CO_2 Based Free-Piston Stirling Heat Pump with Ground Source Heat Reservoir

David BERCHOWITZ[*1], Martien JANSSEN and Marcel van BEEK [*2] and Andreas HAGEDORN [*3]

[*1] *Global Cooling BV, Lagedijk 22, 5705 BZ Helmond, THE NETHERLANDS,* berchowitz@globalcooling.com
[*2] *Re/genT, Lagedijk 22, 5705 BZ Helmond, THE NETHERLANDS,* martien.janssen@re-gent.nl
[*3] *Tracto-Technik, Reiherstrasse 2, 57368 Lennestadt, GERMANY,* andreas.hagedorn@tracto-technik.de

Keywords: *Stirling, Heat Pump, Residential, Carbon Dioxide, Ground Source*

Abstract

The Free-Piston Stirling Engine (FPSE) concept is extended to directly drive a heat pump compressor by using a common piston structure. The working gas is carbon dioxide, which is allowed to freely mix between the engine and the transcritical heat pump cycle. The state of the working gas within the engine remains gaseous at all times. There is an engine efficiency compromise of about 10% using carbon dioxide compared to the tyically used helium. However, the resulting mechanical simplicity by not having to provide a hard separation between the two working fluids easily makes up for the efficiency compromise. The machine is very compact and is partially balanced by employing opposed pistons. Power matching is facilitated by linear alternator / motors. Since the export of electrical power is not the primary purpose, these components do not have to carry the full power output. Calculations are presented assuming a recuperative burner and a residential hydronic heating system. High primary energy ratios (PERs) are possible approaching a factor of 1.6. Sub components of the system are discussed, i.e., burner, engine, compressor and heat exchangers. Details such as leakage, gas bearing and other losses are evaluated in order to obtain a realistic energy accounting. An overall system energy flow diagram and some constructional details are included to explain the basic functioning of the machine. Since heat is captured from the engine, the ground source heat exchanger is smaller and less expensive than an equivalent electrical heat pump. A representative northern European location in Germany has been chosen where high winter heating loads can be expected.

Nomenclature

A = flow area [m^2]
A_w = wetted area [m^2]
C_L = laminar flow friction coefficient (f Re)
d_h = hydraulic diameter ($4A/z$) [m]
f = Fanning friction factor
k = gas thermal conductivity [W/m K]
$\dot{m}$ = heat pump mass flow [kg/s]
Nu = Nusselt number
p = pressure [Pa]
P_{in} = input mechanical power [W]
$Q_{gascooler}$ = heat from gas cooler [W]
V_D = dead volume [m^3]
V_h = heat exchanger volume [m^3]
V_{swept} = swept volume of compressor [m^3]
$|V|$ = volume amplitude [m^3]
Δh_i = enthalpy change due to isentropic compression [J/kg]
Δh_g = enthalpy change in gas cooler [J/kg]
ΔT_{wg} = temperature differential, gas to wall [K]
ε_b , ε_d , ε_l = fractional losses due to bearings, dead volume and leakage
μ = dynamic viscosity [Pa s]
ρ_{suc} = suction gas density [kg/m^3]
ω = frequency [rad/s]

1. Introduction

Fuel-driven heat pumps are known to offer the highest potential for heating efficiency compared to any other system. However, building a unit that would be cost effective for residential applications has long eluded industry. An ideal unit would be long-lived, reliable, cost-effective, quiet and have a high primary efficiency. The only technology reasonably likely to meet these requirements is the free-piston Stirling engine driving a conventional Rankine or perhaps a transcritical heat pump. The General Electric Corporation (GE) first worked on fuelled heat pumps using this principle in 1975 [1]. Later, during the 1980s, Mechanical Technology Inc. (MTI) attempted to develop a different version of a free-piston Stirling heat pump [2, 3]. Later, during the 1990s, Sunpower, Inc. became involved with Stirling driven heat pump development [4]. The MTI unit achieved significant milestones with a heat delivery of 17.5 kW at a Primary Energy Ratio (PER) of 1.5 at an ambient temperature of 8.3°C, including pumps and accessories. The combined engine burner efficiency was 25%. All these machines used separate working fluids for the engine (helium) and the heat pump (refrigerant R-22). This necessitated complicated mechanisms to ensure that the working fluids would remain separated from each other. The GE system employed an inertia driven compressor where the refrigerant was carried in hollow springs to the compressor. MTI opted for a different approach where the engine piston was replaced by an oil-supported diaphragm, which in turn, drove the compressor through a hydraulic coupling. Sunpower, on the other hand, used a linear magnetic coupling between the engine piston and the heat pump. While the effectiveness of the technology had been demonstrated, all these machines suffered from serious technical difficulties stemming from their complicated means of coupling the engine to the heat pump. In recent years, Global Cooling (GC) has been working on a technique that avoids the issue of working gas separation by using a common working fluid for both the heat pump and the Stirling engine [5, 6]. When CO_2 is used as a working fluid, it is able to satisfy all the basic thermodynamic requirements for the heat pump and the Stirling cycle. Therefore no hard seal is required between the heat pump and the engine with the consequence of much reduced complexity and a more compact device. Figure 1 shows the relative sizes of the MTI and the GC machines for roughly the same engine power and efficiency levels.

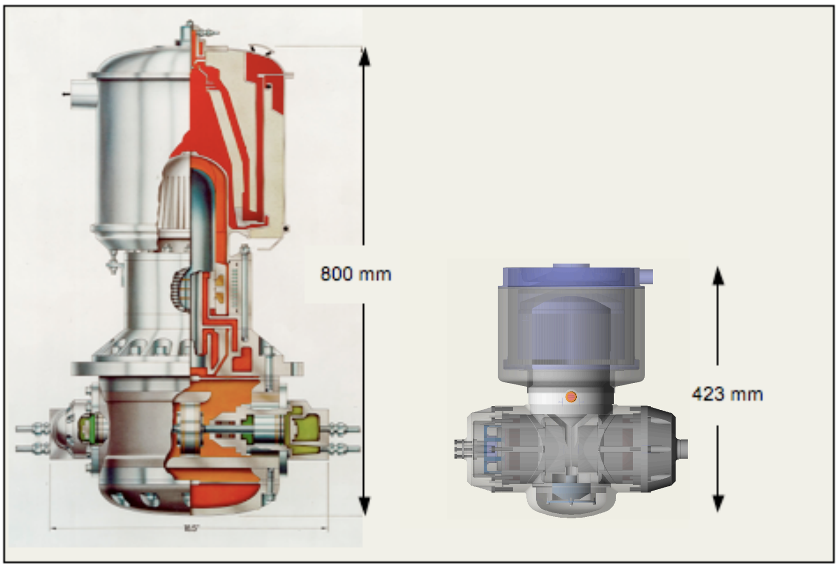


Figure 1 MTI and GC Free-Piston Stirling Heat Pumps at Similar Engine Power Levels

2. The Engine

Carbon Dioxide is not typically used in Stirling engines. To give a preliminary idea of the comparison between CO_2 and helium, consider that in order to obtain similar performance between engines of similar power using different working gases, the total engine irreversibilities should also be similar. Irreversibilities are functions of temperature gradients and viscous dissipation. Therefore an equivalent statement of similar performance would be: for given operating temperatures; if the temperature gradients and viscous dissipation is similar in engines with different working gases but similar powers, then the overall performances should also be similar. Recognising that the flow in Stirling engines is typically laminar with low Womersley numbers, the following method first presented by Berchowitz and Richter [7], may be employed:

Assuming that the flow in the heat exchanger is dictated by the adjacent working space, Laminar flow convective heat transfer and viscous dissipation may be estimated by:

$$\left\langle \dot{Q} \right\rangle = \mathrm{Nu}\, k\, A_w\, \Delta T_{wg} / d_h \qquad \text{[W]} \qquad 1$$

$$\left\langle \Phi \right\rangle \approx \mu C_L\, V_h \left(\frac{\omega |V|}{d_h\, A} \right)^2 \qquad \text{[W]} \qquad 2$$

From Eqs. (1) and (2), and noting the definition of d_h, the heat transfer per unit viscous dissipation is then:

$$\frac{\left\langle \dot{Q} \right\rangle}{\left\langle \Phi \right\rangle} = 4 \frac{\mathrm{Nu}}{C_L} \frac{k}{\mu} \left(\frac{A}{\omega |V|} \right)^2 \Delta T_{wg} \qquad 3$$

For the monocoque designs considered, the ratio Nu / C_L is constant and it is clear that the ratio k / μ will have a strong effect on the design of the heat exchangers. The thermal conductivity of CO_2 is about one tenth that of helium while viscosity is similar.

Assuming similar temperatures, the developed power is proportional as follows:

$$P \propto \omega \langle p \rangle \frac{|V|^2}{V_D} \qquad \text{[W]} \qquad 4$$

From Eqs. (3) and (4) and assuming that the dead volume, V_D is proportional to the flow area A (length is held roughly constant), the following proportionality is obtained:

$$\langle p \rangle A \propto \omega \frac{\mu}{k} \frac{\mathrm{P}}{\Delta T_{wg}} \frac{\langle \dot{Q} \rangle}{\langle \Phi \rangle} \qquad 5$$

Suggesting that, for a given efficiency and power, the product of charge pressure and flow area scales according to $\omega \mu / k$. To simplify the discussion, ω will be held constant. Therefore, for similar $\langle \dot{Q} \rangle / \langle \Phi \rangle$, P, ΔT_{wg} and ω, the following ratios are obtained:

$$\left[\langle p \rangle A\right]_{\mathrm{He}} / \left[\langle p \rangle A\right]_{\mathrm{CO_2}} = 0.162 \text{ at 300K and 0.219 at 1000K} \qquad 6$$

Indicating that the product of charge pressure and flow area for CO_2 engines is much greater than that for equivalently performing helium engines with similar passage geometry. Pressure is not that easily exploited because the engine operates at roughly the suction pressure of the heat pump at about 30 to 35bar. In general, therefore, the CO_2 engine will have a greater diameter to accommodate the larger flow area made up of closely spaced passages. Of course this is a very simple argument and issues such as flow-induced compressibility, regenerator performance, seal leakage and gas hysteresis have not been examined. Nevertheless, it can be reasonably concluded that a CO_2 engine will have some combination of larger flow areas, higher pressures and larger volume amplitudes compared to an equivalently performing helium engine. In the design of the engine used here, some compromise on efficiency has been taken in order to keep the machine compact. Figure 2 shows the expected performance for the engine as determined by simulation at various pressures as may be encountered during operation. High piston amplitudes at high pressures are characterized by a drop-off in efficiency. Power matching will be achieved by piston amplitude modulation, which is an effective means for adjusting the power in free-piston machines.

While a description of the burner is beyond the scope of this paper, it is noted that the efficiency is expected to be at least 85% with a turndown ratio of 10, similar to what has been achieved by the MTI and GE programs [8]. Exhaust heat recovery will recover about 90% of the lost heat.

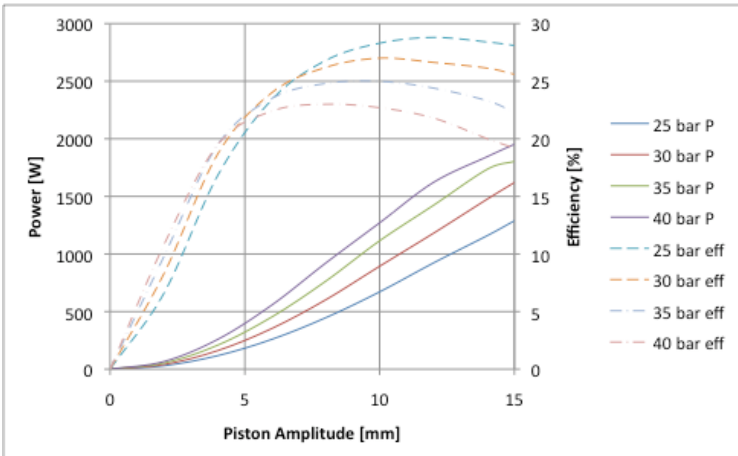


Figure 2 Engine Power and Efficiency versus Piston Amplitude for Various Pressures

3. The Heat Pump

Figure 3 shows the heat pump system as may be used for floor heating. Alternative systems employing radiators and / or domestic hot water and / or space cooling are all possible but outside the scope of this paper. From the figure, it is seen that the working fluid (CO_2) evaporates in the evaporator thereby extracting heat from the brine in the ground loop and, ultimately, from the ground. After the evaporator and prior to entering the compressor, the working fluid passes through the internal heat exchanger (IHE) and exchanges heat with the flow from the gas cooler. It is then compressed from suction pressure (evaporation pressure) to discharge pressure and its temperature rises. After compression, the CO_2 passes through the gas cooler, where heat is rejected to the central heating system and then through the IHE were it is further cooled by gas leaving the evaporator. Finally, the CO_2 is expanded and returns to the evaporator. The Stirling engine rejects its heat to the central heating system as well.

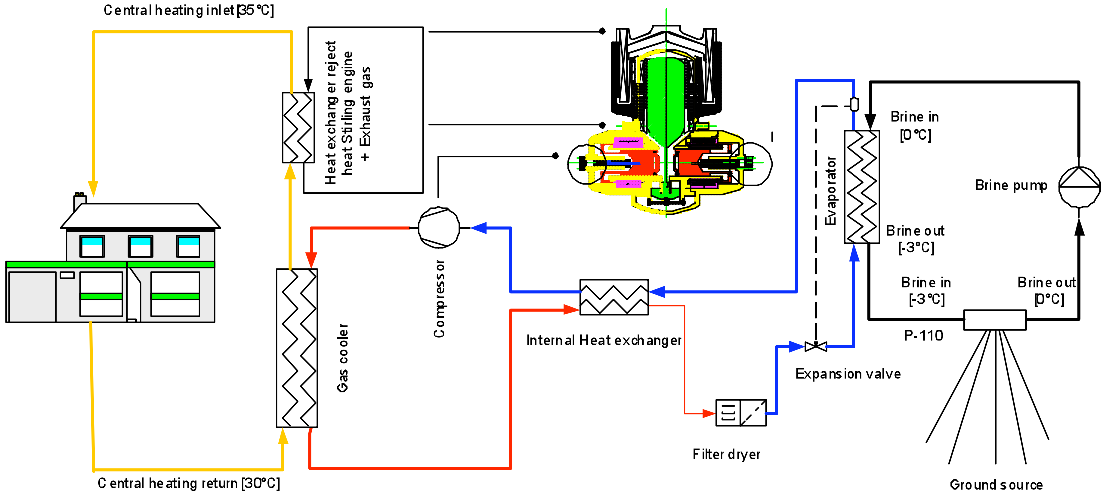


Figure 3 Heat Pump System in Floor Heating Mode (System Temperatures at Rating Condition, EN 14511-2)

4. Compressor

The compressor pistons are directly coupled to the Stirling engine pistons and gas bearings are used throughout. Two indicators describe the performance of the compression process, namely the volumetric (Eq. 7) and isentropic (Eq. 8) efficiencies.

$$\eta_v = \frac{\dot{m}}{\dot{m}_{ideal}} = 2\pi \frac{\dot{m}}{\rho_{suc} V_{swept} \omega} = 1 - (\varepsilon_l + \varepsilon_d + \varepsilon_b) \qquad 7$$

$$\eta_i = \frac{P_{th}}{P_{in}} = \frac{\Delta h_i (1 - \varepsilon_d) \dot{m}_{ideal}}{P_{in}} \qquad 8$$

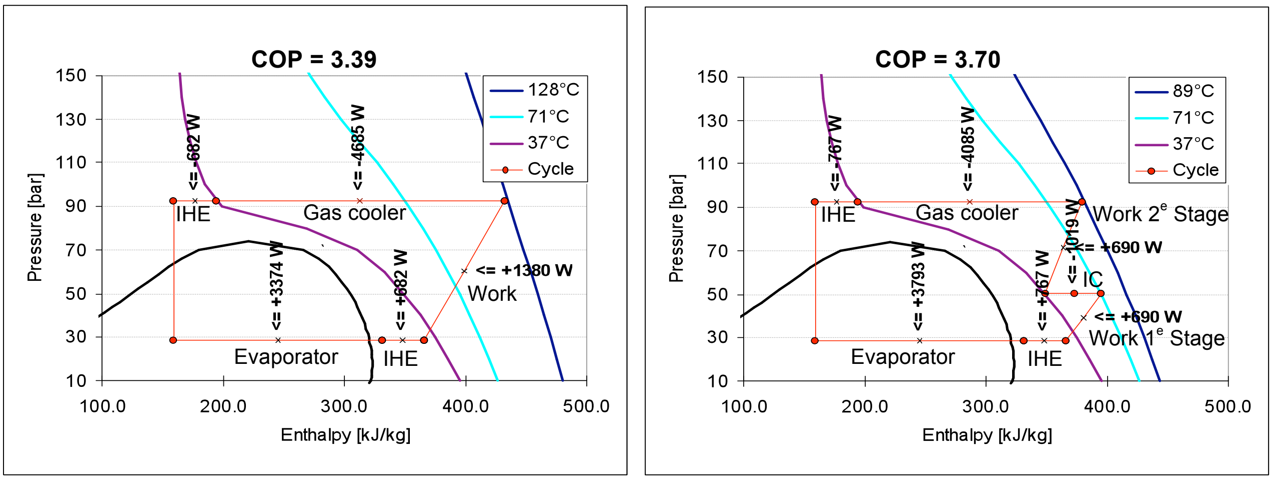
The isentropic losses, manifested in P_{in}, result mainly from heat loss and viscous effects during compression and are in addition to pressure losses across valves and mufflers. It is assumed that these losses can be kept to within 5% of the theoretical compression power P_{th} thereby resulting in η_i = 0.95. The volumetric efficiency is directly related to leakage, dead volume and bearing losses. Calculations including CFD modeling (Fluent 6.2.16) show that it is possible to design an adequate piston bearing configuration in which the total leakage and bearing consumption ($\varepsilon_b + \varepsilon_l$) is within 5% of the mass flow at maximum capacity. The calculations were performed with a piston length of 100 mm with a clearance of 18μm +/- 2μm and a top dead center clearance of 0.5mm. At the design conditions (EN 14511-2) this corresponds to a volumetric loss of ε_d = 5.6%. Therefore a total volumetric efficiency of η_v = 0.90 is estimated for the compressor.

5. Heat pump model

The thermodynamic model for the heat pump uses refrigerant properties from Refprop 6.01. The heat resistances of the gas cooler and evaporator are included by assuming a temperature differential of 7K between the exit temperature of the working fluid and the inlet temperature of respectively the water and brine. Based on experience, the suction gas heat exchanger is taken to have an effectiveness of 75%. The coefficient of performance (COP) of the heat pump is defined by Eq. 9.

$$COP = \frac{Q_{gascooler}}{P_{in}} = \frac{\eta_v \Delta h_{gc}}{\eta_i \Delta h_i (1 - \varepsilon_d)} \qquad 9$$

Because the heat pump has two opposed compressors, two options are possible, namely single or two-stage operation. The compressors are used in parallel for single-stage operation and in series with intercooling for two-stage operation. Figures 3 and 4 show the pressure enthalpy diagrams for these options using temperatures as specified in EN 14511-2. Heat rejected to (+) or the heat absorbed from (-) is shown for each heat exchanger. The required compression work, the characteristic isotherms (gas cooler exit temperature and discharge temperature) and the resulting system COP are also shown. Optimizing the discharge pressure gives a COP of 3.7 for the two-stage and 3.39 for the single-stage, i.e., 8.4% lower. In the two-stage, heat is also extracted from the intercooler (IC).



Figures 3 and 4 Single and Two-Stage Compression Configurations with Discharge Pressure Optimized for Maximum COP at EN 14511-2 Conditions. (IHE = Internal Heat Exchanger, IC = Intercooler)

6. The Ground Source Heat Exchanger (GSHE)

Heat extraction is determined by parameters such as: type of grout, quality of the grouting operation, geometry of borehole and heat exchanger, local climate conditions, as well as the ground thermal conductivity and specific heat capacity. The latter are correlated to the ground natural content of water and silicon dioxide (quartz). Generally, the higher the underground water and quartz content, the shorter the length of borehole heat exchanger (Table 1). The average atmospheric heat flux due to insolation and rainwater seepage, i.e., the energy that flows from the surface into the ground, can be traced down to 20m below the surface. This flux varies between approximately 170W/m² (winter) and 650W/m² (summer) in Northern Europe. The geothermal heat flux, on the other hand, i.e., the heat that flows in the opposite direction (from the inner earth to the surface) is approximately 0.065W/m². The input of atmospheric energy is at least 2600 times higher than the amount of energy provided by the inner earth.

Table 1: Length of GSHE Depends on Heat Demand and Ground Characterisation (Quartz Content)

Ground Characterization	GSHE Length [m] for Different Heat Demands			
	Case A 3.1 kW	Case B 5 kW	Case C 10 kW	Case D 15 kW
Clay (moist, VDI 4640: 40W/m)	34	48	109	163
Silt (moist, VDI 4640: 35W/m)	39	54	124	187
Sand (dry –moist, VDI 4640: 30W/m)	45	63	145	218
Sand (saturated, VDI 4640: 65W/m)	21	29	67	100

(Note: Water Saturated Sand and Rock are similar in Specific Heat Extraction)

For an electric heat pump, the heat extracted from the ground is calculated from the COP. For the Stirling heat pump, a substantial fraction of the heat is provided from the engine and therefore its efficiency together with the burner efficiency and the coupled heat pump COP are needed to calculate the amount of heat extracted from the ground. Table 2 shows the relative fractions depending on the system. In order to calculate the needed length of GSHE, the following further assumptions are necessary:

- There is neither heating nor cooling from June to August.
- According to VDI 4640, the operating time of the heat pump in case of heating (excluding domestic hot water) can be assumed to be 1800 hrs.
- The house is located in the city of Berlin where the ground is characterized by glacial deposits (predominantly water saturated sand). The ground thermal conductivity is 2.4 W/(m K) with a heat capacity of 2.5 MJ/(m³ K) and a geothermal heat flux of 0.070 W/m². Berlin has a continental climate typically characterized by hard winters and warm summers thus reasonably covering the range of temperatures in Western Europe. The average ground surface temperature is 9.0 °C.
- The model was also run for the city of Paris (Atlantic climate, average ground surface temperature, 12.1 °C) to demonstrate the influence of the climate.
- The mean fluid temperature shall not drop below 0°C after 50 years of operation (the model starts in January).
- The ground source heat exchanger consists of a double-U pipe PE DN 32 PN 10 filled with 25% monoethylene-glycol – water mixture (brine).

Simulating this model with the software package EED 3.1.4 yields the results listed in Table 1.Estimated costs of drilling and installing a GSHE for Berlin is approximately 70 € per m. For a system designed to deliver 9.7kW, the overall investment in the ground source would then be about 8,700 € for the electrical heat pump and about 4,600 € in case of the two-stage Stirling heat pump. The Stirling heat pump therefore represents a cost saving of approximately 47% on the total cost of the installation of the GSHE. For warmer regions, an increase of 1K in ground surface temperature represents an additional cost saving of approximately 7% to 8%.

Table 2: GSHE Simulation Results

System	Heat delivered to the home [kW]	Heat extracted from ground [kW]	Total length of GSHE in a vertical borehole [m]	
			Berlin	Paris
Electrical heat pump (COP = 4)	9.7	7.275	124.26	97.05
Electrical heat pump (COP = 4)	9.3	6.975	119.40	93.09
Two-stage Stirling heat pump (PER = 1.61)	9.7	3.793	66.25	50.27
Single-stage Stirling heat pump (PER = 1.55)	9.3	3.374	59.13	44.69

The investment costs for the installation of a GSHE are highly dependent on the calculation methods provided by national regulations and norms. The most important international regulation is the VDI 4640 [9]. This regulation assists the driller in calculating the necessary total length of the GSHE by providing the specific heat extraction value in W/m for the most common types of rock and soil. The driller's calculation is kept basic: the heat load is divided by the specific heat extraction of the ground and the outcome is the total length of the GSHE. However, the results based on VDI 4640 are not very precise. Reuss and Sanner [10] showed that the total length of the borehole heat exchanger is overestimated by up to 25%. This is because the provided values for the specific heat extraction apply only to the most common design and configuration [11]. In consequence, investment costs may be up to 25% higher than necessary, and this significantly affects the return on investment (ROI).

Conclusions

The CO_2 charged free-piston Stirling heat pump has been shown to offer two major benefits, namely high PER and reduced length of GSHE. The first item reduces heating costs while the second reduces installation costs. In addition, the difficulties that plagued the original efforts to develop this technology are avoided by the use of CO_2 and the resulting machine is very compact.

Acknowledgement

This work has been conducted under the Terra Therma Consortium sponsored by the E.U., under the FP6 framework, consisting of the following companies: Baxi (U.K.), Centre for Renewable Energy Sources (Greece), Sustainable Engine Systems, Ltd. (U.K.), Tecnica en Instalaciones de Fluidos SL (Spain), Renewables Ireland, Ltd (U.K.), Re/genT BV (the Netherlands), Tracto-Technik Spezialmaschinen GmbH (Germany), Global Cooling BV (the Netherlands) and Pera Innovation Limited (U.K.). All assistance is gratefully acknowledged.

References

[1] General Electric Corporation, 1977, *Development and Demonstration of a Stirling/Rankine Gas Activated Heat Pump*, G. E. Report 77SDS4254, General Electric Advanced Energy programs, Philadelphia.
[2] Marusak, T.J. and Ackermann, R.A., 1985, *Free-Piston Stirling Engine Development*, GRI Rpt. 85/0117, Chicago.
[3] Ackermann, R.A., Clinch, J.M. and Privon, G., 1987, "Further Developments in the Design of a Free-Piston Stirling Engine Heat Pump for Residential Applications", ASHRAE Transactions, **93, Pt 2**.
[4] Chen, G. and McEntee, J., 1993, Stability criteria and capacity modulation for a free-piston Stirling engine driven linear compressor, *Heat Pump and Refrig. Systems Design, Analysis and Applications*, **29**, ASME: p. 89-93.
[5] Berchowitz, D.M. and Kwon, Y.-R., 2005, "Hermetic Gas Fired Residential Heat Pump", 8th IEAHP, Las Vegas.
[6] Berchowitz, D.M., Janssen M. and Pellizzari, R.O., 2008, "CO_2 Stirling Heat Pump for Residential Use", Int. Refrigeration and Air Conditioning Conference, Paper 2352, Purdue.
[7] Berchowitz D.M. and Richter M., 1985, "3kW Stirling Powered Generator Set", 20th IECEC, Miami Beach, FL.
[8] Arthur D. Little, 1986, *Status of Free-Piston Stirling Engine Driven Heat Pumps - Development, Issues, and Options*, Final Report, Report ORNL/Sub/84-00205/1, US DOE, Cambridge, MA.
[9] VDI 4640, 2008*, "Thermal use of the underground – fundamentals, approvals, environmental aspects"*, Verein Deutscher Ingenieure.
[10] Reuss, M., Sanner, B., 2001*, "Planung und Auslegung von Erdwärmesondenanlagen – Basis einer nachhaltigen Erdwärmenutzung, VDI Richtlinie 4640 und Berechnungsverfahren"*, SIA Dokumentation, Schweiz: p. 14-16.
[11] ibid: p. 9.