HERMETIC ENGINE DRIVEN COMPRESSOR USING R-744 AS REFRIGERANT AND NATURAL GAS AS FUEL

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ABSTRACT

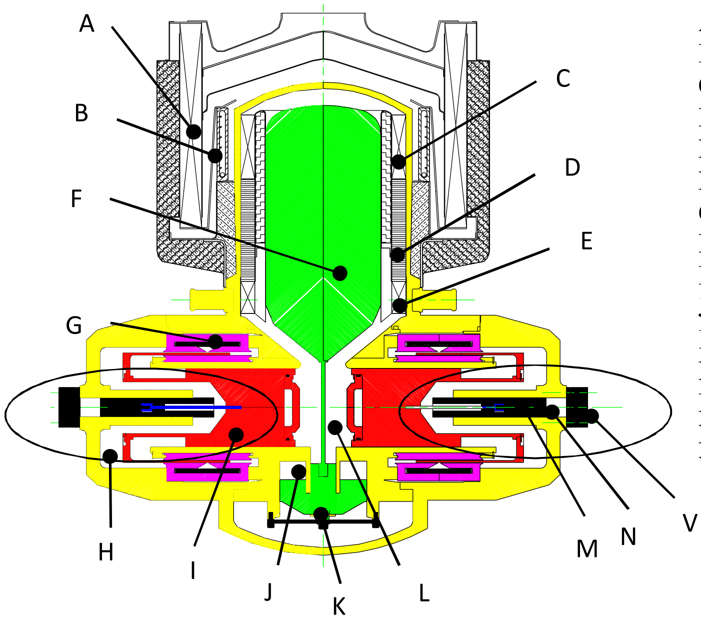
A compact gas-fuelled engine-driven compressor, using R-744 as refrigerant, for a domestic heat pump (10 kW) is being developed. The engine is of the Stirling type, which is combined with the compressor part in a single, completely, sealed system. Input energy is provided by recuperated gas burner. An overall system energy diagram and some principal sketches are presented to explain the basic functioning of the machine. Some pictures of a model of the engine driven compressor integrated into a heat pump are shown to provide more detailed insight in system design and configuration possibilities. It is shown that R-744, due to its thermodynamic properties, is the most suitable working fluid for such a combined system. Estimations of system performance show that high primary energy ratios (PERhs) of up to 1.8 are possible using a small capacity heat pump (4 kW). Approximately, 6 kW of heat is extracted directly from the engine and burner. A comparison with electric heat pumps is given and the pros and cons of the gas fuelled heat pump are presented. Sub components (heat exchangers, engine housing etc.) and their influence on the required refrigerant charge are discussed.

1. INTRODUCTION

This paper presents a summary of the work undertaken under the 6th European Framework Programme project acronym Terra Therma (TT) regarding the development of a novel type of domestic heat pump comprising a linear CO_2 compressor driven by a gas-fired Stirling engine combined with a novel arrangement of slantered ground source heat probes. The probes and their configuration are outside the scope of this paper. Gas fired heat pumps using this principle have been pursued before. Efforts include a Duplex Stirling arrangement by Sunpower in 1983 (Penswick and Urieli, 1984), a free-piston Stirling engine hydraulically driving a Rankine heat pump by Mechanical Technology Inc. (Moynihan and Ackermann, 1980), a free-piston Stirling engine coupled to an inertia compressor (Richards and Auxer, 1978), a second and third effort by Sunpower using an inertial drive and mechanic coupling to a Rankine Heat Pump (Wood *et al*, 2000), (Chen and McEntee, 1991), and various efforts in Japan (MITI, 1986) and Europe (Lundqvist, 1993). All these efforts identified the main advantages of the concept. Namely, its high part load efficiency coupled with a potential for high reliability and long life. However, in every case these efforts failed to produce a practical, cost-effective device mainly due to the perceived need to isolate the working medium of the engine and heat pump. The main advantage of the machine discussed here is that the working fluid of the compressor and Stirling engine are identical and that therefore the total embodiment can be placed inside a single hermetic shell. In the following, a short description of the main characteristics of the developed system and of the experience gained throughout its development campaign is given.

2. ENGINE DRIVEN COMPRESSOR

The engine of the compressor is of the free-piston Stirling engine (FPSE) type. A sketch of the engine driven compressor is presented in Fig. 1. The main advantage of the design is that both the Stirling engine and the heat pump are using the same working fluid. An ideal working fluid for a Stirling engine has low specific heat capacity, resulting in a large increase in pressure for given heat input, high thermal conductivity and low dynamic viscosity (minimum friction losses). As the Stirling engine size reduces with the average pressure of the working fluid and the pressure inside the Stirling and the heat pump low pressure side is coupled, a low boiling point refrigerant is required. Based on this and the requirement that the working fluid may not deteriorate at the operating temperature of the Stirling engine (900 K), the refrigerant R-744 was selected.



A) Recuperator
B) Combustor
C) Hot side internal heat exchanger
D) Regenerator
E) Heat exchanger to absorb reject heat
F) Displacer
G) Alternator
H) Bounce space
I) Stirling piston
J) Gas spring
K) Mechanical spring
L) Work space
M) Compressor piston
N) Compressor space
V) Compressor valve plate

Figure 1: Principle sketch of Stirling engine driven compressor in which the compressor part is indicated by the ovals.

The efficiency of a Stirling engine is proportional to the Carnot efficiency based on the temperature between the hot and cold side of the machine. To obtain a high efficiency, the machine is fitted with a recuperative burner (pos A, B in Fig. 1) having a flame temperature of 1400 K and a hot side temperature of 900 K. The cold side (E) of the machine is directly connected to the water loop of the home heating system. The machine will operate at the eigen frequency corresponding to the masses of the Stirling piston (I) and the displacer (F), the gas springs (J, H) and the loading of the compressor. The work space (L) of the machine is connected to the suction line of the heat pump; hence the internal pressures and the operating frequency vary with the operating condition of the heat pump. The displacer (F) is fitted with a gas spring (J) and a small mechanical spring (K). The kinetic energy of the displacer is mainly stored in the gas spring. The main purpose of the mechanical spring is to align the displacer within the machine. The kinetic and work energy of the Stirling piston is driving the compressor and is partly stored in the bounce space (H). To produce work a phase difference must exist between the motion of the displacer and the working pistons of the Stirling engine. The phase difference depends on the damping resulting from the viscous dissipation of the working fluid when passing through the Stirling engine part (i.e. friction losses over the internal parts, regenerator, the cold and warm side heat exchanger and the connection ducts) and from the inertia of the working gas and is also related to the natural frequency of the displacer. Next to starting the machine, the alternator also functions as a stabilizer during normal operation. The work delivered by the Stirling engine needs to match the work required for driving the compressor and the unbalance is compensated by the alternator. To minimise vibration, the system is fitted with two opposing compressors.

The performance of the Stirling engine is determined by simulation of the mechanical and fluid dynamics of the system, the details of which are beyond the scope of this paper. Final performance results of the engine without the burner are presented in Fig. 2. The X_p curves show the powers delivered at different piston amplitudes and demonstrate the particular advantage of power modulation possible with these devices. The burner employs a high-effectiveness recuperator and therefore is able to provide heat to the engine at an efficiency of between 85 to 90%. NOx is estimated at less than 10 ppm achieved by rapid quenching of the flame.

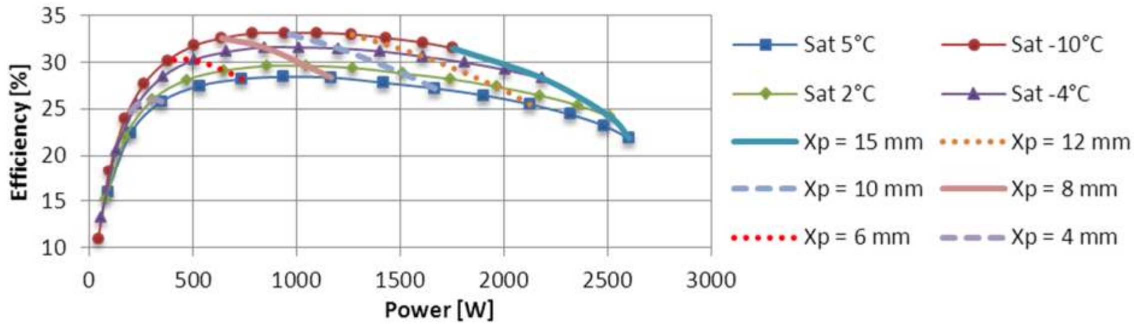


Figure 2: Efficiency versus power for various CO_2 saturation pressures and piston amplitude. Reject temperature is taken at 40 °C and acceptor temperature at 627 °C.

2. HEAT PUMP

A principle sketch of the Terra Therma heat pump is presented Fig.3. From the figure, it is seen that the working fluid 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 an internal heat exchanger (IHE) and exchanges heat with the flow from the gas cooler. It is then compressed from the suction pressure to the discharge pressure and its temperature rises. After compression, the fluid passes through the gas cooler where heat is rejected to the central heating system and then through the IHE where it is further cooled by gas leaving the evaporator. Finally, the fluid is expanded and returns to the evaporator. The Stirling engine and the flue gas recovery system reject heat to the central heating system as well.

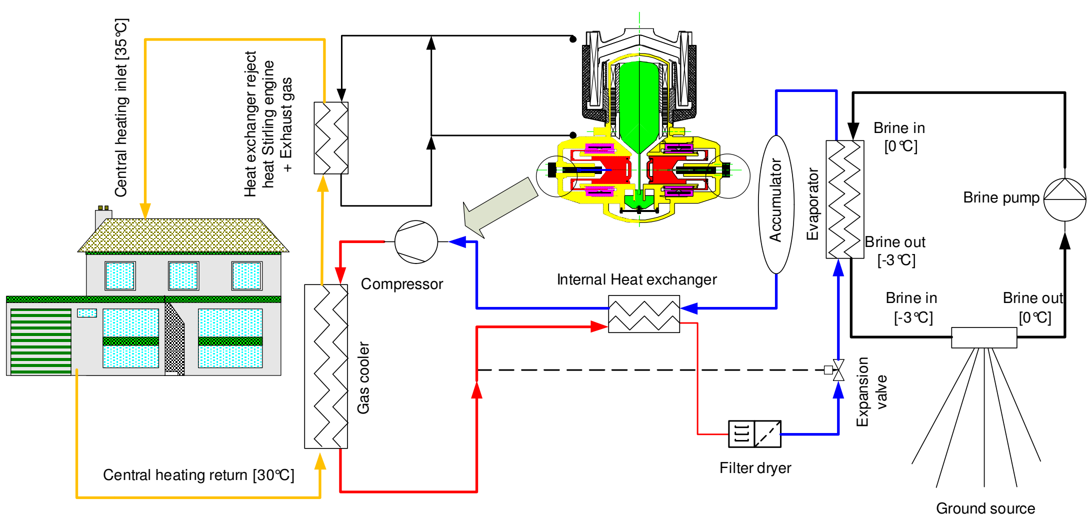


Figure 3: TERRA THERMA heat pump system in floor heating mode "system temperatures at rating condition, EN14511 (Brine for floor heating or similar operation (B0/W 35))".

3. COMPRESSOR PERFORMANCE

The compressor pistons are directly coupled to the Stirling engine pistons and are lubricated by gas bearings. The performance of the compression process is described by the volumetric (Eq. 1) and isentropic (Eq. 2) efficiencies.

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

$$\eta_i = \frac{P_{th}}{P_{in}} = \frac{\Delta h_i (1 - \epsilon_d) \dot{m}_{ideal}}{P_{in}} \tag{2}$$

The isentropic losses, manifested in P_{in}, result mainly from heat loss and viscous effects during compression, in addition to pressure losses across valves and mufflers. It is assumed that these losses can be kept within 5% of the theoretical compression power $P_{th,}$ thereby resulting in η_i = 0.95. The volumetric efficiency is directly related to the leakage over the compressor piston, dead volume and the losses of the gas bearings. Calculations and measurements, including CFD modelling (Fluent 6.2.16) and experimental testing of the compressor part only, show that it is possible to design an adequate piston bearing configuration in which the total leakage and bearing consumption (ε_b + ε_l) is within 5% of the mass flow at maximum capacity. The calculations were performed with a piston length of 100 mm, diametrical clearance of 18 µm +/- 2 µm and a top dead centre clearance of 0.5 mm. At the design conditions (EN 14511 (B0/W35)) this corresponds to a volumetric loss of ε_d = 5.6% (0.5 mm clearance). Therefore a total volumetric efficiency of η_v = 0.90 is assumed for the compressor at this condition.

4. HEAT PUMP MODEL

The thermodynamic model for the heat pump uses refrigerant properties from Refprop 6.01. The heat resistance of the selected gas cooler and evaporator are included through a temperature differential between the exit temperature of the working fluid and the inlet temperature of respectively the water and brine. The efficiency of the IHE was taken at 75% and validated against the manufacturer data of the heat exchanger selected. The model calculates the coefficient of performance (C.O.P.) of the heat pump cycle as defined by Eq. 3 and finds its maximum by discharge pressure optimisation.

$$C.O.P = \frac{Q_{gc}}{P_{in}} = \frac{\eta_i \eta_v \Delta h_{gc}}{\Delta h_i (1 - \varepsilon_d)} \tag{3}$$

Fig. 4 and 5 show the pressure enthalpy and the temperature enthalpy diagram at the design condition. Heat rejected to (+) or the heat absorbed from (-) is shown for each heat exchanger. The required compression work, the characteristic isotherms (gas cooler exit and discharge temperature) and the estimated system C.O.P. are shown at the design condition, EN-14511 (B0/W35). From fig.5 it is seen, from the trend of the isobar of 100 bar, that a large quantity of high quality heat is available in the initial part of the gas cooler (> 2 kW at a temperature of above 60 °C). The C.O.P. of the R-744 cycle is strongly related to the fluid temperature prior to throttling. Calculation of the C.O.P., applying a water flow temperature of 15 °C results in C.O.P. = 5.03 at an approach temperature of 22 °C, albeit at reduced discharge pressure (74 bar) and discharge temperature (92 °C), compared to the design condition with a flow return temperature of 30 °C, where the C.O.P = 3.39 at an approach temperature of 37 °C.

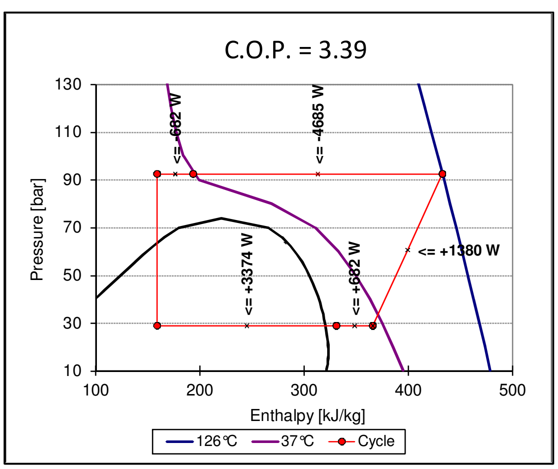


Figure 4: Pres. Enthalpy at EN 14511 (B0/W35)

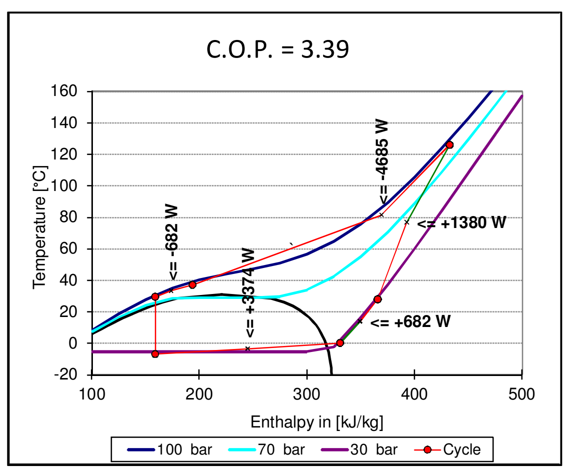
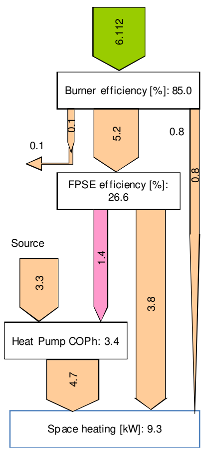


Figure 5: Temp. Enthalpy at EN 14511 (B0/W35)

5. SYSTEM PERFORMANCE

Modelling of the Stirling engine performance resulted in an operating efficiency of 26.6% at the high-power design condition. The burner efficiency is taken at 85%, which is similar to what has been achieved by the MTI and GE programs (Little, 1986). Exhaust heat recovery will recover about 90% of the lost heat. Fig. 6 shows a Sankey diagram presenting the energy flows of the TT heat pump at the design condition. In Fig. 7 the variation of the system efficiency against the loading of the machine is presented.



Fuel Input [kW]	6.112
Burner efficiency [%]	85
Recovery efficiency [%]	90
Burner reject heat [kW]	0.1
Burner recovered heat [kW]	0.8
FPSE Input [kW]	5.2
FPSE efficiency [%]	26.6
FPSE reject heat [kW]	3.8
Mechanical power [kW]	1.4
Heat Pump COPh	3.39
Heat flow source HP [kW]	3.3
Heat flow to sink HP [kW]	4.7
Space heating [kW]	9.3
PERh	1.53
PERc	0.54

Figure 6: Energy flow at EN 14511 (B0/W35)

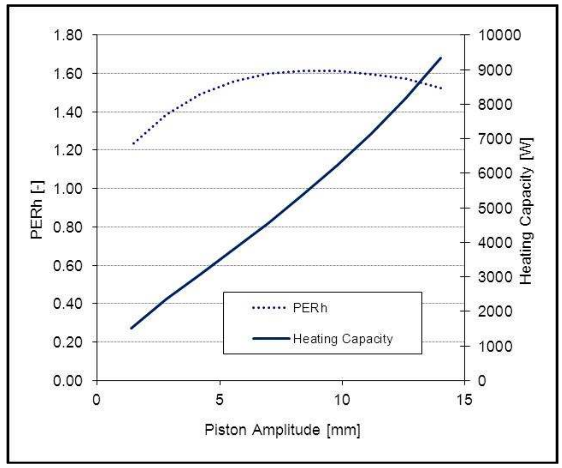


Figure7: Loading vs. PERh at EN 14511 (B0/W35)

6. COMPARISON

In the project, the emphasis was on proving feasibility of the TT heat pump. Therefore, the decision was made to focus on domestic heating only. However, the system has some advantages in producing hot water compared to standard electric heat pumps, although the integration of domestic hot water production is subject for further study; the principle advantages of the TT heat pump are highlighted.

In Europe, electricity is mainly generated through conventional power plants, having an average thermal plant efficiency of 49.8 % and the losses for transformation and distribution are 28.6% (including energy section consumption) (EEA, 2008). This adds up to a total efficiency of primary energy of 35.6% for the electricity arriving at the consumer. Electrical heat pumps typically have a C.O.P of 4.5 at EN14511 (B0/W35). Therefore, a PERh of 1.6 results for electrical heat pumps, which is similar to the PERh estimated for the TT heat pump at this operating condition (PERh = 1.53). In Europe, as of June 2011, average natural gas cost for households was €0.0588/kWh compared to €0.1728/kWh for electricity giving a spark ratio of 2.94 (www.energy.eu). Therefore, the TT heat pump has a substantial lower operation cost than an electrical heat pump. Compared to electrical ground source heat pumps the TT heat pump has the advantage that a smaller portion of heat is absorbed from the ground. Thereby, reducing the installation costs of the ground source heat exchanger by a factor two. Estimations for installation of the TT heat pump in Berlin showed a difference in installation costs of the ground source heat exchanger of 4200,- EUR compared to an electrical driven heat pump of equal heating capacity (Berchowitz *et al*, 2009).

As shown in Section 4, the TT heat pump has a large quantity of high quality heat available in the initial part of the gas cooler. Section 4 also showed that the water inlet temperature has a large impact on system C.O.P. For the example calculation, operation at an approach temperature of 22 °C, a total system PERh of 1.8 results. Therefore, the TT heat pump has the potential to generate hot tapping water at high efficiency, albeit smart integration of the system is required. Due to the increase in required pressure ratio, electrical heat pumps in general have relatively low efficiency when producing hot water. Calculations show a reduction in system C.O.P. from 4.4 at the design condition to a C.O.P. = 3.3 for an electric R-407C heat pump operating at a condensation temperature of 65 °C, an evaporation temperature of -7 °C and a sub-cooling and superheating of respectively 5 and 43 K, assuming an isentropic compressor efficiency of 70%.

7. DESIGN PRESSURE

For commercialisation, the system must comply with the applicable directives (i.e. Machine Directive, Pressure Directive). The only deviation, except from the Stirling driven compressor, from a standard heat pump originates from the use of R-744 (CO_2) as refrigerant and the corresponding relatively high pressure requirements. Type approved R-744 refrigeration components are commercially available; therefore, except for the engine no difficulties were encountered.

The hot section part of the Stirling driven compressor is designed based on maximum 1% creep after 100.000 operating hours applying a pressure of 100 bar at the operating condition (900 K head temperature) and 160 bar at cold conditions. To ensure safe design of the machine a safety factor of 2 is desired, hence maximum pressures of 50 and 80 bar are set for operation and rest.
To minimise the maximum system pressure and to comply with the applicable directives the system would have to be commercialised as complete assembled package. To avoid the blockage of liquid refrigerant and the possibility of extremely high pressures inside specific sections of the heat pump, no closing valves are fitted. In fact, at rest the refrigerant can move freely through the system and the pressure is constant throughout the system. In such designs the mass of refrigerant, the internal volume of the system and the maximum system temperature dominate the maximum pressure at rest.

Calculations indicate that the maximum refrigerant charge is required for the condition with high evaporation temperature and high discharge pressure (optimum discharge pressure at relatively warm return temperatures of the heating water). For example, the variation in refrigerant charge between this condition and at design condition according EN14511 (BO/W35) is approximately 140 grams of CO_2.

To cope with these variations in required refrigerant charge, the system is fitted with an accumulator, which is mounted directly after the evaporator. Often heat pump systems are controlled on the evaporator superheat (by varying the refrigerant throttling). Due to the accumulator, such control cannot be applied. To control the refrigeration loop (evaporator filling) and to ensure that the system will run at the highest possible efficiency, it was decided to control the system via pressure control on the discharge. Such control can be created using the discharge pressure as control signal for an electromagnetic expansion valve.

In the following table, a calculation of the refrigerant charge required and the rest-pressure (based on a maximum temperature of 60 °C) are given. In this calculation it is assumed that the evaporator exit is at saturated vapour condition and the accumulator is only filled with vapour. The calculation is based on applying commercially available plate heat exchangers suitable for CO_2 from SWEP.

Table 1: Refrigerant charge and resulting rest pressure at 60 °C.

Operating conditions				
Discharge pressure	110	[bar]		
Evaporation temperature	5	[°C]		
Temperature at rest	60	[°C]		
Pressure at rest	**64.2**	[bar]		
Component	Internal Volume [dm^3]	Refrigerant Mass [kg]	Specific volume [kg/dm^3]	Remarks / heat exchanger type
Stirling Engine	3.000	0.224	0.075	
Gas cooler	0.183	0.062	0.339	B17H *8
IHE (Discharge)	0.122	0.094	0.774	BDW16DWH*6
Evaporator	0.378	0.147	0.388	B12H*14
IHE (Suction)	0.183	0.020	0.108	BDW16DWH*6
Tubing	0.020	0.008	0.414	
Accumulator	1.000	0.114	0.114	
Total	**4.886**	**0.669**	**0.137**	

From the table it is seen that the internal heat exchanger requires quite a large quantity of refrigerant. By removing the IHE, hence reducing the efficiency of the system by approximately 5% on compressor C.O.P and 2% on overall system efficiency (PERh), the refrigerant quantity required reduces to 0.56 kg and the pressure at rest reduces to 58.7 bar. Further reduction of the rest pressure can be obtained by increasing the volume of the accumulator or by decreasing the internal volume of evaporator, gas cooler or tubing. Note that an evaporation temperature of 5 °C corresponds to a pressure of 40 bar; hence significant reduction of rest pressure requires large changes in system volume.

8. CURRENT STATUS

A working prototype was not built under the TT project but is seen as a logical next step in order to demonstrate the technology. The design phase was concluded with a rapid prototype construction of the Stirling driven compressor, which has been used for validation of the design and to display the working principle. In Fig. 8 a picture of a heat pump fitted with the gas driven compressor is shown. As a product, the total mass of the engine driven heat pump with burner is estimated at around 30 kg. A contained material cost has been estimated at 184, - EUR per unit (December 2009). Contained material costing does not include labour, part fabrication, amortization, shipping or any other fabrication cost. It represents a minimum cost associated with the material required to manufacture the device. Production costs would therefore be naturally higher than the contained material cost.

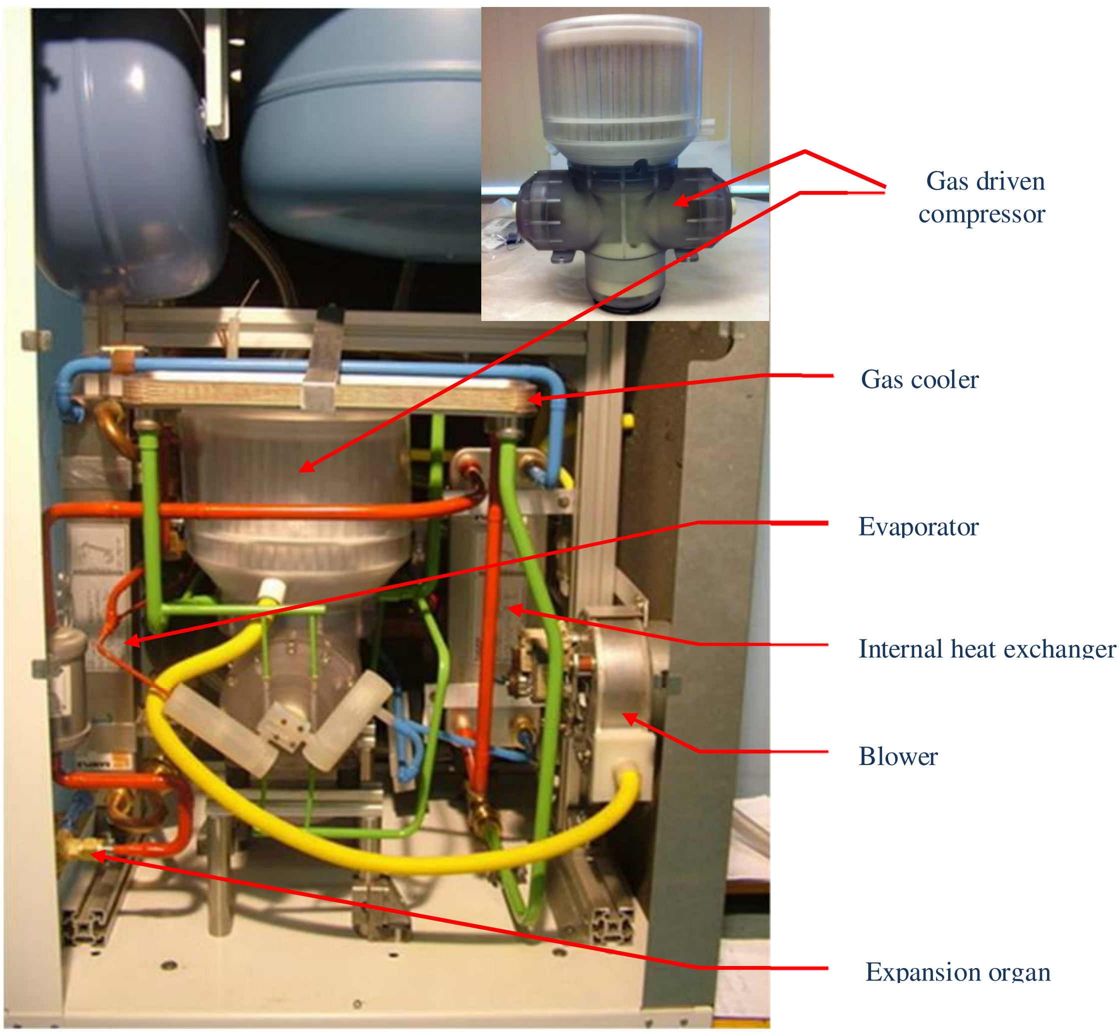


Figure 8: Gas driven compressor integrated in heat pump “prototype is built using R-407C components, high pressure refrigerant circuit in red, low pressure refrigerant circuit in blue, water circuit in green, air supply in yellow and brine circuit in copper “

9. CONCLUSIONS

The Terra Therma project showed that R-744 is an acceptable fluid for the contradicting requirements of the heat pump and the Stirling engine and that a practical product package can be designed around this concept. From comparison of the performance predictions with high-efficiency electrical heat pumps, similar thermal efficiencies have been found for these devices. Since natural gas costs are generally lower than electrical costs per unit of energy, a substantial cost savings is possible to the end user. High pressures in a sealed system can be adequately and safely dealt with by proper design considerations. However, the engine driven compressor will not be an off-shelf replacement of an electrical compressor but rather part of a complete integrated system.

ACKNOWLEDGMENT

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NOMENCLATURE

P_{in}	Input mechanical power (W)	ε_b	Fractional loss due to gas bearings (-)
P_{in}	Input mechanical power (W)	ε_d	Fractional losss due to dead volume (-)
Q_{gc}	Heat rejection gascooler (W)	ε_l	Fractional losses due to leakage (-)
V_{swept}	swept volume of compressor (m^3)	η_i	isentropic efficiency (-)
X_p	Piston displacement (mm)	η_v	volumteric efficiency (-)
Δh_i	Enthalpy change due to isentropic compression ($J\cdot kg^{-1}$)	ρ_{suc}	suction gas denisty ($kg\cdot m^{-3}$)
Δh_{gc}	Enthalpy change in gascooler ($J\cdot kg^{-1}$)	ω	Frequency ($rad\cdot s^{-1}$)

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