic Stirling cryocoolers driven by crank-shaft mechanisms can generate a higher system efficiency [103]. This cooler was designed for a cooling capacity of 4 W and to operate at cryogenic temperatures of 77 K. Stirling refrigerators have been put to practical use in the fields of superconduction and space engineering for low cooling capacity cryocoolers, in order to obtain temperatures in the range of 10-15 K [75]. It is reported that mechanical cryocoolers represented a significant enabling technology for NASA's earth and space science enterprises [86].

Researchers applied different analysis techniques to design the Stirling cryocoolers and come up with different results. Microcomputer simulation for Stirling cryocoolers was developed using isothermal assumption in compression and expansion volumes [101]. The performance of Stirling cryocoolers has been analyzed by considering an adiabatic model for compression and expansion spaces [10]. It is reported that the Stirling cryocoolers are divided and studied as either fully isothermal (for constant volume process) or fully adiabatic (compression space and expansion space) and a numerical model was developed [13]. [70] developed a two-stage Stirling Cryocooler and experimented it for an infrared astronomical satellite. The researchers reported the design of the cooler and the experimental results including cooler performance, vibration, thermal vacuum, and lifetime tests. The Stirling cycle cryocooler with a low-cost, high-capacity was studied to meet different cryogenic system applications [78]. In the study, a cold end temperature near to 50 K was reported.

In the year 2002, [96] applied the concept of finite-time thermodynamics to study the effect of internal and external irreversibility for Stirling and Ericsson refrigerator at different operating conditions. Further, the researchers developed a thermo-economic optimization mode in the year 2004, to maximize the cooling load per unit cost of the system using finite time thermodynamics for an irreversible Stirling cycle cryogenic refrigerators [97]. In this research, they considered external irreversibilities due to the finite temperature difference between the system and the external reservoirs and the internal irreversibilities caused by the regenerative heat loss. A high power Stirling cryocooler has been analyzed through a numerical simulation using the SAGE software [106] and a result showed the cold head could reach a no-load cooling temperature of 38 K and a cooling capacity of 1012 W at 77.35 K with a power supply of 7.72 kW. A mathematical model has been developed based on thermodynamic theory to evaluate the lifetime of the split Stirling refrigerator [49]. In the same year, Stirling cryogenic refrigerator cycles were analyzed including both internal and external irreversibilities by a multi-objective optimization technique [3]. The optimization work includes reduction of the input power with maximization of the cooling load and the coefficient of performance (COP). [66] developed an isothermal model for a Stirling cryocooler by considering various losses at the same time. A high cooling capacity single-stage Stirling cryocooler driven by the kinetic driving system was developed and studied systematically [107]. The researchers found a cooling capacity of 700 W at 77 K and a relative Carnot efficiency of 18.2 % as shown in Fig. 15. A CFD model has been developed and validated using experimental work, to investigate the effect of various parameters for the purpose of developing an efficient miniature alpha type Stirling cooler capable of cooling to a temperature lower than $-40\,^oC$ [6].

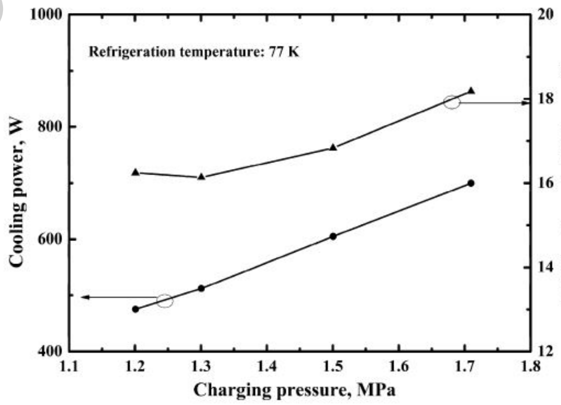

Fig. 15. cooling power and relative Carnot efficiency versus charging pressure [107]

The effect of operational and geometrical parameters on the performance of the cooler has been studied by different researchers and the associated losses were described. [73] pointed out that in the micro/nano scaled Stirling refrigeration devices the surface area of the system (boundary) has a major effect on the refrigeration heat and COP that in addition to other parameters. [33] analyzed an optimal performance of regenerative cryocoolers in the year 2011 by considering the losses due to imperfect regeneration and viscous dissipation. Results revealed that optimum performance could be found at large warm space to the regenerator void volumes ratio. [66] investigated the effects of various parameters on cooling performance of the

Stirling cryocooler and found that the biggest heat loss was due to conduction loss and the biggest work loss was due to the mechanical friction loss. The researchers also described the trend of the input power, cooling power, and COP variation with the phase shift between displacer and piston as seen in Fig. 16. The parametric study confirmed that a phase angle of 90^o and a regenerator porosity of 50 % would be desirable for the optimum performance of the investigated Stirling cooler, different from the higher porosity typically applied to Stirling engines [6].

4.3. Free-piston Stirling cryocooler(FPSC)

The free piston configuration, which was initially designed by W. Beale in the late 1960's, is one of the most novel applications of the Stirling cycle machine [14]. [1] described the phasor analysis to analyze and understand the operation of free-piston Stirling cryocoolers. [76] tested the performance of free piston/free displacer type Stirling cryocoolers and investigated the effect of operating parameters on the cooling performance. In the year 2003, the authors further evaluated the effect of the phase shift between the piston and displacer through experiments [77]. [51] conducted a dynamic analysis of small free-piston-type Stirling refrigerator to understand the characteristics of the refrigerator using an isothermal thermodynamic model. A two-dimensional axisymmetric Computational Fluid Dynamics (CFD) model was developed to characterize the thermodynamic losses for cryocooler application [55]. Theoretical and experimental studies were conducted to evaluate the phase shift characteristics including displacement, pressure and expansion phase difference for free piston and free displacer Stirling cryocooler [29]. Due to its attractive virtue of compact size and having high thermal efficiency, the free piston Stirling cryocooler (FPSC) was reported as a good as for high cooling capacity ranging from 80 K to 120 K applications [109].

In recent years, many researches have been conducted on high-capacity single-stage free piston Stirling cryocoolers. These cryocoolers could supply several hundred watts of cooling power at around 77 K, but the lowest temperature is difficult to reach. A numerical model was developed based on the thermoacoustic principle to optimize a two-stage free piston Stirling cryocooler system which could operate below 30 K [104]. The researcher reported that a cooling power of 141 W at 77 K in the first stage and 60 W cooling power at 30 K in the second stage were achieved simultaneously using PV power of about 2.23 kW. Experimental work was conducted on a two-stage free piston Stirling cooler so as to fulfill the necessity of the high temperature superconductor (HTS) motor applications [105]. A lowest cold head temperature of 27.6 K was reported with an input electric power of 3.12 kW and a mean pressure of 2.58 MPa.

A new concept that uses a pair of metal diaphragm was proposed to seal and suspend the displacer for free-piston Stirling cryocooler using SAGE modeling tools and a prototype was constructed [20]. The researcher achieved a cryogenic temperature and reported a no-load temperature of 56 K and a cooling load of 29 W at 77 K. [19] constructed CFD model as a tool to understand the underlying fluid dynamics and heat transfer mechanisms that happened inside a diaphragm Stirling Cryocooler with the aim to improve the performance. The results of CFD analysis showed good agreement with experimental values and also highlighted possible areas of improvement such as increasing the length of the warm heat exchanger to achieve relatively better cooling performance. Furthermore, the model showed that the gas was found most coldest and at the highest velocity in the center of the expansion space.

4.4. Stirling type Pulse Tube Cryocooler (SPTC)

The basic Pulse tube refrigerator was first described with experimental models in the year 1964 [41]. After it was first perceived in the mid-1960s, pulse tube refrigerator has been an academic interest until the middle of 1980s as the demand for cryogenic temperature steadily increases [80]. Since then, improvements in its efficiency have been increased rapidly. [68] included an orifice tube inside the pulse tube close to the warm end and obtained a cooling temperature of 105 K. [89] introduced the concept of double Inlet Pulse tube refrigerator. Mathematical investigation and experimental findings have confirmed that the double inlet pulse tube cooler has higher performance as compared with the orifice pulse tube refrigerator. It was reported that on the heels of basic research, commercial developers are harnessing acoustic processes in gases to make reliable, inexpensive engines and cooling devices with no moving parts and a significant fraction of Carnot's efficiency [91]. Since there is no or only one moving part in pulse tube refrigerators, the different arrangements of these refrigerators such as an orifice, double inlet, and inertance tube, pulse tube refrigerators were considered as superior to most other Stirling refrigerators [68, 89, 94, 113].

The performance of a pulse tube expander was experimentally investigated [61]. [95] developed a high capacity miniature pulse tube Cryocooler which could provide large cooling power for space application over a range of temperatures. Based on efficiency, maintainability, operational flexibility, the feasibility of integration and performance/cost ratio criterion, pulse tube refrigerator was found to be applicable for HTS from 20 K to 77 K temperature ranges [43]. [115] described the rapid evolution of pulse tube cryocooler from a laboratory-based prototype to a commercial offering that produces cooling of 200 W at 80 K.

[110] developed a thermodynamic model for single stage Stirling type pulse tube cooler and constructed a prototype for experimentation. The researchers' test result showed the cooler reached a lowest no-load temperature of 37 K within 30 minutes and a cooling capacity of 50 W at 55 K with an input power of 3.4 kW. The researchers further reported that at 80 K of cooling temperature, 11% Carnot efficiency has been obtained. The design and fabrication of large capacity, single stage, Stirling type, Pulse tube

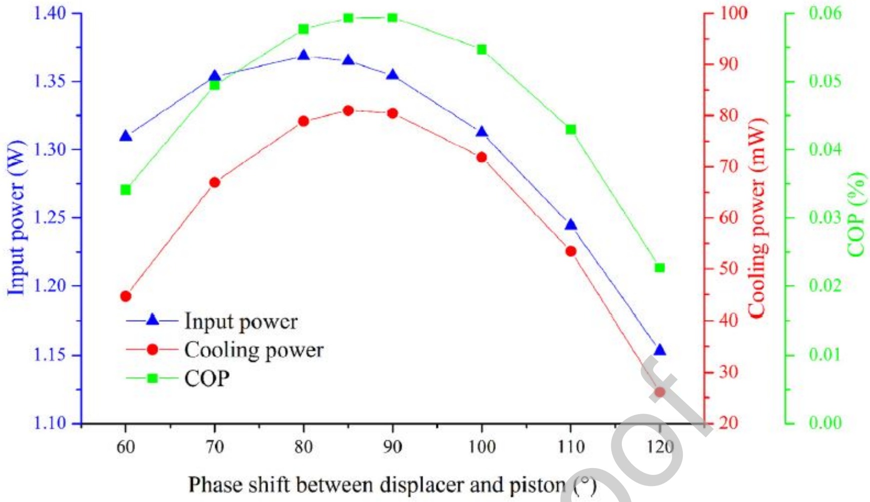

Fig. 16. Input power, cooling power, and COP trend vs phase shift between displacer and piston.[66]

Table 1: Summary of Stirling cooler performance

Author & year	Method of Research	Main Findings	Remark
[106]	Numerical simulation	a no-load temperature of 38 K & 1012 W at 77.35 K with a PV power of 7.72 kW	Kinetic drive
[107]	Experimental test	a cooling power of 700 W at 77 K & a relative Carnot efficiency of 18.2% has been achieved	Kinetic drive
[66]	Isothermal model	the biggest heat loss is conduction loss & biggest work loss is the mechanical friction loss	Kinetic drive
[6]	CFD modeling & testing	the optimum performance found at phase angle of 90^o and a regenerator porosity of 50%	Alpha kinetic drive
[104]	Numerical optimization	141 W at 77 K in the first stage & 60 W at 30 K in the second stage at input power of 2.23 kW	FPSC
[102]	Numerical & experimental	a lowest temperature of 27.6 K with an input power of 3.12 kW & a mean pressure of 2.58 MPa	FPSC
[28]	Design & testing	lowest temperature of 15.5 K & 386 mW/20 K cooling power at 246 W input power	single stage SPTC
[27]	Design & testing	lowest temperature of 3.6 K & 6 mW/4.2 K cooling power at 250 W input power	three-stage SPTC

refrigerator(PTR) was developed followed by a sequence of design modifications which have been focused on the optimization of the flow transition [79].

An experimental investigation was conducted and a numerical model was developed using the SAGE software to understand the origin of parasitic streaming in the regenerator of Stirling-type pulse tube cryocooler [34]. [114] introduced a nodal numerical model for the simulation of the pulse tube Stirling machine. It has been reported that two single-stage high-frequency coaxial PTCs, with each of them reaching a cooling temperature lower than 30 K and input power of less than 250 W have been designed and tested [108]. It has been further explained that for single stage PTC, optimization with both double-inlet and inertance tube is of crucial to achieving temperature below 30 K. A single-stage high-frequency multi-bypass pulse tube cryocooler(PTC) was designed and experimentally tested [25]. The researchers reported that a single-stage multi-bypass PTC achieved the lowest temperature of 18.6 K with an input power of 268 W, and it was the lowest temperature reported so far with single stage PTC. In the same year, the author designed and tested a single stage Stirling type pulse-tube cryocooler by optimizing the regenerator and a new lower temperature of 15.5 K has been recorded [28]. The new optimized design was reported as it could deliver a cooling power of 386 mW at 20 K for an input power of 246 W which was found to be comparable to the two-stage Stirling pulse tube cryocooler with the same input power.

A three stage high frequency pulse tube cryocooler (HPTC) has been developed, and a no-load temperature of 3.6 K has been reported using helium-4 as working gas [27]. The temperature found from the research has been reported as the lowest temperature found for high frequency pulse tube cryocooler. The researcher has made four improvements (on compressor, regenerator material, interance tube and on cold inlet structure) to optimize the design. [53] designed a pulse tube cryocooler to suit the demands of small liquid natural gas (LNG) distribution stations which could work at a temperature of 120 K, cooling power of 1.2 KW and a relative Carnot efficiency greater than 20 %. The researchers pointed out that one-third of the acoustic work is dissipated in the inertance tube and resulted in efficiency deterioration.

[112] developed a high frequency coaxial single stage like structure and multi-bypass PTR which could achieve a no-load temperature of 13.9 K at an input power of 250 W . [111] established a two-dimensional axis-symmetric CFD model of a miniature coaxial Stirling pulse tube cooler. A numerical simulation was conducted for an inertance tube PTR by removing the dissipative inertance tube and reservoir and replacing a mass-spring displacer directly coupled to a compression space with an objective of recovering power dissipated [102]. From the simulation result, researchers concluded that the COP could be significantly improved due to the extra power recovered by the mass-spring displacer. Theoretical analyses and modeling were conducted to evaluate the dynamic and thermodynamic characteristics of moving-coil linear compressor in the inertance tube Stirling pulse tube cryocooler [31]. The researchers deduced the characteristics governing equations. To verify the theoretical analyses and modeling conducted [31] and experimental investigations were carried out [32]. A better efficient cascade cryocooler was analyzed and experimentally tested, that is capable of recovering most of the expansion work wasted as heat in a pulse-tube cryocooler [107]. A modified type pulse tube refrigerator was proposed numerically to reduce the direct current gas losses that could have an impact on cooling power in the orifice type pulse tube refrigerator by introducing an additional compressor [87].

The effect of the regenerator and pulse tube length on the efficiency of a pulse tube cryocooler was analyzed [54]. The effects of geometrical parameters of the linear compressor on the performances of Stirling-type pulse tube cryocooler were mathematically investigated and some experimental work was performed [32]. [92] analyzed a double -inlet pulse tube refrigerator focusing on the interaction between Gedeon streaming and the local temperature. Experimental work was conducted and a 32 % cooling efficiency improvement was found for a cascade cryocooler as compared with a traditional single-stage pulse-tube cryocooler [107]. In the report these cascade pulse tube cryocoolers, the second-stage cooler covered around 18.1 % of the overall cooling power. A cold end temperature of 98 K was achieved by a modified pulse tube refrigerator based on the CFD simulation [87], whereas a simple orifice type pulse tube refrigerator with the same dimension reached a cold end temperature of 130 K.

[54] reported that the pulse tube cryocooler could offer 520 W of cooling power at a cooling temperature of 80 K with a relative Carnot efficiency of 18.2%. [112] developed a pulse tube refrigerator that could be a potential choice especially for small cooling loads at a low temperature such as 31.3 K and 20 K for 50 W and 100 W power inputs respectively. [111] analyzed and calculated the different types of regenerator losses, and reported that the pressure drop is dominant for most regenerator's length and decrease monotonously from warm space to cold space. Double -inlet pulse tube refrigerator was simulated to analyze the impact of Gedeon streaming on the efficiency of the machine [92].

5. Reviews on moderate temperature Stirling refrigerating machine

A prototype free-piston Stirling refrigerator intended for domestic cooling was experimentally tested and results were reported [38]. [40] reported that the gas cycle refrigeration systems including the Stirling refrigerator have the highest cycle efficiency at lower source temperatures. According to the classical thermodynamics, the reversible Stirling refrigerating cycle has the same coefficient of performance with a reversible Carnot refrigeration cycle for the same temperature range under the perfect regenerative conditions [23]. [16] discussed the impact of irreversibilities for moderate temperature Stirling refrigerating machine with reference to the machine design. The researcher reported the configuration of free-piston Stirling cycle refrigerating machine resulted in better performance than the crank driven Stirling refrigerating machine. The free piston Stirling refrigeration machines were found superior to the vapor compression refrigeration machine for domestic purpose [17, 21]. The researchers further described that the FPSCs are far more superior than the VCR in their respective COP. [17] explained the development of free-piston Stirling refrigeration machine continues with noticeable achievements and they are expected to meet or exceed the life time and reliability expectations of ordinary refrigerators while being cost competitive. Based on the optimization work conducted in the year 1996, it was found that the free-piston Stirling coolers together with super insulated cabinets and cold stores offer an ideal combination for practical photovoltaic powered refrigerators [18].

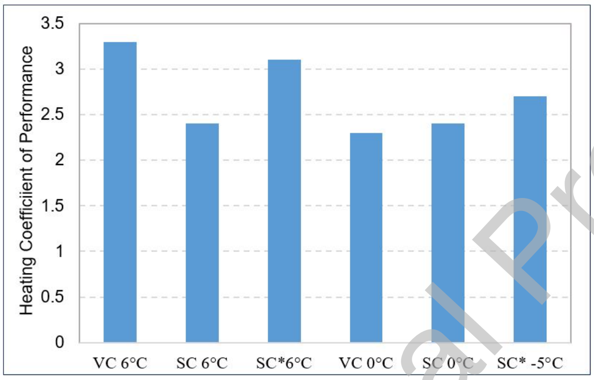

Fig. 17. results from the Stirling-cycle heat-pump development program [48]

VC is typical vapour compression machine, SC is Stirling machine prior to seal development program, SC* is Stirling machine after seal development program. The indoor temperature in all cases is 20C.

The impact of finite-rate heat transfer and regenerative losses on Stirling refrigerators cooling performance was investigated [24]. A finite time heat transfer analysis was also conducted in the year 1998 for an air refrigeration cycle with non-isentropic compression and expansion [26]. The researcher derived the relation between COP and cooling load with pressure ratio. [22] developed an irreversible cycle model to predict the performance and the input power required for the Stirling refrigerator optimized on a specified cooling capacity. Some dimensionless parameters has been proposed to link the entropy variation rate and the temperature differences at the heat exchangers for an irreversible refrigerators [42].

Researches were conducted on an optimal relation between the cooling rate and the coefficient of performance of the refrigerating machine [24, 22, 84]. [48] reported that there are conditions for Stirling refrigerating machines to provide higher performance than vapor compression machines. The finding of the researchers can be seen in Fig. 17. The maximum cooling capacity was obtained when the volume ratio was almost equal to the temperature ratio for the developed beta type prototype refrigerator [75]. The researchers also found that nitrogen had a better cooling capacity than helium. [73] pointed out the refrigeration heat capacity and COP of large-scaled Stirling refrigeration are independent with the surface area. The COP of V-type Stirling-cycle refrigerator near-ambient cooling conditions has been reported to be comparable with the vapor compression refrigerating systems and applicable for domestic refrigeration [9, 65] and it is confirmed as seen in Fig. 17. The impact of three different fluids (air, hydrogen, and helium) on the performance of a Stirling refrigerator was investigated and the result showed that almost all the three working gases behave in a similar way but hydrogen slightly outperforms most of the time [93].

A Beta-type Stirling cycle machine of 100 W refrigeration capacity for household refrigeration application was mathematically designed and experimentally tested [75]. The researchers studied the effect of parameters such as dead volume ratio, the ratio of the compression volume to the expansion volume, types of working fluids, and the piston and displacer phase difference on the refrigerating performance. [9] conducted a thermodynamic control volume analysis on a V-type Stirling-cycle refrigerator subjected to periodic mass flow and evaluated the work done, instantaneous pressure, and coefficient of performance of the Stirling refrigerator. The V-type integral Stirling refrigerator (VISR) for domestic application was investigated based on simulation and experimentations and the parameters such as the power intake and the coefficient of performance were analyzed under different rotating speeds and charged pressures [65]. Furthermore, [93] investigated the thermodynamic performance of a V-type Stirling refrigerating machine in a view for higher cooling capacity and COP as an alternative to vapor compression type refrigerator. In this research, the effect of porosity of regenerator, the phase angle between the pistons, pressure and speed on COP and cooling capacity for the three working fluids had been investigated. [84] developed a thermodynamic model for an exergy flow analysis for Stirling refrigerator. [88] pointed out that one-dimensional differential models are important tools for the design and optimization of regenerative machines since they require far less computing time than multi-dimensional computational fluid dynamics models and are still capable of describing the various losses.

Stirling heat pump was investigated as a realistic alternative to vapor compression machine [3, 48, 2]. A non-linear mathematical model was developed for an air-filled alpha Stirling refrigerator by incorporating air thermodynamics, heat transfer from the walls, as well as heat transfer and fluid resistance in the regenerator and different

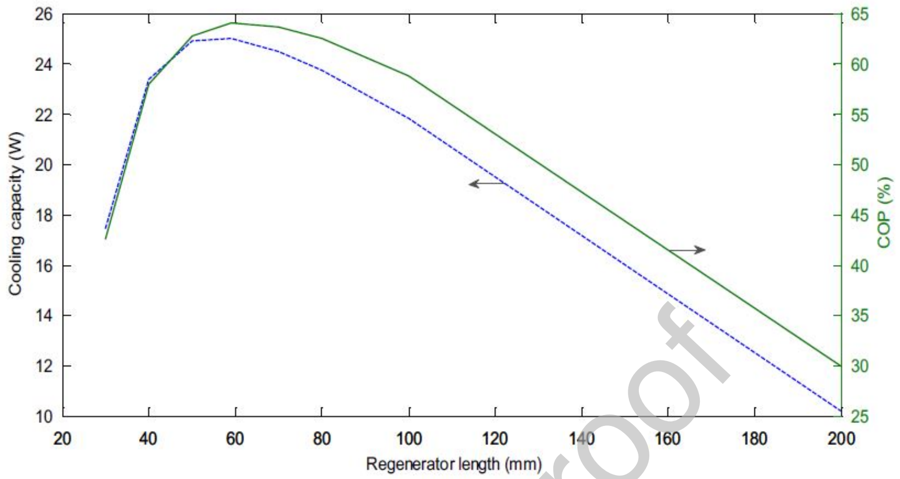

Fig. 18. Cooling capacity and COP evolution versus regenerator length [45]

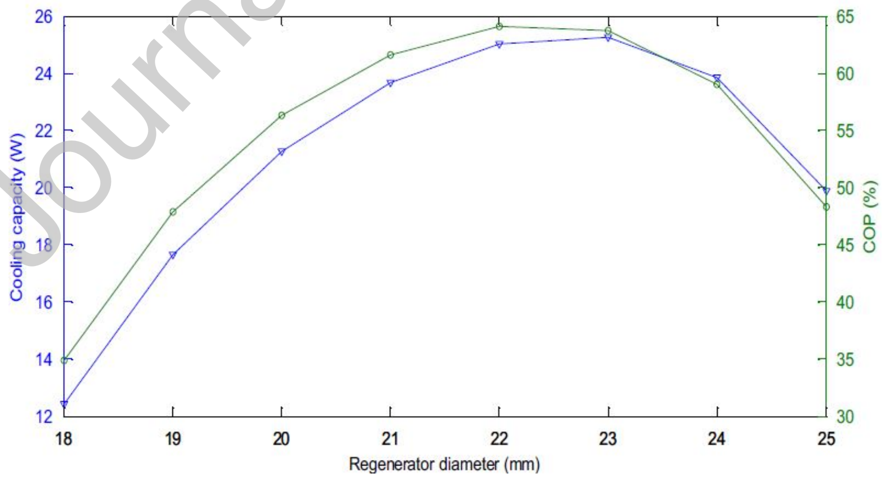

Fig. 19. Stirling refrigerator performances versus regenerator diameter [45]

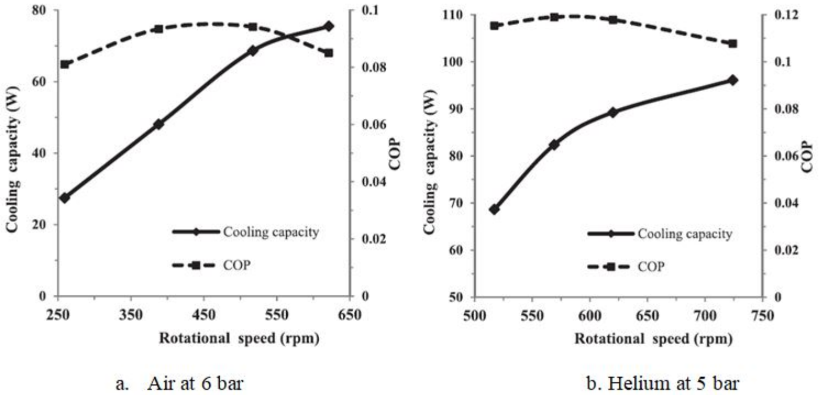

Fig. 20. The variation of the cooling capacity and COP with rotational speed for air and helium gases [12]

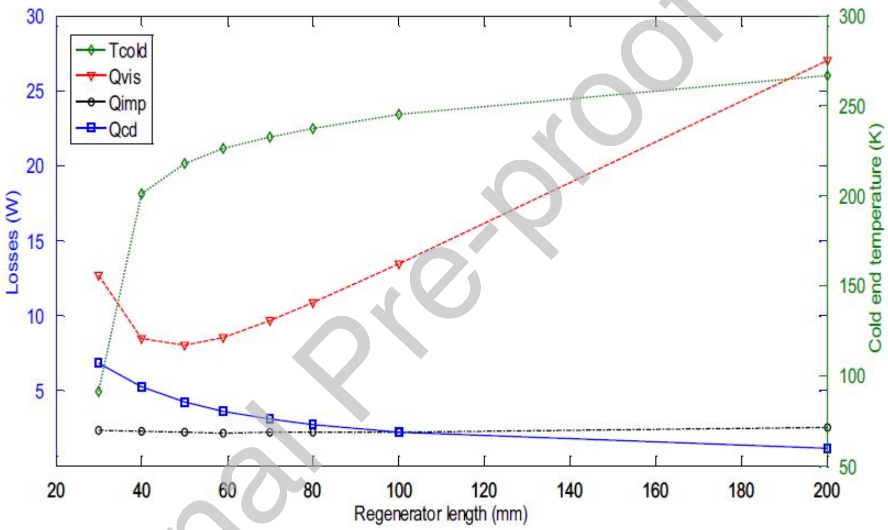

Fig. 21. Cold end temperature and losses evolution versus regenerator length. [45]

variables were determined for both working spaces [67]. Furthermore, the researchers tested the machine experimentally by replacing the piston-cylinder assembly with a flexible chamber and investigated the performance of the refrigerator. [4] optimized a Stirling heat pump using multi-objective criteria in finite time. The researchers simultaneously considered the three objective functions such as the maximization of heating load and coefficient of performance, and the minimization of input power of the Stirling heat pump. The researcher analyzed both external and internal irreversibilities. External irreversibility is due to a finite temperature difference between the gas and the heat exchangers while the internal irreversibility is due to the regenerative heat loss and entropy generation. In the year 2015, the author used the finite time thermodynamics approach for thermo-economic multi optimization of Stirling heat pumps [2]. [47] evaluated the performances of a Beta type Stirling refrigerating machine having a regenerative displacer by considering complex phenomena which are related to compressible fluid mechanics, thermodynamics and heat transfer for the energy analysis. To design an experimental Stirling cycle refrigeration unit, an adiabatic numerical model was developed [58]. The research presented a simple time discretization model considering the cylinders as adiabatic spaces and the results were verified by another modeling technic called the full three dimensional CFD. The result of the analysis showed the compliance of the adiabatic model with the full 3D CFD model. Various Stirling refrigerators were analyzed with and without a regenerator and the coefficient of performance of the machine was evaluated [69]. [45] developed a thermodynamic model and conducted an experimental validation to optimize an air-filled Beta-type Stirling refrigeration machine with special attention to evaluate the effect of geometrical parameters such as dead space volume and swept volume of the compression and expansion

Table 2: Review summary of Stirling refrigerator for moderate cooling

Author	Method of research	main Findings	Remark
[38]	Experimental	For a cold end temperature of -33^0C and hot end temperature of 18^0C, the COP was found to be 0.217	FPSR
[17]	Assessment & testing	reported 60% of Carnot efficiency achievable for domestic refrigeration	FPSR
[24]; [26]	investigation	Internal loss has major impact than external irreversibility	
[22]; [84]	investigation	the optimal relation ship between the cooling rate and COP identified	
[75]	Simulation & testing	Hydrogen has better cooling capacity than helium and dead volume has major impact on cooling capacity	Beta
[9]	Thermodynamic analysis	COP of the Alpha machine is comparable with the VCR	Alpha
[65]	Simulation and experiment	helium and nitrogen -20 to -600c cold end temperature ,100w	Alpha
[93]	Thermodynamic analysis	hydrogen relatively performs better 230-360K ,450W	Alpha
[48]; [2]	Experiment & multi-objective	Stirling heat pump is a good alternative to vapor compression heat pump even have better performance	Heat pump
[58]	Adiabatic model & experiment	The developed model could find applications for design and optimization due to its simplicity	Alpha
[45]	Thermodynamic & experiment	the optimal value of diameter and length for beta type stirling refrigerator prototype is about 22mm and 60 mm	Beta
[12]	Experiment & Analytical inv.	minimum achievable temperature are -94.2^oC and -42.6^oC for helium and air when the rotational speed and working pressure varied between 260 and 775 rpm and 4 and 7 bar, respectively	Gamma

space. The regenerator parameters on the performance of refrigerator were also analyzed and optimized by the researchers(see Fig. 18 and 19). [21] developed a mathematical formulation using loss factor to free piston Stirling cooler for domestic refrigeration application. Multi-objective optimization has been performed using non-ideal adiabatic analysis to optimize Gamma type Stirling refrigerator [12]. The researchers indicated that the cooling capacity increased with the increase of rotational speed based on analysis and experimental results where the COP has attained a maximum value. Furthermore, they described that the optimum COP value for the air occured at a lower rotational speed than that of helium(see in Fig. 20).

An experimental result of COP for free piston Stirling refrigerator has been reported to be 0.217 for a cold end temperature of $-33\,^oC$ and hot end temperature of $18\,^oC$ [38]. The efficiencies of free-piston Stirling refrigeration machine working at domestic refrigerating temperature were reported to exceed 32% of Carnot efficiency and are expected to achieve very high efficiencies and replace the domestic refrigerators while achieving cost competitiveness [17]. [74] conducted an experimental work and reported the COP values for FPSCs operating with warm head temperatures close to $30\,^oC$ were typically found between 2 and 3 for cold head temperatures around $0\,^oC$, falling to around 1 for temperatures approaching $-40\,^oC$. [47] predicted that the drop-in heat exchanger efficiency would influence the pressure ratio and so the refrigeration capacity would be reduced substantially. [45] evaluated different losses associated with Stirling refrigerator that directly affect cooling performance of the refrigerator and they described these losses are functions of regenerator diameter and length as seen in Fig. 21. The researchers pointed out that the optimal values of diameter and length for the prototype Beta-type Stirling refrigerator are about 22 mm and 60 mm respectively.

6. Conclusion

Currently, the vapor compression refrigeration (VCR) technologically exists at the saturated level and is suited for household cooling applications. Although, VCR is an efficient technology especially for moderate temperature refrigeration applications, because of CFC refrigerants the environmental concern is increasing. Stirling refrigerators for moderate temperature application is one of the technologies that is brought from the concern about the use of CFC refrigerants and their effect on the environment. A detailed review of literature has been conducted about the research and development made on the Stirling cycle refrigerators. The main aim of this, review paper is to provide comprehensive information about regenerative Stirling cycle refrigeration machines. This Stirling cycle refrigerators are the counterpart of work producing Stirling cycle engines and generally have the same configurations and driving mechanisms as their counterparts. The review covers the working principle, configurations and drive mechanisms of reversed regenerative Stirling cycle machine, detailed review on the development status of Stirling coolers for low temperature and moderate temperature applications. The main conclusions are:

- Lots of researches have been done on low-temperature coolers with different drive mechanisms. As a result of this, Stirling coolers have reached comparable performance with other technologies and are widely applied at cryogenic temperature applications.
- The number of researches are limited on Stirling cycle refrigerators for moderate temperature applications. This is mainly because of the current most commonly used vapor compression refrigerating technology is found at the optimal technological level and cost.
- Recently the demand for alternative refrigerating technology is increasing due to environmental concern resulted mostly from the existing technology CFC gas emissions.
- For Stirling cryocoolers, the pulse-tube drive mechanism with multi-stage has achieved the lowest temperature with a three-stage high-frequency pulse tube cryocooler (HPTC) a no-load temperature of 3.6 K reported.
- Even though the choice of heat exchangers design and working volumes for Stirling machine depends on the application area, most researches that have done so far particularly on moderate temperature Stirling cycle refrigerator are directly experimented by reversing the existing Stirling engine. For instance, the heater and cooler of the engine are acting as chiller and hot heat exchanger respectively for refrigerating machine. Furthermore, the regenerator parameters including the porosity and the relative size of working volumes have a greater effect on the performance of the machine. In the Stirling engine, all the components are designed and manufactured to produce the maximum power output or efficiency. Hence, such reversing a machine configuration may not produce the optimum cooling or COP.
- Results from the researches done so far on the Stirling cycle refrigerating machines for moderate temperature household application shows that Stirling cycle machines are promising alternatives to the current technology in use for the same application.
- It has been recognized that one of the challenges of Stirling machines is the cost of production as it demands higher accuracy manufacturing to minimize gas leakage as well as to create smooth motion between parts and the material has to be selected to withstand the high temperature and pressure. The operating temperature and pressure ranges of moderate temperature cooling machine are by far lower

than that of its counterpart. Therefore, it could be possible to reduce the cost of cooling machine through designing and selecting a reasonable material.

- The other development problems of all Stirling machines including coolers is its difficulty to avoid gas leakage and different thermal losses. In addition, for moderate temperature cooling application, developing Stirling refrigeration machines that could outperform the conventional VCR is still challenging. However, global warming potential (GWP) concern has to be taken in to account for the comparison of different technologies.

- The researches done so far on moderate temperature Stirling refrigerator did not show clearly the configurations that have better performance for a different range of cooling temperatures.

The global concern to create clean environment and the increased demand in refrigeration; on the other hand the high global warming potential of refrigerates used in conventional VCR technology lead to the increased interest in alternative technologies. Stirling cycle refrigerator for moderate cooling application is found to be a potential candidate. However, the Stirling cycle machines as a general have several limitations, the efforts done so far on the design of Stirling refrigerating machine is very limited and does not consider fully the design and developmental requirements of the cooling machine for particular applications. Therefore, future researches on Stirling cycle moderate temperature cooling machines have to be designed considering the operating condition as well as suitable configuration, all components must be produced, and tested based on the cooling temperature requirement.

References

[1] Ackermann, R. (1981). Dynamic analysis 0f a small free-piston res0nant cry0refrigerat0r. In *Refrigeration for Cryogenic Sensors and Electronic Systems: Proceedings of a Conference Held at the National Bureau of Standards, Boulder CO, October 6-7, 1980*, volume 607, page 57. US Department of Commerce, National Bureau of Standards.

[2] Ahmadi, M. H., Ahmadi, M. A., Bayat, R., Ashouri, M., and Feidt, M. (2015). Thermo-economic optimization of stirling heat pump by using non-dominated sorting genetic algorithm. *Energy Conversion and Management*, 91:315–322.

[3] Ahmadi, M. H., Ahmadi, M. A., Mohammadi, A. H., Feidt, M., and Pourkiaei, S. M. (2014a). Multi-objective optimization of an irreversible stirling cryogenic refrigerator cycle. *Energy Conversion and Management*, 82:351–360.

[4] Ahmadi, M. H., Ahmadi, M.-A., Mohammadi, A. H., Mehrpooya, M., and Feidt, M. (2014b). Thermodynamic optimization of stirling heat pump based on multiple criteria. *Energy Conversion and Management*, 80:319–328.

[5] Ahmadi, M. H., Ahmadi, M.-A., and Pourfayaz, F. (2017). Thermal models for analysis of performance of stirling engine: A review. *Renewable and Sustainable Energy Reviews*, 68:168–184.

[6] Ahmed, H., Almajri, A. K., Mahmoud, S., Al-Dadah, R., and Ahmad, A. (2017). Cfd modelling and parametric study of small scale alpha type stirling cryocooler. *Energy Procedia*, 142:1668–1673.

[7] Aksoy, F., Solmaz, H., Karabulut, H., Cinar, C., Ozgoren, Y., and Polat, S. (2016). A thermodynamic approach to compare the performance of rhombic-drive and crank-drive mechanisms for a beta-type stirling engine. *Applied Thermal Engineering*, 93:359–367.

[8] Altin, M., Okur, M., Ipci, D., Halis, S., and Karabulut, H. (2018). Thermodynamic and dynamic analysis of an alpha type stirling engine with scotch yoke mechanism. *Energy*, 148:855–865.

[9] Ataer, Ö. E. and Karabulut, H. (2005). Thermodynamic analysis of the v-type stirling-cycle refrigerator. *International Journal of Refrigeration*, 28(2):183–189.

[10] Atrey, M., Bapat, S., and Narayankhedkar, K. (1990). Cyclic simulation of stirling cryocoolers. *Cryogenics*, 30(4):341–347.

[11] Babaelahi, M. and Sayyaadi, H. (2015). A new thermal model based on polytropic numerical simulation of stirling engines. *Applied Energy*, 141:143–159.

[12] Batooei, A. and Keshavarz, A. (2018). A gamma type stirling refrigerator optimization: An experimental and analytical investigation. *International Journal of Refrigeration*.

[13] Bauwens, L. (1994). Adiabatic losses in stirling cryocoolers: a stratified flow model. *Cryogenics*, 34(8):627–633.

[14] Beale, W. T. (1969). Free piston stirling engines-some model tests and simulations. Technical report, SAE Technical Paper.

[15] Beale, W. T. (1971). Stirling cycle type thermal device. US Patent 3,552,120.

[16] Berchowitz, D. (1992). Free-piston stirling coolers. In *International Refrigeration and Air Conditioning Conference.*

[17] Berchowitz, D. M. (1993). Free-piston rankine compression and stirling cycle machines for domestic refrigeration. In *Greenpeace Ozon Safe Conference, Washington, DC.*

[18] Berchowitz, D. M. (1996). Stirling coolers for solar refrigerators. In *International Appliance Technical Conference, West Lafayette, US.*

[19] Caughley, A., Sellier, M., Gschwendtner, M., and Tucker, A. (2016a). Cfd analysis of a diaphragm free-piston stirling cryocooler. *Cryogenics*, 79:7–16.

[20] Caughley, A., Sellier, M., Gschwendtner, M., and Tucker, A. (2016b). A free-piston stirling cryocooler using metal diaphragms. *Cryogenics*, 80:8–16.

[21] Chaudhari, P., DSouza, D., Borkar, S., Haribhakta, V., and Trimbake, S. (2016). Mathematical formulation of free piston stirling cooler for domestic refrigeration using loss factor. *International Journal of Scientific & Engineering Research*, 7(11):1505–1510.

[22] Chen, J. (1998). Minimum power input of irreversible stirling refrigerator for given cooling rate. *Energy conversion and management*, 39(12):1255–1263.

[23] Chen, J. and Yan, Z. (1993). Regenerative characteristics of magnetic or gas stirling refrigeration cycle. *Cryogenics*, 33(9):863–867.

[24] Chen, J. and Yan, Z. (1996). The general performance characteristics of a stirling refrigerator with regenerative losses. *Journal of Physics D: Applied Physics*, 29(4):987.

[25] Chen, L., Jin, H., Wang, J., Zhou, Y., Zhu, W., and Zhou, Q. (2013a). 18.6 k single-stage high frequency multi-bypass coaxial pulse tube cryocooler. *Cryogenics*, 54:54–58.

[26] Chen, L., Wu, C., and Sun, F. (1998). Cooling load versus cop characteristics for an irreversible air refrigeration cycle. *Energy Conversion and Management*, 39(1-2):117–125.

[27] Chen, L., Wu, X., Wang, J., Liu, X., Pan, C., Jin, H., Cui, W., Zhou, Y., and Wang, J. (2018). Study on a high frequency pulse tube cryocooler capable of achieving temperatures below 4 k by helium-4. *Cryogenics*, 94:103–109.

[28] Chen, L., Zhou, Q., Jin, H., Zhu, W., Wang, J., and Zhou, Y. (2013b). 386 mw/20 k single-stage stirling-type pulse tube cryocooler. *Cryogenics*, 57:195–199.

[29] Chen, X., Wu, Y. N., Zhang, H., and Chen, N. (2009). Study on the phase shift characteristic of the pneumatic stirling cryocooler. *Cryogenics*, 49(3-4):120–132.

[30] Cheng, C.-H. and Yu, Y.-J. (2011). Dynamic simulation of a beta-type stirling engine with cam-drive mechanism via the combination of the thermodynamic and dynamic models. *Renewable energy*, 36(2):714–725.

[31] Dang, H., Zhang, L., and Tan, J. (2016a). Dynamic and thermodynamic characteristics of the moving-coil linear compressor for the pulse tube cryocooler. part a: Theoretical analyses and modeling. *international journal of refrigeration*, 69:480–496.

[32] Dang, H., Zhang, L., and Tan, J. (2016b). Dynamic and thermodynamic characteristics of the moving-coil linear compressor for the pulse tube cryocooler: Part b–experimental verifications. *International Journal of Refrigeration*, 69:497–504.

[33] De Boer, P. (2011). Optimal performance of regenerative cryocoolers. *Cryogenics*, 51(2):105–113.

[34] Dietrich, M., Yang, L., and Thummes, G. (2007). High-power stirling-type pulse tube cryocooler: Observation and reduction of regenerator temperature-inhomogeneities. *Cryogenics*, 47(5-6):306–314.

[35] Duband, L. (2015). Space cryocooler developments. *Physics Procedia*, 67:1–10.

[36] Egas, J. and Clucas, D. M. (2018). Stirling engine configuration selection. *Energies*, 11(3):584.

[37] Erol, D., Yaman, H., and Doğan, B. (2017). A review development of rhombic drive mechanism used in the stirling engines. *Renewable and Sustainable Energy Reviews*, 78:1044–1067.

[38] Fabien, M. (1991). Evaluation of the free-piston stirling cycle for domestic cooling applications. *Int. Congr. of Refrig. Montreal, Quebec, Canada.*

[39] Formosa, F. and Despesse, G. (2010). Analytical model for stirling cycle machine design. *Energy Conversion and Management*, 51(10):1855–1863.

[40] Gauger, D. C. (1993). *Alternative technologies for refrigeration and air conditioning applications.* PhD thesis.

[41] Gifford, W. E. and Longsworth, R. (1964). Pulse-tube refrigeration. *Journal of Engineering for Industry*, 86(3):264–268.

[42] Grazzini, G. and Rocchetti, A. (2014). Thermodynamic optimization of irreversible refrigerators. *Energy conversion and management*, 84:583–588.

[43] Gromoll, B. (2004). Technical and economical demands on 25k–77k refrigerators for future htsseries products in power engineering. In *AIP Conference Proceedings*, volume 710, pages 1797–1804. AIP.

[44] Haarhuis, G. (1978). The mc 80-a magnetically driven stirling refrigerator. *Cryogenics*, 18(12):656–658.

[45] Hachem, H., Gheith, R., Aloui, F., and Nasrallah, S. B. (2017). Optimization of an air-filled beta type stirling refrigerator. *Inter-*

national Journal of Refrigeration, 76:296–312.

[46] Hachem, H., Gheith, R., Aloui, F., and Nasrallah, S. B. (2018). Technological challenges and optimization efforts of the stirling machine: A review. *Energy Conversion and Management*, 171:1365–1387.

[47] Hachem, H., Gheith, R., Nasrallah, S. B., and Aloui, F. (2015). Impact of operating parameters on beta type regenerative stirling machine performances. In *ASME/JSME/KSME 2015 Joint Fluids Engineering Conference*, pages V001T22A002–V001T22A002. American Society of Mechanical Engineers.

[48] Haywood, D., Raine, J., and Gschwendtner, M. (2002). Stirling-cycle heat-pumps and refrigerators–a realistic alternative?

[49] He, Y.-L., Zhang, D.-W., Yang, W.-W., and Gao, F. (2014). Numerical analysis on performance and contaminated failures of the miniature split stirling cryocooler. *Cryogenics*, 59:12–22.

[50] Hirai, H., Suzuki, Y., Hirokawa, M., Kobayashi, H., Kamioka, Y., Iwakuma, M., and Shiohara, Y. (2009). Development of a turbine cryocooler for high temperature superconductor applications. *Physica C: Superconductivity*, 469(15-20):1857–1861.

[51] Hong, Y., Park, S., Kim, H., and Koh, D. (2003). Dynamic analysis of a free piston stirling refrigerator. In *Cryocoolers 12*, pages 103–108. Springer.

[52] Horn, S. and Walters, B. (1974). Split cycle cryogenic cooler with rotary compressor. US Patent 3,853,437.

[53] Hu, J., Chen, S., Zhu, J., Zhang, L., Luo, E., Dai, W., and Li, H. (2016). An efficient pulse tube cryocooler for boil-off gas reliquefaction in liquid natural gas tanks. *Applied energy*, 164:1012–1018.

[54] Hu, J., Zhang, L., Zhu, J., Chen, S., Luo, E., Dai, W., and Li, H. (2014). A high-efficiency coaxial pulse tube cryocooler with 500 w cooling capacity at 80 k. *Cryogenics*, 62:7–10.

[55] Huang, T., Caughley, A., Young, R., and Chamritski, V. (2008). Cfd simulation and experimental validation of a diaphragm pressure wave generator. pages 385–390. Georgia Institute of Technology, Cryocoolers 16, International Cryocoolers Conference Inc., Boulder, Colorado.

[56] Ipci, D. and Karabulut, H. (2018). Thermodynamic and dynamic analysis of an alpha type stirling engine and numerical treatment. *Energy Conversion and Management*, 169:34–44.

[57] Jafari, S., Mohammadi, B., and Boroujerdi, A. (2013). Multi-objective optimization of a stirling-type pulse tube refrigerator. *Cryogenics*, 55:53–62.

[58] Jan, W. and Marek, P. (2016). Mathematical modeling of the stirling engine. *Procedia Engineering*, 157:349–356.

[59] Khirzada, H. (2016). *Optimisation et caractrisation d'un moteur Stirling de faible puissance pour la gnration lectrique*. PhD thesis, Univ. Franche-Comte.

[60] Kirk, A. C. (1874). On the mechanical production of cold.(includes plates and appendix). In *Minutes of the Proceedings of the Institution of Civil Engineers*, volume 37, pages 244–282. Thomas Telford-ICE Virtual Library.

[61] Kirkconnell, C. (2002). Experimental investigation of a unique pulse tube expander design. In *Cryocoolers 10*, pages 239–247. Springer.

[62] Kirkley, D. (1962). Determination of the optimum configuration for a stirling engine. *Journal of Mechanical Engineering Science*, 4(3):204–212.

[63] Köhler, J. and Jonkers, C. (1954). Fundamentals of the gas refrigeration machine. *Philips Tech. Rev*, 16(3):69–78.

[64] Kohler, J. W. (1968). The stirling refrigeration cycle in cryogenic technology. *The Advancement of Science*, 25:261.

[65] Lean, S., Yuanyang, Z., Liansheng, L., and Pengcheng, S. (2009). Performance of a prototype stirling domestic refrigerator. *Applied Thermal Engineering*, 29(2-3):210–215.

[66] Li, R. and Grosu, L. (2017). Parameter effect analysis for a stirling cryocooler. *International Journal of Refrigeration*, 80:92–105.

[67] McFarlane, P. K. (2014). Mathematical model and experimental design of an air-filled alpha stirling refrigerator. Master's thesis, University of Notre Dame, Indiana.

[68] Mikulin, E., Tarasov, A., and Shkrebyonock, M. (1984). Low-temperature expansion pulse tubes. In *Advances in cryogenic engineering*, pages 629–637. Springer.

[69] Mungan, C. E. (2017). Coefficient of performance of stirling refrigerators. *European Journal of Physics*, 38(5):055101.

[70] Narasaki, K., Tsunematsu, S., Ootsuka, K., Kyoya, M., Matsumoto, T., Murakami, H., and Nakagawa, T. (2004). Development of two-stage stirling cooler for astro-f. In *AIP Conference Proceedings*, volume 710, pages 1428–1435. AIP.

[71] Narendra N Wadaskar and, D. S. and R.D.Askhedkar, D. (2017). Analysis of coefficient of performance & heat transfer coefficient on sterling cycle refrigeration system. *Int. Journal of Engineering Research and Application www.ijera.com*, 7(8):78–85.

[72] Ni, M., Shi, B., Xiao, G., Peng, H., Sultan, U., Wang, S., Luo, Z., and Cen, K. (2016). Improved simple analytical model and experimental study of a 100 w β-type stirling engine. *Applied Energy*, 169:768–787.

[73] Nie, W., He, J., and Du, J. (2009). Performance characteristic of a Stirling refrigeration cycle in micro/nano scale. *Physica A: Statistical Mechanics and its Applications*, 388(4):318–324.

[74] Oguz, E. and Ozkadi, F. (2000). An experimental study on the refrigeration capacity and thermal performance of free piston stirling coolers.

[75] Otaka, T., Ota, M., Murakami, K., and Sakamoto, M. (2002). Study of performance characteristics of a small stirling refrigerator. *Heat TransferAsian Research*, 31(5):344–361.

[76] Park, S., Hong, Y., Kim, H., Koh, D., Kim, J., Yu, B., and Lee, K. (2002). The effect of operating parameters in the stirling cryocooler. *Cryogenics*, 42(6-7):419–425.

[77] Park, S., Hong, Y., Kim, H., and Lee, K. (2003). An experimental study on the phase shift between piston and displacer in the stirling cryocooler. *Current Applied Physics*, 3(5):449–455.

[78] Penswick, L., Olan, R. W., Williford, I., Draney, S., and Buchholz, G. (2014). High-capacity and efficiency stirling cycle cryocooler. *International CryocoolerCon-ference, lnc., Boulder, CO*, pages 155–162.

[79] Potratz, S., Nellis, G., Maddocks, J., Kashani, A., Helvensteijn, B., Rhoads, G., and Flake, B. (2006). Development of a large-capacity, stirling-type, pulse-tube refrigerator. In *AIP Conference Proceedings*, volume 823, pages 3–10. AIP.

[80] Radebaugh, R. (2000). Development of the pulse tube refrigerator as an efficient and reliable cryocooler. *Proc. institute of refrigeration, London*.

[81] Radebaugh, R. (2009). Cryocoolers: the state of the art and recent developments. *Journal of Physics: Condensed Matter*, 21(16):164219.

[82] Radebaugh, R., Zimmerman, J., Smith, D. R., and Louie, B. (1986). A comparison of three types of pulse tube refrigerators: new methods for reaching 60k. In *Advances in Cryogenic Engineering*, pages 779–789. Springer.

[83] Ramos, J. A. A. (2015). *Thermodynamic Analysis of Stirling Engine Systems: Applications for Combined Heat and Power*. PhD thesis.

[84] Razani, A., Dodson, C., and Roberts, T. (2010). A model for exergy analysis and thermodynamic bounds of stirling refrigerators. *Cryogenics*, 50(4):231–238.

[85] Reader, G. T. and Hooper, C. (1983). *Stirling engines*. E. and F. Spon, New York, NY, USA.

[86] Ross Jr, R. G. and Boyle, R. F. (2006). An overview of nasa space cryocooler programs.

[87] Rout, S. K., Behura, A. K., Dalai, S., and Sahoo, R. K. (2017). Numerical analysis of a modified type pulse tube refrigerator. *Energy Procedia*, 109:456–462.

[88] Sauer, J. and Kuehl, H.-D. (2017). Numerical model for stirling cycle machines including a differential simulation of the appendix gap. *Applied Thermal Engineering*, 111:819–833.

[89] Shaowei, Z., Peiyi, W., and Zhongqi, C. (1990). Double inlet pulse tube refrigerators: an important improvement. *Cryogenics*, 30(6):514–520.

[90] Sowale, A., Kolios, A. J., Fidalgo, B., Somorin, T., Parker, A., Williams, L., Collins, M., McAdam, E., and Tyrrel, S. (2018). Thermodynamic analysis of a gamma type stirling engine in an energy recovery system. *Energy conversion and management*,

165:528–540.
[91] Swift, G. W. (1995). Thermoacoustic engines and refrigerators. *Physics today*, 48(7).
[92] Tang, K., Feng, Y., Jin, T., Jin, S., and Yang, R. (2017). Impact of gedeon streaming on the efficiency of a double-inlet pulse tube refrigerator. *Applied Thermal Engineering*, 111:445–454.
[93] Tekin, Y. and Ataer, O. E. (2010). Performance of v-type stirling-cycle refrigerator for different working fluids. *international journal of refrigeration*, 33(1):12–18.
[94] Tominaga, A. (1992). Phase controls for pulse-tube refrigerator of the third generation. *TEION KOGAKU (Journal of Cryogenics and Superconductivity Society of Japan)*, 27(2):146–151.
[95] Tward, E., Nguyen, T., Godden, J., and Toma, G. (2004). Miniature pulse tube cooler. In *AIP Conference Proceedings*, volume 710, pages 1326–1329. AIP.
[96] Tyagi, S., Kaushik, S., and Singhal, M. (2002). Parametric study of irreversible stirling and ericsson cryogenic refrigeration cycles. *Energy conversion and management*, 43(17):2297–2309.
[97] Tyagi, S., Lin, G., Kaushik, S., and Chen, J. (2004). Thermoeconomic optimization of an irreversible stirling cryogenic refrigerator cycle. *International journal of refrigeration*, 27(8):924–931.
[98] Urieli I., (2018) Stirling Cycle Machine Analysis.
[99] Urieli, I. and Berchowitz, D. M. (1984). *Stirling cycle engine analysis*. A. Hilger Bristol.
[100] Walker, G. (1980). *Stirling engines*. Oxford University Press, New York, NY.
[101] Walker, G., Weiss, M., Fauvel, R., and Reader, G. (1989). Microcomputer simulation of stirling cryocoolers. *Cryogenics*, 29(8):846–849.
[102] Wang, K., Dubey, S., Choo, F. H., and Duan, F. (2016a). Modelling of pulse tube refrigerators with inertance tube and mass-spring feedback mechanism. *Applied energy*, 171:172–183.
[103] Wang, K., Sanders, S. R., Dubey, S., Choo, F. H., and Duan, F. (2016b). Stirling cycle engines for recovering low and moderate temperature heat: A review. *Renewable and Sustainable Energy Reviews*, 62:89–108.
[104] Wang, X., Dai, W., Zhu, J., Chen, S., Li, H., and Luo, E. (2015). Design of a two-stage high-capacity stirling cryocooler operating below 30k. *Physics Procedia*, 67:518–523.
[105] Wang, X., Zhu, J., Chen, S., Dai, W., Li, K., Pang, X., Yu, G., and Luo, E. (2016c). Study on a high capacity two-stage free piston stirling cryocooler working around 30 k. *Cryogenics*, 80:193–198.
[106] Xu, Y., Cai, Y., Sun, D., Shen, Q., Zhao, X., Zhang, J., and Cheng, Z. (2014). Study on a high-power stirling cryocooler. In *International Cryocooler Conference*, volume 18.
[107] Xu, Y., Sun, D., Qiao, X., Yan, S., 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*, 83:78–84.
[108] Yang, L., Xun, Y., Thummes, G., and Liang, J. (2010). Single-stage high frequency coaxial pulse tube cryocooler with base temperature below 30 k. *Cryogenics*, 50(5):342–346.
[109] Yu, G., Dai, W., Qiu, J., Zhang, L., Li, X., Liu, B., Wu, Z., Xu, J., Luo, E., and Li, H. (2014). Initial test of a stirling cryocooler with a high cooling capacity. In *International Cryocooler Conference*, volume 18, pages 169–175.
[110] Yuan, J. and Maguire, J. (2005). Development of a single stage pulse tube refrigerator with linear compressor. In *Cryocoolers 13*, pages 157–163. Springer.
[111] Zhao, Y. and Dang, H. (2016). Cfd simulation of a miniature coaxial stirling-type pulse tube cryocooler operating at 128 hz. *Cryogenics*, 73:53–59.
[112] Zhou, Q., Chen, L., Zhu, X., Zhu, W., Zhou, Y., and Wang, J. (2015). Development of a high-frequency coaxial multi-bypass pulse tube refrigerator below 14 k. *Cryogenics*, 67:28–30.
[113] Zhu, S. and Matsubara, Y. (2004). Numerical method of inertance tube pulse tube refrigerator. *Cryogenics*, 44(9):649–660.
[114] Zhu, S. and Nogawa, M. (2010). Pulse tube stirling machine with warm gas-driven displacer. *Cryogenics*, 50(5):320–330.
[115] Zia, J. (2005). A commercial pulse tube cryocooler with 200 w refrigeration at 80 k. In *Cryocoolers 13*, pages 165–171. Springer.

Declaration of interests

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.