

Experimental Characterization of a Thermoacoustic Travelling-Wave Refrigerator

M. Pierens, J.-P. Thermeau, T. Le Pollès and P. Duthil

Abstract—The performances of a thermoacoustic travelling-wave refrigerator are presented. Developed in the frame of the European project called THATEA, it is designed for providing 600 W at a temperature of 233 K with an efficiency of 40 % relative to the Carnot efficiency. This paper presents the device and the results of the first measurements. For a cooling power of 210 W, a coefficient of performance relative to Carnot of 30 % is achieved when the refrigerator is coupled with an existing standing-wave engine.

Keywords—Refrigeration, sustainable energy, thermoacoustics, travelling-wave type heat pump

I. INTRODUCTION

THERMOACOUSTIC energy conversion is a generic technology that can be potentially used in a lot of applications involving heat: cold or electricity production from heat, cold or electricity source in the field of industry, building or transportation. In spite of a complex physical principle, including many scientific disciplines such as acoustics, thermodynamics, heat transfer, electroacoustics, the practical implementation of thermoacoustics is relatively simple.

Thermoacoustics energy conversion allows for converting heat into mechanical energy and vice versa. It is fundamentally based on the heat transfer between a pressured noble gas having a high thermal conductivity and a solid porous medium having a high heat capacity. The gas is oscillating and alternatively subjected to compressions and expansions which are induced and phased in time thanks to a high power acoustic wave. The mechanical energy of this intense sound is then converted into a heat transport along the direction of the acoustic propagation. One may place two heat exchangers at the ends of the porous medium to create a thermoacoustic heat pump. The gas is involved in a specific thermodynamic cycle whose properties remain only on the propagation features of the acoustic wave. For example, Stirling or Ericsson cycles are obtained using travelling waves and the inherent topology of the acoustic resonator. On a similar manner, a powerful sound can be generated inside the porous medium subjected to a thermal gradient induced by two heat sources yielding an engine vocation to the system. It then propagates within the gas and is amplified by means of an acoustic resonator.

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Thus, thermoacoustic systems have as main advantages the absence of any moving part, reducing investment costs and increasing reliability, and the use of inert and neutral gas for the environment. This technology is considered as very versatile using generic components to supply a wide range of applications. A thermoacoustic heat pump is not limited by the temperature of any fluid phase change and is indeed able to produce cold at temperature ranging from cryogenics to the ambient or heat at ambient to some hundreds of Celsius degrees. The coupling of a thermoacoustic wave generator supplying a thermoacoustic heat pump or refrigerator leads to a tri-thermal machine for which the external heat source supplying the system depends on the application and can be virtually anything: solar energy, heat wastes recovery, gas or biomass combustion, etc. This is similar for a thermoacoustic device consisting in a thermoacoustic wave generator driving an alternator for electricity production. The development of thermoacoustic devices may lead to energy or working cost savings and are considered as alternative for sustainable energy conversion.

This paper focuses on the experimental test of a thermoacoustic refrigerator developed in the frame of a European project funded under the EU's seventh framework program (FP7). The project name is THERMOACOUSTIC Technology for Energy Applications (THATEA) and its objective is to advance the basic and technological knowledge in the field of thermoacoustics. The project assesses the feasibility of thermoacoustic applications to achieve conversion efficiencies up to 40% of the maximum theoretical Carnot's efficiency.

In the following of this paper, we will first present our experimental bench consisting of a fully instrumented travelling wave type thermoacoustic refrigerator driven by a standing wave thermoacoustic generator. Then, the performances of the refrigerator will be given in terms of cooling power and coefficient of performance for different working conditions. Discussion about our results will then be given.

II. EXPERIMENTAL DEVICE

The experimental device (Fig. 1) consists in a Brayton thermoacoustic engine driving a refrigerator via an acoustic resonator. The design was carried out by use of the calculation code CRISTA [1]. This code is based on Rott's equations - modified by Swift - describing the linear propagation of an acoustic wave in a channel, where viscous and thermal dissipation (or gain) are taken into account. Some non linear effects such as acoustic minor losses or viscous dissipations induced by turbulence are added in the model. Considering plane wave propagation, the fluid equations of state, mass,

momentum and energy are then averaged in the direction transverse to the acoustic propagation to solve a 1D linear system. They are then discretized by use of second order finite differences scheme and integrated along the propagation direction by use of a fourth order Runge-Kutta algorithm. This leads to a shooting method inserted in a Newton-Raphson algorithm to converge towards the imposed boundary conditions. Giving the device geometry, materials properties of the components, the type of gas, and some working conditions such as the drive ratio (the ratio between the amplitude of pressure oscillations and the mean pressure; noted Dr and usually expressed in %), this computation code allowed for providing the acoustic field, the power on the different heat exchangers and hence the efficiency of a thermoacoustic system.

The working medium is helium gas at a mean pressure of 4.0 MPa to increase the power density of the sound wave. The operating resonance frequency of the system is 120 Hz. The refrigerator was designed to pump 600 W of heat at a temperature of 233 K with a COP of 40 % relative to the theoretical COP of Carnot.

A. The Brayton thermoacoustic engine

The Brayton thermoacoustic engine, which was used as an acoustic source to drive the refrigerator in this study, already existed in our laboratory. It had been designed and built four years ago as a versatile experimental set-up for testing different thermoacoustic components. For convenience and accuracy of the measurements, the heating power of this engine is provided by a resistive heating element delivering up to 7 kW of power at temperature up to 700 °C.

B. The acoustical resonator

The role of this element is triple: (i) it acts as a pressure vessel and stores the acoustic energy; (ii) it sets the resonance frequency; (iii) it transmits the net acoustic output power of the thermoacoustic engine to the inlet of the thermoacoustic refrigerator. The 304L stainless steel resonator consists in a 2.47 m long cylindrical straight duct having an internal diameter of 80 mm and a wall thickness of 2 mm. It ends with a 0.3 m long cone-shape that permits a smooth transition to the 90 mm diameter entrance of the refrigerator. A pressure relief valve protects the system from excessive pressure. Numerical simulations with CRISTA code predicted that the thermoacoustic wave engine may supply an acoustic power of 360 W with a drive ratio of 5 % at the refrigerator inlet.

C. The thermoacoustic travelling wave refrigerator

The travelling-wave type thermoacoustic refrigerator has a toroidal shape (Fig. 1). This resonator topology allows for an efficiency improvement of about 50 % by: (i) setting up a travelling wave within the porous medium thus permitting the achievement of a Stirling type cycle and (ii) acting as a feedback loop that recovers the acoustic power at the outlet of the porous medium and injects it at its inlet [2].

1) The thermoacoustic energy conversion core

This unit contains the high heat capacity porous medium that is called a regenerator, in reference to Stirling machines. It is made of a stack of 280 mesh stainless steel screens with a

wire diameter of 25 μm and a porosity of 78.4 %. This regenerator is encased in a 21 mm long 304L stainless steel tube with an inside diameter of 90 mm and a wall thickness of 1 mm. At each side of the regenerator are placed the heat exchangers.

The cold heat exchanger is dedicated to pump heat at a working temperature of 233 K. It is made of a copper cylinder having a diameter of 90 mm and a length of 20 mm. The oscillating helium gas circulates within this exchanger through 2225 drilled holes, each one having a diameter of 1.2 mm. A resistive heating element is inserted in a groove at the periphery of this exchanger to simulate a heat load.

The second heat exchanger operates at room temperature (293 K) and is called the aftercooler. It rejects to a mid temperature source the heat pumped on the cold heat exchanger plus the heat not used in the thermodynamic cycle. This cross flow shell-and-tube type heat exchanger has a length of 20 mm and an overall diameter of 90 mm. It is made of 1039 parallel copper tubes having an internal diameter of 1.6 mm and brazed on two copper flanges. The secondary fluid (water) passes around each one for cooling. The aftercooler temperature is maintained at 293 K by a water circulation. Below the aftercooler is placed a membrane in order to suppress continuous mass flow (Gedeon streaming) induced by the acoustic wave within the resonator loop which may deteriorate the refrigeration performances.

The buffer tube is a tapered duct with an angle of 5.2° to avoid the Rayleigh streaming which may occur within this element [3]. It is made of 304L stainless steel with a wall thickness of 1 mm. It has an inlet diameter of 90 mm and a length of 35 mm. At each side of this buffer tube are placed flow straighteners made of copper screens to avoid jet streaming.

The final heat exchanger is at room temperature and is similar to the cold heat exchanger excepting a length of 5 mm.

2) The insulation

The thermal insulation of the energy conversion core and of the buffer tube is made of two layers of glass wool surrounded by thermal shields made of stainless steel strips.

3) The phase shifter network

This network is made of an acoustic inductance in serial with an acoustic compliance. Hence, it allows to modify the phase angle between the pressure and velocity to reach a near zero difference value within the regenerator in order to involve a Stirling type cycle. Moreover, it acts as a feedback loop that recovers the acoustic power at the output of the regenerator to inject it at its input. The inertance tube has an inside diameter of 84.9 mm and a length of 340 mm. The capacitance has a diameter of 97 mm and a length of 110 mm. One end of the capacitance is connected to the aftercooler while the other end is linked to the final heat exchanger.

D. The instrumentation

Many sensors are used to characterize the refrigerator performances. This requires measuring temperatures, mean and dynamic pressures, electric power provided to the cold heat exchanger and water flow within the aftercooler. Software communicates with sensors, display data and record them.

The mean pressure inside the system is measured by a piezoresistive sensor located just after the engine cold heat exchanger.

Several dynamic pressure sensors are used at different locations to measure: (i) the acoustic power at the output of the engine and at the refrigerator inlet by use of the two-sensors method [4], (ii) the acoustic field within the loop by means of four sensors (two on the inertance, one on the capacitance and one at the junction between the inertance and the refrigerator inlet).

Type K thermocouples are used for temperature measurement. Two are located along the regenerator and four along the thermal buffer tube at the outer side of the resonator wall. In addition, four sensors measure the temperature near dynamic pressure sensors and one the temperature of the resistive heating element. In the cold heat exchanger, two additional thermocouples are placed in the copper shell (at diametrical opposite positions) and another one measures the gas temperature at the center of it. The aftercooler temperature is deduced by averaging the measured upstream and downstream water temperatures. By quantifying the water mass flow rate, the heat power rejected can be calculated.

III. RESULTS AND DISCUSSION

A heat machine and especially a refrigerator, is usually characterized by a Coefficient Of Performance (COP) defined as the ratio of the useful energy to the energy given to the system. Moreover, in order to compare system performances by taking into account the working temperature levels, this COP is usually compared to the COP of the ideal Carnot thermodynamic cycle (COP_C):

$$COP = \frac{Q_c}{W} \quad \text{and} \quad COP_C = \frac{T_{cold}}{T_{hot} - T_{cold}}$$

The Coefficient Of Performance Relative to Carnot (COP_R) is also called exergetic efficiency and is defined as:

$$COP_R = \frac{COP}{COP_C}$$

The following experimental protocol was used to test the refrigerator. After starting the thermoacoustic engine by increasing the temperature of the hot heat exchanger, the heating power is raised to its maximum value. At the same time, the refrigerator cold heat exchanger temperature is maintained at 233 K by providing heat to this element through the resistive heating element. Once a stationary state is reached, temperatures, acoustic and electric powers, as well as water flows are recorded and a data processing is made. Then, the acoustic output power of the thermoacoustic engine (and the drive ratio) is successively decreased by decreasing the heating power to several power levels ranging from 7 kW to 2 kW, always maintaining the refrigerator cold heat exchanger temperature at 233 K.

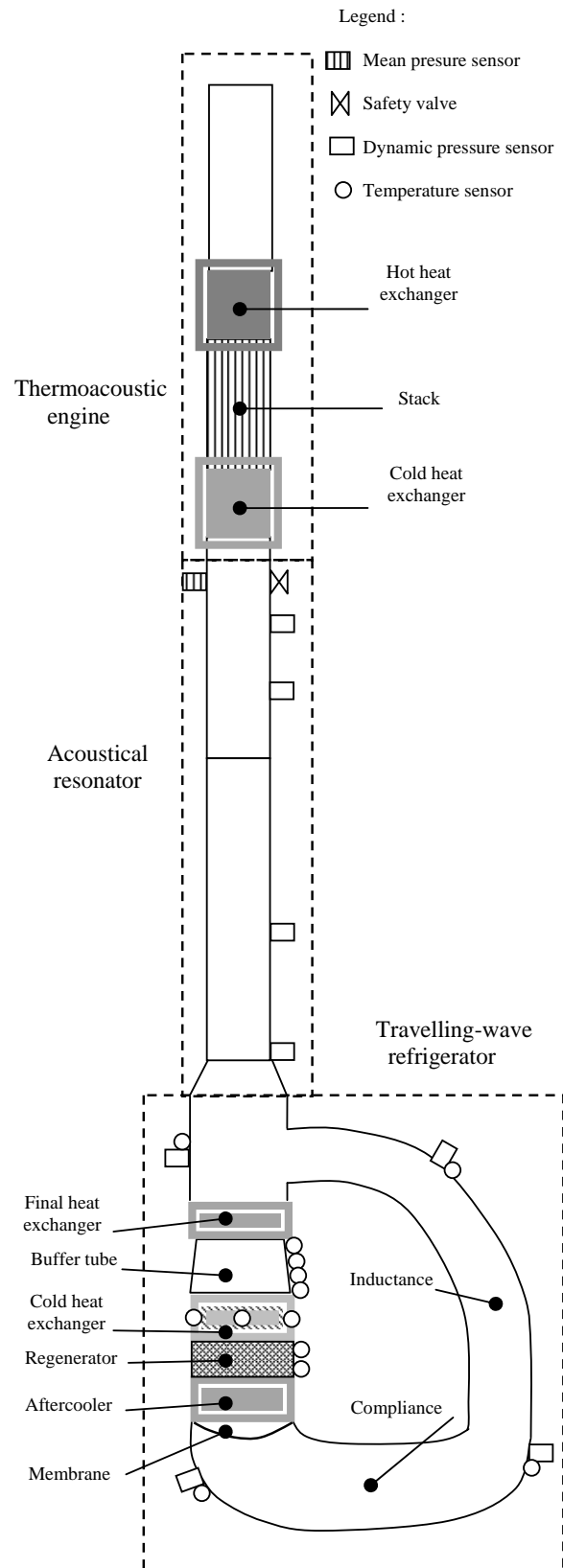


Fig. 1 Scheme of the experimental device

Fig. 2 shows the cooling power measured at the cold heat exchanger at a fixed temperature of -40°C and for different drive ratios. It is noticed that the cooling power increases linearly with the drive ratio until a value of 3.4 %. The corresponding COPR are presented on Fig. 3. The best performance was obtained for a drive ratio of 3.4 %: a cooling power of 210 W is achieved with a COPR of 30.3 %. It must be noted here that the thermoacoustic refrigerator has not been tested at the design working conditions. Indeed, currently we use a shorter transition between the resonator and the refrigerator. It is made of aluminum alloy and has a length of 0.05 m. This results in a larger operating frequency (130 Hz) than the design one (120 Hz).

Those first results have shown that the pressure difference between the two ends of the inductance was not strong enough compared to the first design. In order to correct this problem, we decide to increase the value of the inductance. From a mechanical point of view the only way was to add inserts in order to reduce the flow section within this element. Three different inserts have been made of PVC. They all have the same length of 0.340 m but have different diameters: 0.03 m, 0.04 m and 0.05 m. They are maintained at the center of the inductance by a set of suspensions made of springs.

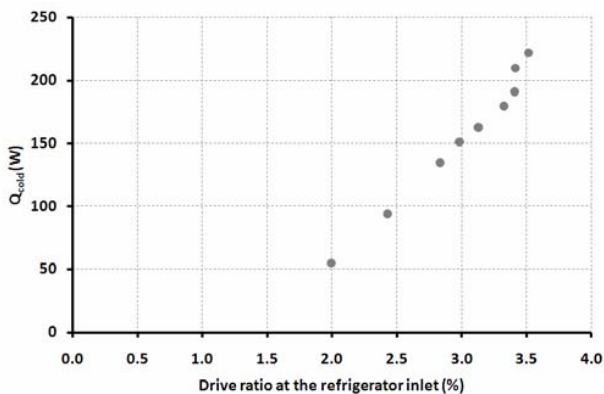


Fig. 2 Cold power as function of Drive ratio for a cold heat exchanger temperature of 233K

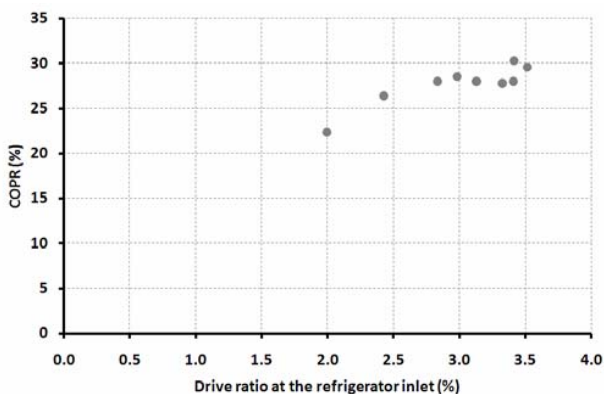


Fig. 3 COPR as function of Drive ratio for a cold heat exchanger temperature of 233K

Test runs with the three different inserts placed in the inductance allow us to visualize the changes of the operating

points of the refrigerator. Fig. 4 shows the changes of the cooling power and the corresponding COPR induced by the three inserts. The heat power input is fixed to 7 kW. As predicted by CRISTA, increasing the insert diameter results in increasing the cooling power but lowering the efficiency of the refrigerator. Indeed, in the loop configuration of the refrigerator, the drive ratio in the regenerator is directly influenced by the pressure drop in the inductance. Higher the pressure drop in this element is, higher the drive ratio and the acoustic power within the regenerator is. As a result, the pumping power is increased. However, the inserts: (i) increase the surface of the inductance leading to more dissipations and a lower efficiency, (ii) can change slightly the phase inside the regenerator.

By placing an insert of 50 mm, a maximum of 290 W of cooling power at -40°C is reached, with a COPR of 26 % and at a drive ratio of 3.8 % at the inlet of the refrigerator.

It should be noted that the efficiency of the thermoacoustic engine is improved by placing an insert within the inductance because more acoustic power is produced with the same heating power.

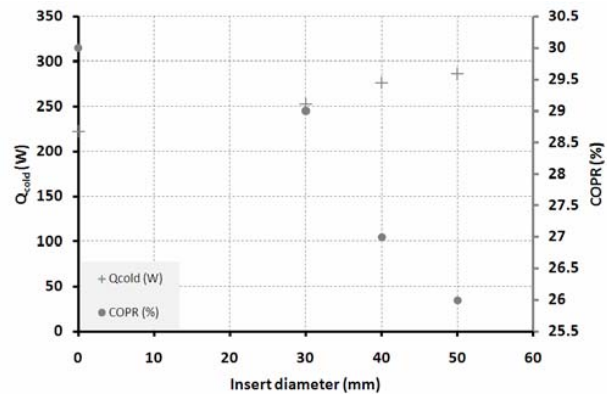


Fig. 4 Evolution of Q_{cold} and COPR as function of inductance insert diameter for a cold heat exchanger temperature of 233K

To complete the investigation about the effect of the resistive part of the impedance of the refrigerator, we will next change the regenerator length. This will allow for increasing generated acoustic power driving the refrigerator and for changing strongly the phase between pressure and velocity inside the regenerator.

IV. CONCLUSION

A travelling-wave thermoacoustic refrigerator, driven by a Brayton thermoacoustic engine, has been designed, constructed and tested. Although results described in this paper are different from design values, we have already obtained a coefficient of performance relative to Carnot of 30 % with a cold power of 210 W at 233 K. Moreover, the numerical simulation code CRISTA has well described the behavior of the refrigerator when an insert is put in the inductance. Several test runs with different inserts in the inductance have shown that the output acoustic power of the engine can be improved to obtain a higher drive ratio at the refrigerator inlet and thus a better cooling power but with

smaller energy conversion efficiency. Improvements will be made in order to achieve the right drive ratio while increasing coefficient of performance relative to Carnot to meet the design operating point.

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