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E. Oguz
Arcelik A.S.

F. Ozkadi
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AN EXPERIMENTAL STUDY ON THE REFRIGERATION CAPACITY AND THERMAL PERFORMANCE OF FREE PISTON STIRLING COOLERS

Emre Oguz, Fatih Ozkadi
Arcelik A.S.
Research & Development Center
81719, Tuzla, Istanbul TURKEY

ABSTRACT

The refrigeration capacity and thermal performance characteristics of two prototype free piston Stirling coolers were investigated experimentally. Thermal load was applied to the cold head of the Stirling cooler by two resistance heaters in a so-called adiabatic box, and steady-state characteristics of the coolers were evaluated. For a certain input voltage, different tests were carried out to determine the variation of the performance with the cold head temperature, hence the cold head temperature of the coolers varied from -40°C to 0°C. Since the refrigeration capacity of a free piston Stirling cooler changes with the applied voltage, tests were repeated for three different input voltages for one of the coolers. The capacity data were simply correlated in terms of the cold head temperature to predict the change of the refrigeration capacity with the applied voltage for constant warm and cold head temperatures.

INTRODUCTION

Since the novel invention of Robert Stirling in 1816, the Stirling cycle machines have always been of great importance for the researchers and engineers to generate electrical or mechanical power more efficiently or to reduce the energy consumption of the refrigeration devices. The commercial Stirling cycle coolers have been used in cryogenic applications since 1940s but it has not been possible yet to utilise the cycle for domestic refrigeration applications. However, with the invention of the free piston technology in the early 1960s by William Beale [1], the Stirling cycle now seems to be a challenging alternative for domestic refrigeration.

Theoretical Stirling cycle consists of two isothermal and two isovolumetric heat transfer processes where the isovolumetric heat transfer takes place in a regenerative manner. While rejecting thermal energy to the environment during the isothermal compression process, the gas absorbs thermal energy from the environment during the isothermal expansion. Therefore, a Stirling cycle cooler may be determined by a high and a low temperature which are generally named warm and cold head temperatures. Extensive literature exists both on the thermodynamics and gas dynamics of Stirling cycle and the performance analysis of Stirling cycle machines [2, 3, 4].

A detailed study on the refrigeration capacity and thermal performance of free piston Stirling coolers was presented by Mc Donald et al. [5]. The Stirling coolers had been designed for a particular application to cool an insulated volume of 0.3 m^3 and a thermal lift of 124 W at 23°C and 4°C ambient and cabinet temperatures respectively. The performance map presented, covers the refrigeration capacity and input power data of the cooler for a cold head temperature range of -10°C to -90°C and a piston amplitude range of 4.5 to 6.5 mm where the warm head temperature had been assumed to be constant at 45°C.

An interesting study on the free piston coolers and Stirling cycle cooled domestic refrigerators was presented by Berchowitz [6]. The coefficient of performance values of a prototype Stirling cooler for three different thermal lifts, namely 20, 40 and 70 W were given and the COP value was reported to be approximately 3.0 for 0°C cold and 30°C warm head temperatures. Additional information on the energy consumption predictions of domestic refrigerators which uses advanced insulation technologies like Vacuum Insulation Panels or Vacuum Insulation Components and operating with a free piston Stirling cooler, may be found in this study.

In a recent study by Berchowitz et alia, the construction and test results of a Stirling cycle cooled portable cool box are given in detail [7]. A 40 W capacity free piston cooler had been used to cool a 40 liter cabinet, providing the thermal energy exchange by a system called thermosyphon employing carbondioxide as the heat transfer media. Presented experimental results are stated to be much better than Peltier or small vapour compression systems.

Another study including thermosyphon installation as the heat transfer system was presented by Green et al. [8]. A Stirling cycle cooler built by Oxford University had been integrated into a freezer cabinet and the layout of the cabinet had been modified accordingly. The fluids used in the thermosyphon system were water and isobutane for the warm and cold circuits respectively. Although the energy consumption of the Stirling cycle cooled freezer is reported to be 17% less than the original equipment, the authors state that the Stirling cycle freezer would consume 12% less energy than the mechanical compression system on a like for like basis.

EXPERIMENTAL SETUP AND PROCEDURE

Free Piston Stirling Coolers

Free piston Stirling coolers used in the current study are the first model prototypes of Global Cooling. The cooler may be defined as a pressure vessel which operates by shuttling approximately 1 gram of Helium back and forth by the combined movements of two parts, namely the piston and the displacer. While the piston that compresses the gas is driven by a linear motor, the displacer is moved by the pressure difference.

Since the surfaces enclosing the compression and expansion spaces are relatively small and the temperature differential between the gas and the heat sink / source must be as small as possible to obtain higher COP values, additional heat exchangers are supplied on the cold and warm heads of the cooler. Hence, it is possible to circulate a secondary media such as water to increase the heat transfer effectiveness. The Stirling coolers used in this study are 50 Hz AC units. Since the piston amplitude is directly proportional to the RMS of the drive voltage, the refrigeration capacity of the coolers may easily be adjusted for different heat loads.

Test Riq

A schematic diagram of the so-called adibatic box and the position of the free piston Stirling cooler is given in Figure 1. The cold surface of the cooler is enclosed in a box which was insulated to prevent heat-leak / heatgain to or from the environment. Conventional XPS (Extruded Polystyrene) with a thermal conductivity of 28 mW/m.K was used as the inner layer of insulation. At the outer layer, vacuum insulation panels with a thermal conductivity of 6 mW/m.K were placed to minimise the heat transfer. It was theoretically calculated that the vacuum insulation panels (VIPs) had decreased the heat transfer rate approximately 70% for the same operating conditions. A plastic layer was used to protect the VIPs from possible damages and finally the insulation was covered with a metal case. Two cylindrical resistance heaters were placed on a copper disc and the disc was mounted on the cold surface of the Stirling cooler. The heaters used were rated 100 W at 220 V and controlled by variacs.

For the warm side of the Stirling cooler, a hydraulic circuit was provided. Water was pumped from a reservoir first through the heat exchanger on the cooler to cool the warm end of the cooler and then through another heat exchanger to reject thermal energy to the ambient. The second heat exchanger was provided with a fan to increase the heat transfer rate from the water to the ambient air. A valve was also provided to control the flow of water.

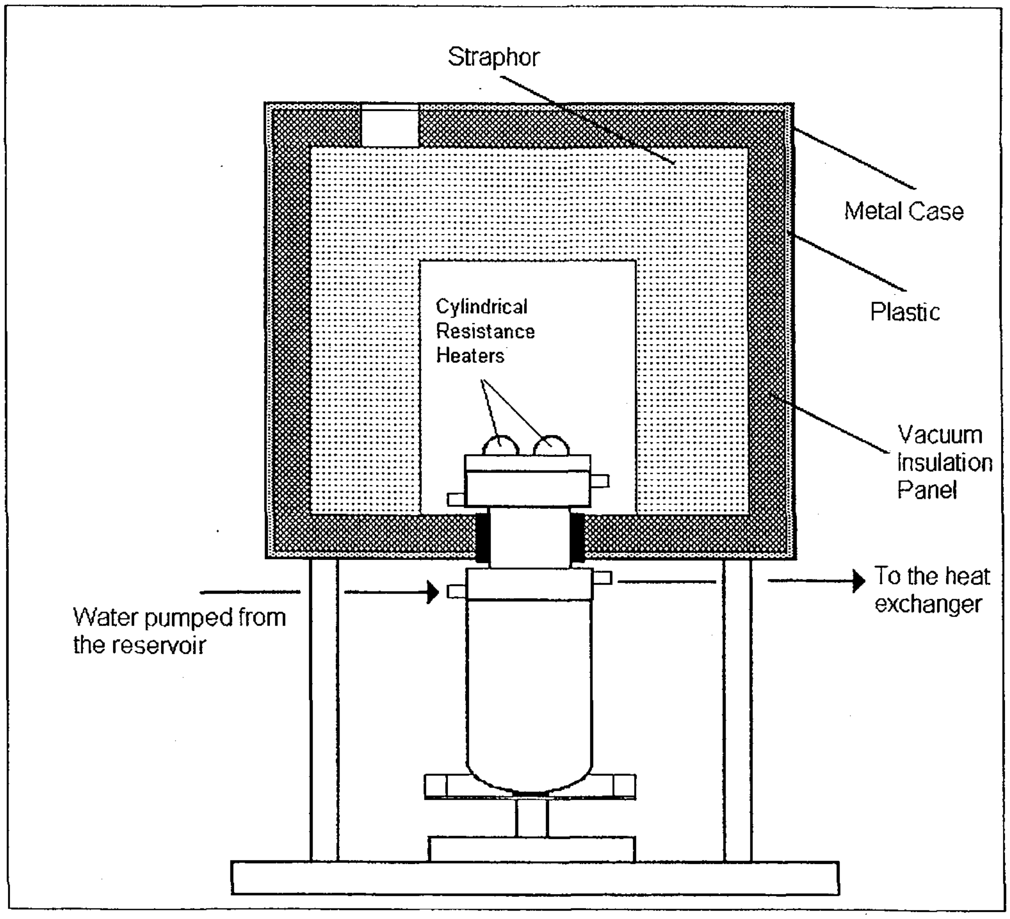

Figure 1. Schematic diagram of the so-called adiabatic box and the FPSC.

Instrumentation and Experimental Procedure

The parameters measured during the tests were the cold and warm head temperatures of the Stirling cooler, the temperatures of the ambient air and the air inside the adiabatic enclosure, the temperatures of the inner and outer surfaces of the so-called adiabatic box. For the temperature measurements type T thermocouples and a data logger were used with an accuracy of ±0.3°C. Electrical parameters measured during the experiments were the voltage input, current and the power consumption of the cooler and the power consumptions of the resistance heaters. The wattmeter used had an accuracy of ±0.1%, ±0.2% and ±1% for voltage, current and power readings respectively. Uninterruptable power supply was used for all of the tests.

Three thermocouples were placed at the inner surfaces and six thermocouples were placed at the outer surfaces of the enclosure to predict the heat transfer rate theoretically. The maximum rate of heat transfer was predicted to be 2.5 W without the vacuum insulation panels. After the VIPs had been placed to enhance the effectiveness of the insulation, the heat transfer for the same operating conditions was predicted to be less than 1.5 W.

At the beginning of all of the tests, the voltage input of the cooler being tested and the thermal load applied by the heaters were adjusted to certain levels. Since it would take more time to reach steady-state at certain cold head temperatures, the thermal load was chosen as the independent parameter. After a certain time interval the system was observed to reach steady-state and the coolers were run for approximately one more hour at steady-state. Cooldown and steady-state periods of a typical test are given in Figure 2. For clarity, the temperatures of the outer and inner surfaces are not given here.

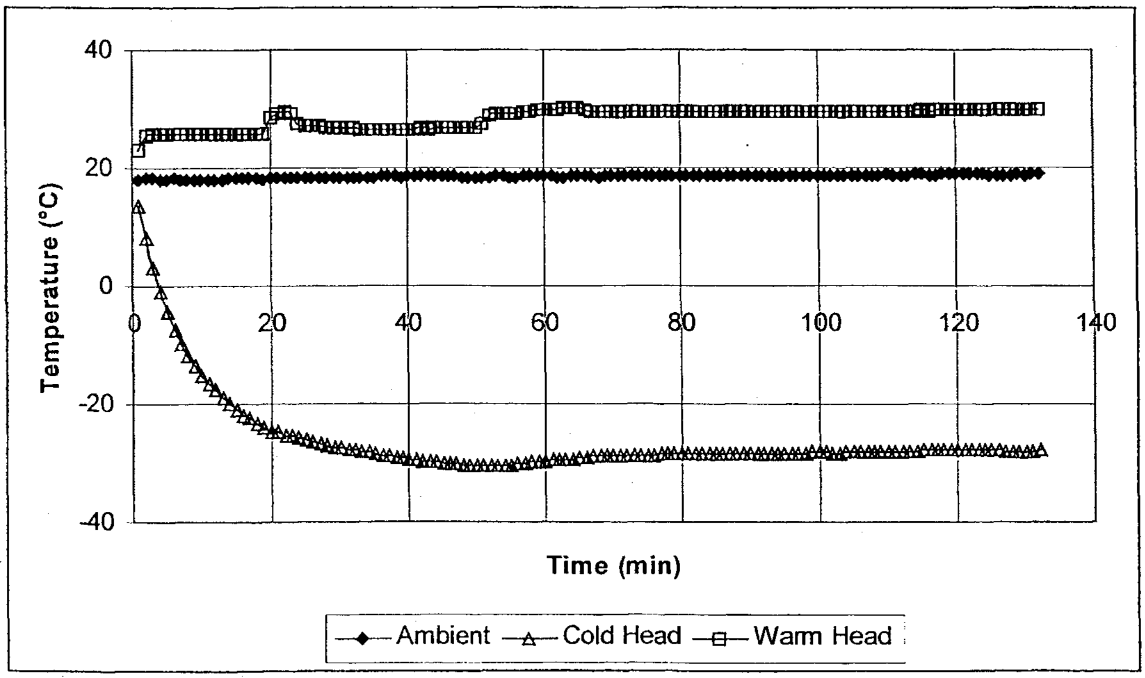

Figure 2. Cooldown and steady-state periods of a typical test.

After a test was completed, the steady-state interval was determined using the fluctuations in the temperatures recorded, and the average values of measurement parameters were calculated in the same interval. Except two of the 15 tests conducted on two different Stirling coolers, the variation of the cold head temperature during the specified steady-state intervals was well within ±0.5°C, during the two tests the variation reaching ±0.8°C. The specified steady-state periods vary between 40 to 110 minutes, most of the periods lasting for more than an hour.

Since a simple calorimetric method is used, for a certain input voltage and cold and warm head temperatures, the refrigeration capacity of the cooler may be determined from the steady-state form of the first law of thermodynamics applied to the control mass in the adiabatic enclosure , which may be written as follows

$$\dot{Q}_{load} - \dot{Q}_{cap} - \dot{Q}_{leak} = 0 \quad (1)$$

where subscripts load, cap and leak refer to the thermal load applied by the heaters, the refrigeration capacity of the cooler and the heat leak to the environment respectively. Accordingly, the coefficient of performance of the cooler may be calculated as

$$COP = \frac{\dot{Q}_{load} - \dot{Q}_{leak}}{\dot{W}_{cooler}} \quad (2)$$

where $\dot{W}_{cooler}$ represents the power consumption of the cooler.

TEST RESULTS

The refrigeration capacity of the two coolers tested are given in Figure 3. Cooler #1 was tested for 100, 110 and 120 V and cooler #2 was tested for approximately 80 V. Since all of the data points are plotted on the same graphic, a common warm head temperature must be stated for the chart to be meaningful. However, the warm head temperature during different tests varied between 28 to 31°C and hence, the common warm head temperature may be declared as 30°C.

Referring to Figure 3, it must be stated that curve fitting was not applied to the data of cooler #2 for clarity. It is easily seen that for constant warm head temperature and input voltage the refrigeration capacity may easily be represented as a linear function of the cold head temperature. Evaluating this feature, linear functions are obtained for cooler #1 and the change of the refrigeration capacity with the input voltage for constant cold head temperatures is presented in Figure 4.

Referring to Figure 4, it must be remembered that only the data between 100 and 120 V are experimental and could be represented in a linear fashion. Therefore, considering the refrigeration capacities at input voltages higher than 140 V, one may expect high discrepancies because of the complicated processes that occur in a free piston Stirling cooler.

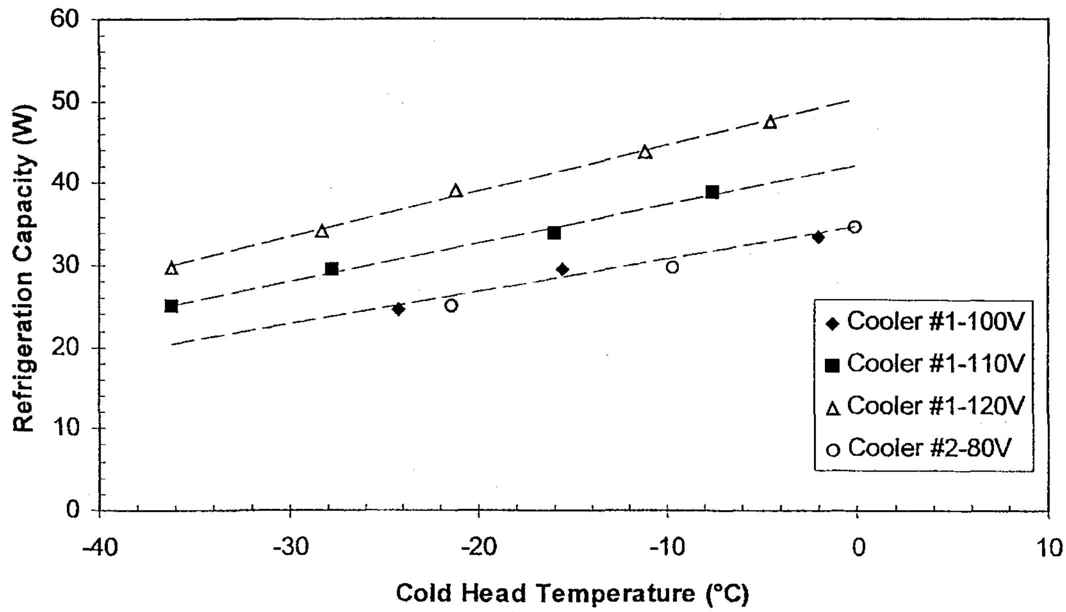

Figure 3. Refrigeration capacity of coolers #1 and #2 (Warm head temperature:30°C)

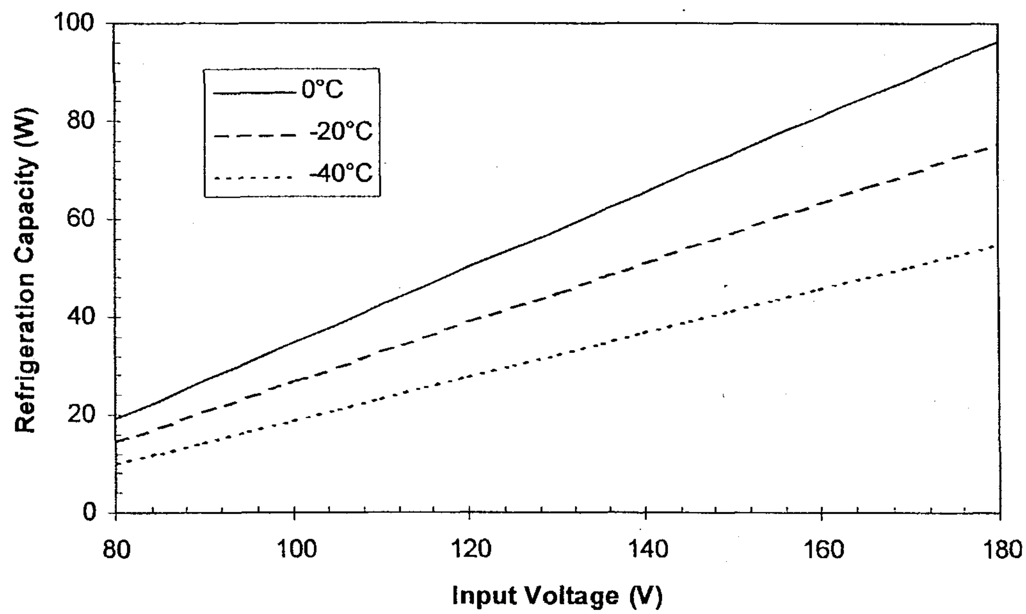

Figure 4. Refrigeration capacity (predicted) as a function of input voltage (Cooler #1).

The coefficient of performance values for both of the coolers are presented in Figure 5. There seems to be some descripancy as high as 25% between the coolers. This difference is thought to be caused by the thermocouple soldering process at the very beginning of the experiments which could overheat the internal components of the cooler #1. On the other hand, COP values of 2.2 or more could be reached with this cooler with a cold head temperature of –2°C and a warm head temperature of 30°C.

In Figure 6, the temperature-independant COP values are presented as the fraction of the Carnot COP. Referring to Figure 6, it must be noted that the Carnot COP values are based on the cold and warm head temperatures.

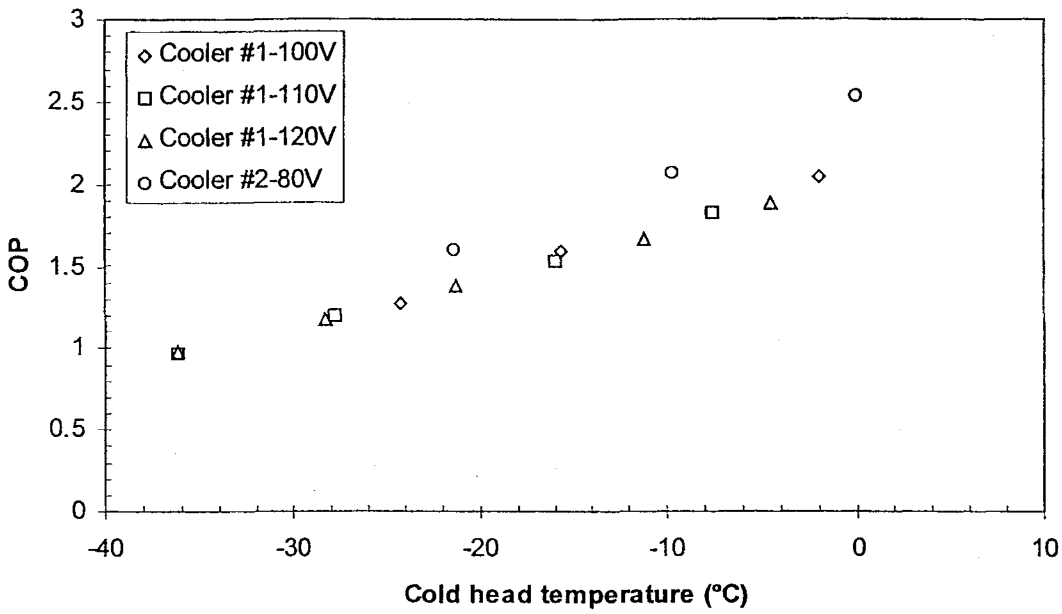

Figure 5. COP values for coolers #1 and #2 (Warm head temperature: 30°C)

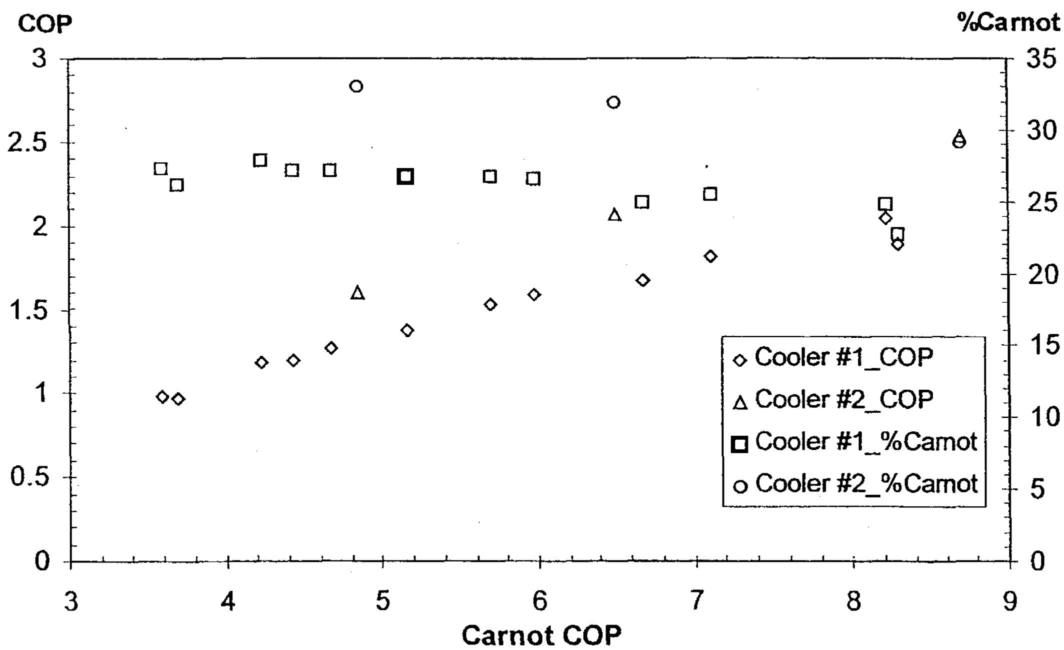

Figure 6. COP and Carnot Fraction for coolers #1 and #2.

CONCLUSION

A method for measuring the refrigeration capacities and thermal performances of free piston Stirling coolers is described and the test results for two different coolers are presented. According to the results of the current study, it may be concluded that :

- For a warm head temperature of 30°C, the refrigeration capacities of free piston Stirling coolers tested may be modulated from 10 to 100W by simply varying the input voltage. Therefore, it can be concluded that Stirling coolers operate in a similar manner to the variable capacity compressors (VCCs).
- For 0°C / 30°C cold and warm head temperatures and an input voltage of approximately 80V, COP value of 2.5 is measured for the cooler. This result has been encouraging to integrate the coolers to refrigerators (fully freshfood compartment).
- Free piston Stirling technology is a challenging alternative to be used instead of conventional compressors and the experimental and theoretical studies on this subject should continue in the future.

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