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Development of a thermoacoustic travelling-wave refrigerator

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In the framework of the European project called THATEA focusing on energy conversion efficiencies of thermoacoustic systems, a thermoacoustic travelling-wave refrigerator has been developed. Its performances are presented in this paper. This system provides 210W of cooling power at a cold temperature of 233 K with a COP of 30% relative to the theoretical Carnot COP. We first explain the working principle of such a system, and then we present the refrigerator, its instrumentation and its experimental test bench. Finally we give the results obtained from our experimental campaigns investigating the effect on system performances by modifying two key components geometry and dimensions: the buffer tube and the inductance.

1   Introduction

The Thermoacoustic Technology for Energy Applications (THATEA) European project [1], funded under the EU's seventh Framework Program (FP7), has for objective to advance the basic and technical knowledge in the field of thermoacoustics. It assesses the feasibility of thermoacoustic applications to achieve conversion efficiencies up to 40 % of the ideal Carnot efficiency.

A thermoacoustic refrigerator is a heat machine that converts a mechanical power, here in the form of an intense acoustic wave, into a heat transfer from a cold to a hot source. The thermoacoustic energy conversion takes place in a porous medium with a high heat capacity that is called regenerator. Inside it, a gas particle with a high thermal conductivity is subjected to an acoustic wave. Thereby, gas particle pressure and velocity oscillate around a mean value. According to the phase shift between the combination of compressions/expansions and displacement induced by the acoustic wave propagation, the thermodynamic cycle experienced by the gas particle is modified. In the case of a travelling-wave, acoustic pressure and velocity are in phase inducing a Stirling type cycle. Over an acoustic period, heat has been carried along the direction of the acoustic propagation from one extremity to the other of the porous medium. By putting heat exchangers at the end of it, we obtain a refrigerator or a heat pump.

One part of this project, the Work Package 2 (WP2), comprises two steps. The first one is the development of a thermoacoustic travelling-wave refrigerator that goes from the design to the tests. The second step is the integration of this thermoacoustic energy conversion core into a low-temperature multi-stage thermoacoustic system realized by Aster [2] to complete a tri-thermal machine.

The specifications of the refrigerator are a cold temperature (T_C) of 233 K and a cold power (Q_C) of 540 W. The objective is to get a Coefficient Of Performance Relative (COP_R) to the theoretical COP of Carnot (COP_C) of 40%. The operating frequency of the system is 120 Hz. It is filled with helium gas at a mean pressure of 4.0 MPa to increase the power density of the sound wave.

For the thermoacoustic design, we use a calculation code named CRISTA [3]. Its working principle is detailed later in this paper. One objective of the mechanical design is to get a resonator thickness as low as possible that ensures mechanical resistance from mean pressure. By that way, conduction losses on the cold heat exchanger are minimized. However, components length given by CRISTA are small that creates thermal bridges. To reduce those thermal losses, the work is done in an iterative manner between mechanical and thermoacoustic design to take into account respective constraints and to converge to a workable solution.

To characterize experimentally the performances of the refrigerator, we use an existing thermoacoustic standing-wave generator.

In the following of this paper, the first part is dedicated to the presentation of the calculation code, the related design method and numerical results of the simulations concerning refrigerator performances. The experimental bench and its instrumentation are also described. Then the performances of the thermoacoustic travelling-wave refrigerator will be given for different working conditions, compared with calculation code estimations and then discussed.

2   Numerical and experimental resources

To supply the refrigerator, we use an existing standing-wave thermoacoustic generator built five years ago for testing different components. Not optimized to this application, it is modified to provide more acoustic power to the refrigerator.

2.1   CRISTA

CRISTA is a numerical calculation code based on Rott's equations, modified by G. Swift, describing the linear propagation of an acoustic wave in a channel, where viscous and thermal dissipations (or gains) 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 one dimension 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 into 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 more working conditions such as the drive ratio (the ratio between the amplitude of acoustic pressure oscillations and the mean pressure; noted DR and usually expressed in %), this computation code provides the acoustic field, the power on the different heat exchangers and hence the efficiency of a thermoacoustic system.

The design of the thermoacoustic experimental system begins with a preliminary design based on a simple model. It gives approximate components geometry. Then, some of them are optimized, by using parametric study when necessary. Thermal and acoustic losses are minimized, to converge towards the required performances. Thereafter, the architecture is improved with the mechanical design.

CRISTA predicts an acoustic power at the refrigerator inlet of about 360 W with a DR of 5 %. The usable cold power is about 685 W that leads to a COP_R of 49 %.

2.2 Experimental device

The experimental device (Figure 1) consists in a thermoacoustic engine supplying the refrigerator with acoustic power and a resonator to link and adapt sub-systems together.

The heating power of the engine is provided by a heating resistive element delivering up to 7 kW at a temperature up to 700 °C. The acoustic resonator has a diameter of 80 mm and a length of about 2.5 m. It ends with a 0.3 m long cone-shape to ensure a smooth transition to the 90 mm diameter refrigerator entrance. 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.

This refrigerator is a travelling-wave type obtained by use of a torus resonator shape. This 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 thermoacoustic energy conversion core to inject it at its inlet [4].

The conversion core contains a high heat capacity porous medium called regenerator in reference to Stirling machines. It is made of a 21 mm long stack of 280 mesh stainless steel screens with a wire diameter of 25 μm and a porosity of about 78 %. At both side of this unit are placed heat exchangers.

The cold heat exchanger is made of a copper cylinder having a diameter of 90 mm and a length of 20 mm. It is composed of 2225 drilled holes with a diameter of 1.2 mm in which the gas oscillates. For a refrigeration application, the thermoacoustic heat pumping takes places in this element. In our case, to simulate the presence of a heat load, we use a resistive heating element inserted in a groove at the periphery of the heat exchanger.

The second is called aftercooler and is at room temperature (293 K). This cross flow shell-and-tube type heat exchanger has 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 aftercooler temperature is maintained at room temperature by water circulating around each tube. Below the aftercooler is located a membrane in order to suppress continuous mass flow (Gedeon streaming [5]) induced by the acoustic wave within the resonator loop that decrease refrigerator performances.

The buffer tube is a tapered duct to avoid the Rayleigh streaming which may occur in this element [6]. It has an inlet diameter of 90 mm and a length of 35 mm. At both sides are placed flow straighteners made of copper screens to avoid jet streaming.

To limit caloric intake, the thermoacoustic energy conversion core and the buffer tube are insulated by two layers of glass wool surrounded by thermal shields made of stainless steel strips.

Above the buffer tube is the final heat exchanger. It is at room temperature and similar to the cold heat exchanger excepting a length of 5 mm. From CRISTA's results, this part does not exchange heat and does not seem to be useful.

The phase shifter 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.
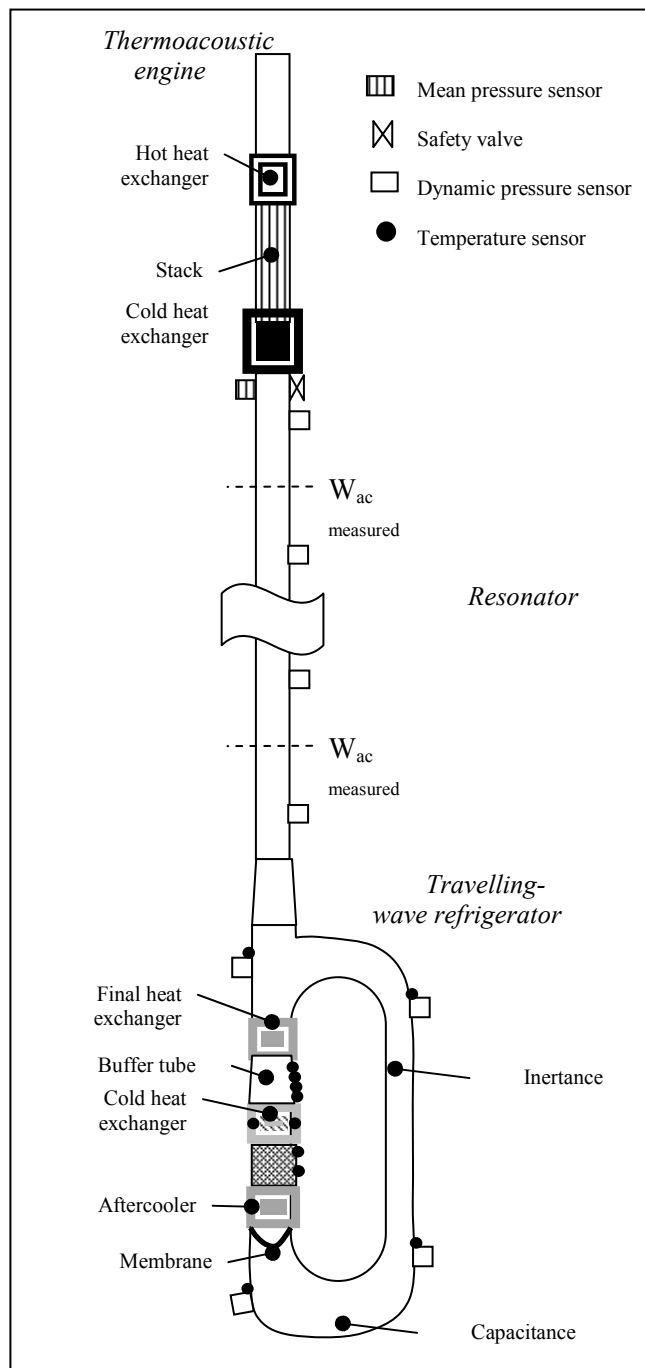

Figure 1: Scheme of the experimental device.

Concerning 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-sensor method [7], (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). 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.

For safety reasons, a pressure relief valve is putted on the acoustical resonator.

3   Results and discussions

To characterize heat machines energy conversion performances, the COP, defined as the ratio of the cold power to the acoustic power at the refrigerator inlet is used. To compare systems performances, the criterion is the COP_R, also called exegetic efficiency, because it takes into account the heat sources temperature levels. It is defined as the ratio of the COP to the COP_C. The COP_C is the ratio of cold temperature to the temperature difference between hot and cold sources.

3.1   Experimental set-up

The same protocol is applied for each measurement campaign. The first step is the starting of the thermoacoustic wave. A low electric power is supplied on the resistive heating element of the engine hot exchanger while water at ambient temperature is circulating in the cold exchanger. After one hour and a half, the acoustic wave rises in the resonator always around an onset temperature of 280 °C. At this moment, heat pumping starts in the travelling-wave refrigerator. The refrigerator cold exchanger is maintained at a temperature of 233 K, by dissipating electric power in the resistive heating element, for the rest of the test run. The water circulation in the aftercooler is also opened. Then the electric power is increased to its maximum value (7 kW). Once a steady state is reached, a data processing is made. The electric power on the engine is decreased to several values from 7 kW to 2 kW. That corresponds to a DR ranging from 3.5 to 2.8 %. Thus we obtain the behaviour of the system as, among others, the evolution of the cold power and COP_R as function as the acoustic power or the drive ratio at the refrigerator inlet.

3.2   Buffer tubes

During the first tests, the system worked not as expected. The cold temperature was reached after a long time and temperatures on the refrigerator cold heat exchanger and along the buffer tube were unstable. From temperatures measured on the refrigerator, we have also observed the cooling of the phase shifter network and the resonator connecting the refrigerator to the engine. The COP_R of the refrigerator was about 6%. Rayleigh Streaming occurred in the system. We have then modified the shape of the buffer tube and moved the membrane below the aftercooler above the buffer tube. In this operation, the final heat exchanger has been removed.
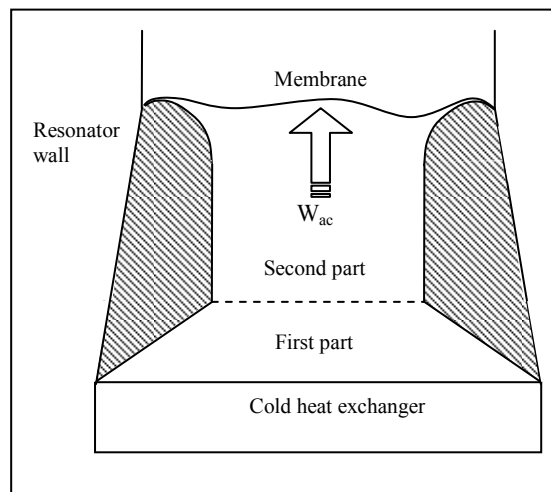

Figure 2: Scheme of the new buffer tube shape.

The new buffer shape, as shown in Figure 2, has been declined in four versions. The dimensions are given in Table 1. It is made of two parts. The duct in contact with the cold heat exchanger is conic and is a transition between the cold heat exchanger and the next part. Then there is a straight duct that is rounded at its end to prevent membrane deterioration. The overall length is kept constant at a value of 69 mm. The diameter at the buffer tube extremities is 90 mm.

Table 1: Dimensions of the buffers tubes

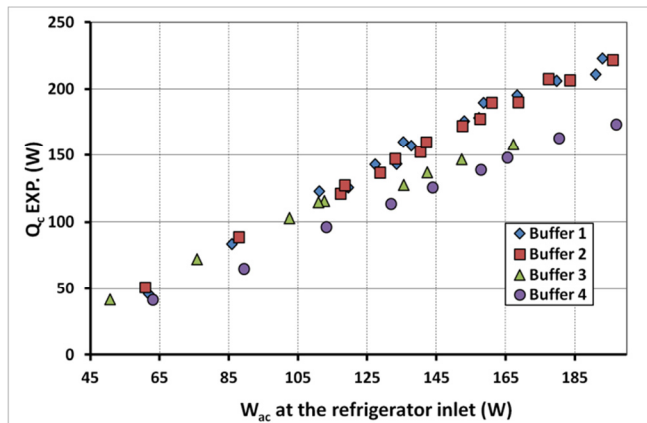| Buffer version	Second part	
	Diameter (mm)	Length (mm)
1	56	41
2	50	45
3	50	50
4	64	41

Figure 3: Graphic of the evolution of experimental cold power as function as acoustic power at the refrigerator inlet for different buffer versions.

The Figure 3 shows the evolution of the cold power of the thermoacoustic travelling-wave refrigerator as function as acoustic power at the refrigerator inlet. In that representation, the slope of each curve is the COP of the refrigerator. Acoustic power at the refrigerator inlet is ranging from 50 W to 200 W and cold power from 40 W to 220 W. As predicted by the theoretical model, the evolution of Q_c is linear as function as acoustic power and that is particularly true for buffers 1 and 2 for which curves are similar. Near the value of 110 W of acoustic power, there is a slope modification for buffers 3 and 4, the COP_R decreases. The maximum COP_R is 33 % and reached with the buffer 1 with an acoustic power at the refrigerator inlet of 160 W. In comparison, the worst is 23% (buffer 4) at the same acoustic power.

From those observations, it is possible to say that the different buffer versions have an impact on the refrigerator performances. The COP_R varies according to versions from 23 % to 33 %. It is increased from previous configuration from 6 % to 30 % at a cold temperature of 233 K. The important improvement is the modification of the buffer tube shape and the membrane position change. In fact, with membrane at this location, we have experimentally observed that resonator cooling is avoided. For the following, the buffer 2 is used to characterize the system performances.

3.3 Inductance inserts

From analyze of previous experimental measurements, we have observed a low acoustic pressure drop inside the inductance. To correct this problem, we have decided to change this element in order to increase the inductive effect of the phase shifter network. By taking into account mechanical constraints (resonator dimensions are fixed), we have putted inserts made of PVC in the inductance. They have the same length as the inductance, 0.340 m, but different diameters: 30 mm, 40 mm and 50 mm. They are maintained at the centre of the duct by a set of suspensions made of springs. This technical solution has for consequences: (i) a smaller cross section area and a greater surface in regards to the gas oscillations leading to higher acoustics velocities in the inductance and so more dissipations and (ii) a slight modification of the phase shift between acoustic pressure and velocity in the regenerator that can change refrigerator performances.
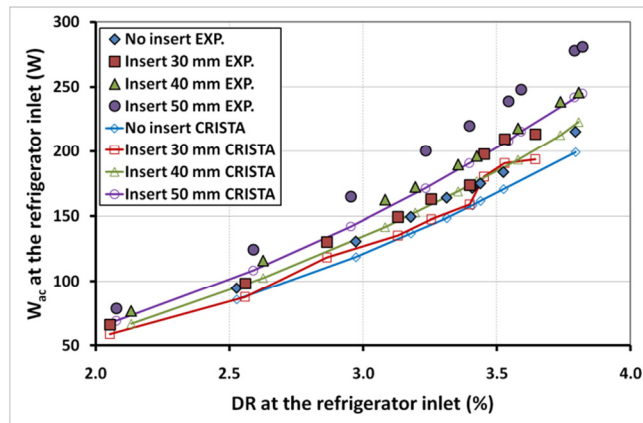

Figure 4: Evolution of the acoustic power as function as the drive ratio at the refrigerator inlet with and without the inserts between CRISTA results and experimental measurements.

The graphic above (Figure 4) shows a good agreement between experimental measurements and CRISTA. The calculation code underrates the acoustic power at the refrigerator inlet and the maximum relative difference is 15 %. In comparison with buffer tubes experimental results, the maximum acoustic power increases from 215 W to 290 W and cold power from 220 W to 285 W (without insert and insert 50 mm).
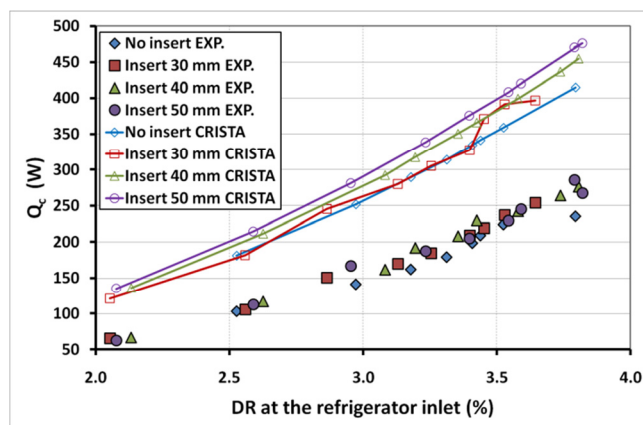

Figure 5: Evolution of the cold power as function as the drive ratio at the refrigerator inlet with and without the inserts between CRISTA results and experimental measurements.

Concerning the cold power (Figure 5), CRISTA predicts an evolution of Q_c vs. DR that is similar to experimental results. However, the relative difference between CRISTA and experimental cold power is important with a minimum of 55%. With the insert diameter increase, the cold power evolves in the same manner but the system has a lower COP_R.

It should be noted here that for the same drive ratio, the acoustic power at the refrigerator inlet increases with the presence of an insert and with its diameter. The performances of the engine are better because the coupling between engine and refrigerator is improved. Unfortunately, the acoustic power gain is not beneficial for the refrigerator performances. In fact, cold power is higher because there is more acoustic power but there are also more dissipations in the inductance. That is why the COP_R is almost same. The comparison of results from calculation

code and measurements shows that CRISTA makes a good estimation of the acoustic power at the refrigerator inlet but overrates the cold power and so the COP_R of the system.

4 Conclusion

In this paper, the design, construction and tests of a thermoacoustic travelling-wave refrigerator are described and performances, focused on energy conversion efficiency, are given. Driven by an existing standing-wave engine, adapted to this application, this refrigerator allows pumping heat at a cold temperature of 233 K with a cold power of 210 W. These values are far from the design (685 W@233 K with a COP_R of 49 %) but this is encouraging compared to the first tests.

Although the difference between calculation results and experimental measurements is important, except for acoustic power at the refrigerator inlet, the behavior of the system by putting a new buffer tube or an insert in the inductance is well described by CRISTA. One reason of this difference is due to three-dimensional effects occurring at the junction between the refrigerator and the phase shifter network inlet and the buffer tube end. At this location the wave-interaction is not in the model and so not taken into account by CRISTA.

The new buffer shape combined with moving membrane position improves significantly the refrigerator performances (COP_R changes from 6 % to 30 %). Concerning different buffer versions, experimental results show an impact on the operating point of the refrigerator. The version 4 has poor performances in comparison with versions 1 and 2.

Inserts in the inductance underline that a better coupling is possible between the engine and the refrigerator to get more cold power but with smaller energy conversion efficiency. The regenerator length as also been increased by 8 % to modify the phase shift between acoustic pressure and velocity in it, and insert in the inductance removed. This solution, as the first one does not change the operating point of the refrigerator significantly.

Finally, the cooling thermoacoustic energy conversion core has been integrated into the multi-stage travelling-wave system. When placed in this configuration with low drive ratios, efficiency is similar. That indicates the flexibility and freedom of implementation of such a system.

Acknowledgments

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