FR9501248
CEA-CONF--11914

STATUS OF PULSE TUBE DEVELOPMENT AT CEA / SBT

A. Ravex and P. Rolland
CEA / Département de Recherche Fondamentale sur la Matière Condensée
Service des Basses Températures
17 rue des Martyrs 38054 Grenoble Cedex 9 - FRANCE

ABSTRACT

Interest in the pulse tube comes from its potential for high reliability and low level of induced vibration.

A numerical model has been developed to provide a tool for practical design. It has been successfully validated against the experimental results obtained with a single stage double inlet pulse tube which has achieved a temperature of 28 K at a frequency of a few Hz.

Further developments have demonstrated the capability of operating a pulse tube at higher frequencies in association with a Stirling pressure oscillator.

Current projects include coaxial geometry for miniature pulse tubes with linear resonant pressure oscillators. A 4 K multistaged pulse tube is also in development.

INTRODUCTION : PULSE TUBE REFRIGERATORS DESCRIPTION

The basic pulse tube refrigerator was first described by Gifford and Longsworth (1) in 1964. It is shown schematically in Figure 1. The pulse tube itself is a thin walled cylinder with heat exchangers located at each end. It is supplied through a regenerator with pressure waves produced either by a pressure oscillator or by a three way distributor associated to a compressor. The cooling effect relies on heat exchange between the gas and the tube wall known as surface heat pumping (2, 3). Such a device operates at low frequencies (1-5 Hz) and has achieved a low temperature of 124 K (4). The low efficiency of this basic design was certainly the main reason why the pulse tube remained for a long time undeveloped.

In 1984 Mikulin et al (5) modified the basic pulse tube design by adding a valve (V1) and a reservoir volume at its closed end. They reached a temperature of 105 K using air as the working fluid and soon afterward Radebaugh et al (6) reached 60 K using helium. This new design shown schematically in Figure 1 is referred to as the orifice pulse tube.

An analytical model describing the behaviour of the orifice pulse tube has been developed by Storch and Radebaugh (7) to gain a better understanding of the refrigeration process.

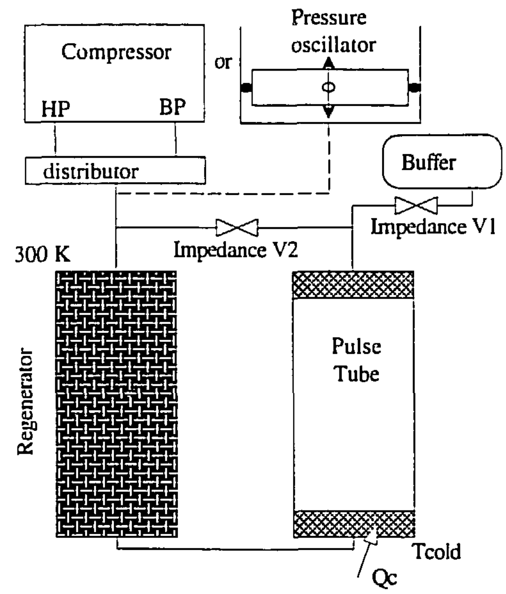

Figure 1: Schematic of a pulse tube refrigerator

This enthalpy flow analysis is based on the first law of thermodynamics : an energy balance on a control volume at the cold end of the pulse tube shows that the time averaged enthalpy flow over a period τ, assuming an ideal gas behaviour, is given by :

$$<\dot{H}> = \frac{C_P}{\tau} \int_0^{\tau} \dot{m}\, T\, dt$$

where $\dot{m}$ is the mass flow rate, C_P is the specific heat of gas at constant pressure and T the gas temperature.

Since the mass flow rate $\dot{m}$ is equal to $\rho.A_{pt}.u$ where ρ is the density of the gas equals to $\rho = P/rT$ for an ideal gas (P: pressure), u is the local gas velocity and A_{pt} is the cross section area of the pulse tube, the time-average enthalpy flow rate can be written as:

$$<\dot{H}> = \frac{C_P A_{pt}}{r \tau} \int_0^{\tau} u\, P\, dt$$

The net cooling power $\dot{Q}_c$ is then given by the following expression:

$$<\dot{Q}_c> = <\dot{H}> - <\dot{H}_r>$$

where $<\dot{H}_r>$ is the average enthalpy flow from the regenerator.

If the cyclic variations of pressure and velocity are assumed to be sinusoidal, the phase shift angle ϕ between them is the important parameter governing the cooling process. A direct comparison with Stirling cycle can thus be done. In a Stirling cryocooler the displacer/expander in the cold finger leads the motion of the pressure oscillator piston by about 90°. As a result the mass flow rate (i.e gas velocity) and the pressure at the cold end are approximately in phase leading to the expected refrigeration effect. In a pulse tube the proper phase relationship is obtained by the adjustment of the orifice impedances and reservoir volume at room temperature.

Although the orifice pulse tube refrigerator has a greatly improved efficiency in comparison with the basic pulse tube, the mass flow rate through the regenerator is still very large in comparison with a Stirling cryocooler and consequently the specific cooling power and ultimate temperature achieved aren't as good. This is mainly due to the fact that a large volume of gas with no refrigerative effect flows through the regenerator into the pulse tube be cause of the pressure oscillations. Zhu et al (8) have suggested the double inlet pulse tube concept to overcome this disadvantage and they reached a temperature of 42 K (9). As schematically shown in Figure 1 the second valve (V2) directly connects the hot end of the pulse tube to the pressure wave generator in this configuration.

Due to these successive improvements in performance the interest in the pulse tube refrigerators has grown rapidly in the last few years. Because they have no moving components in the low temperature region, they have the potential for high reliability and low vibration at the cold tip which are of major importance in satellite applications.

Recently Tward et al (10) have reported a system efficiency for a double inlet pulse tube operated with a flexure bearing Oxford type pressure oscillator which is very close to that of a comparable Stirling cryocooler.

THEORETICAL MODELISATION AND EXPERIMENTAL VERIFICATION

We have undertaken an experimental characterisation and a thermal modelisation of the double inlet refrigerator to get a better understanding of its operation and a practical tool for further system sizing and efficiency calculations (11).

EXPERIMENTAL SET UP AND RESULTS - The pressure oscillation is generated by an helium compressor connected to the pulse tube by way of an electromagnetic 3 way solenoid valve.

The regenerator consists of a stainless steel tube (18 mm inner diameter, 170 mm long) filled with 180 mesh stainless steel wire gauze discs.

Several pulse tubes (stainless steel tube 200 mm long) with inner diameter ranging from 10 mm up to 20 mm have been tested. At both end of the pulse tubes copper gauze discs are brazed for flow straightening and heat exchange.

At room temperature a buffer volume is connected to the pulse tube through an adjustable needle valve (V1) and a by pass adjustable needle valve (V2) is also inserted between the warm ends of the regenerator and the pulse tube. These two valves allow for tests in any type of configuration (basic, orifice or double inlet pulse tube).

The ultimate cold end temperature (no net cooling power) has been measured for various pulse tube diameters, pressure wave frequencies and opening of the needle valves. Typical experimental results corresponding to the optimised needle valves adjustments are reported in Figure 2. An ultimate temperature T = 28 K has been obtained in the double inlet configuration for the 14 mm inner diameter pulse tube at a frequency of 3 Hz with a pressure wave amplitude ΔP = 7 bar and a mean pressure P = 12 bar.

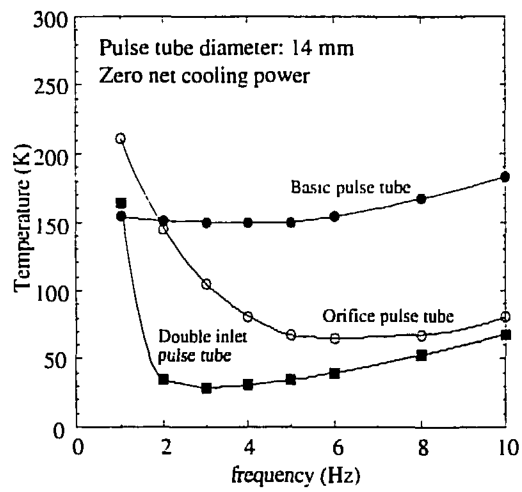

Figure 2: Ultimate temperature versus frequency

Cooling power measurements are also reported in Figure 3. They have been performed at frequencies and valve opening adjustments corresponding to the lowest temperature previously achieved with no net cooling power. In the double inlet configuration a net cooling power of 16 W at 80 K has been obtained.

The influence on the cooling performances of several parameters such as geometry of the pulse tube, regenerator mesh gauze disc matrices, pressure oscillation amplitude and frequency, mean pressure and valves opening adjustment has been systematically studied for further comparison with theoretical calculations from the theoretical model.

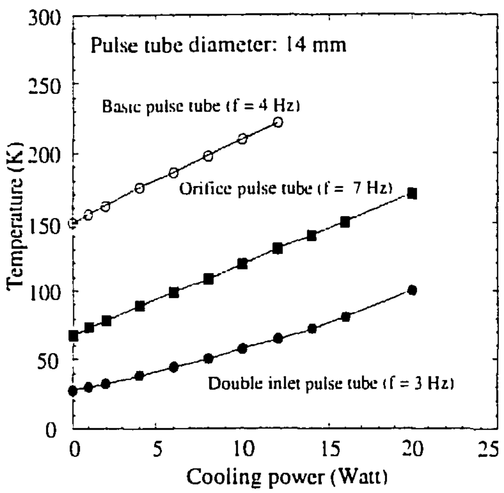

Figure 3: Cooling power for various configurations

MODELISATION - We have developed a numerical model to describe the double inlet pulse tube refrigerator based on the conservation of mass and first law of thermodynamics. The helium is assumed to behave as a perfect gas.

The gross refrigeration is calculated taking into account two independent contributions : the enthalpy flow in the bulk of the gas as described by Storch and Radebaugh (7) and the enthalpy flow due to heat exchange with the wall as described by Wheatley (2) or Richardson (3). The thickness of the layer in which the surface heat pumping effect occurs is calculated taking into account the thermal diffusivity of helium and solving radial transient heat transfer equations. In the remaining central volume an adiabatic plug flow is assumed.

Compression in the compressor or pressure oscillator is assumed to be isothermal. The hot and cold heat exchangers in the pulse tube are assumed to be perfect. The needle valves characteristics have been experimentally determined and the appropriate pressure drop relation versus mass flow rate is introduced in the calculation.

To determine the net cooling power from the gross refrigeration, parasitic contributions should be determined and subtracted. The thermal conduction through the pulse tube and regenerator stainless steel walls as well as through the regenerator metallic screens (empirical law) are thus calculated and taken into account. The contribution to the thermal loss resulting from the regenerator thermal inefficiency is calculated with a special subroutine developed in our laboratory in which the theoretical correlations for heat transfer in metallic screen packages proposed by Kays and London (12) are used. Related correlations for pressure drop are also used to determine the pressure oscillation amplitude attenuation through the regenerator resulting in a gross cooling power reduction.

Results predicted by this model are found to be in fairly good agreement with experimental data. For example, in Figure 4 the measured net cooling powers at 100 K for various valves opening and as a function of frequency are compared with calculated data.

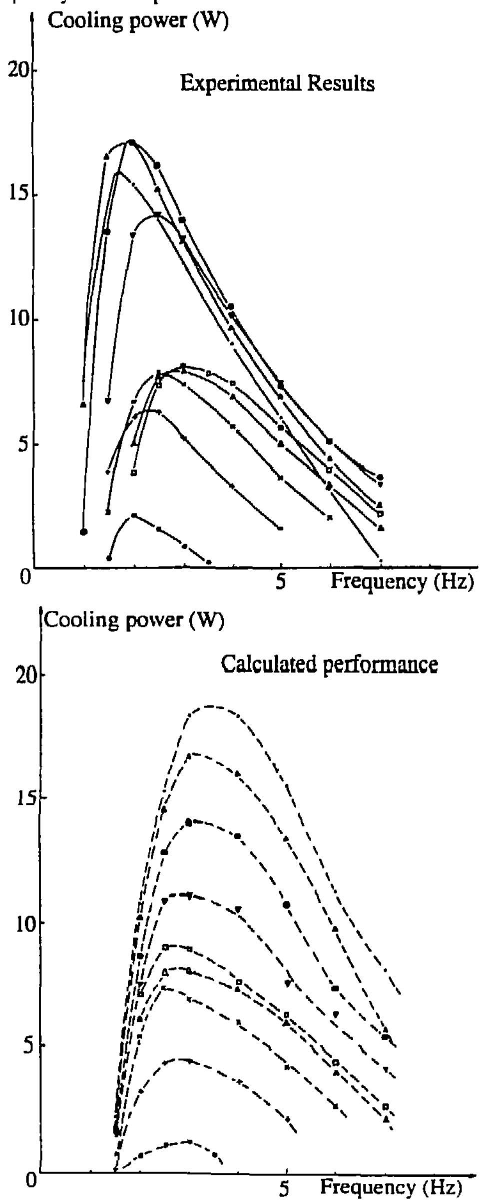

Figure 4: Comparison between experimental and calculated performance. The various curves correspond to different valves opening

This model validated by the experiment will be an efficient tool for further design and optimisation of pulse tube refrigerators.

MINIATURE PULSE TUBE COOLERS

Miniature Stirling coolers are presently widely used for infrared detectors cooling. The strong requirement for high reliability has lead to many technological improvements : linear motor drives, clearance seals, frictionless bearings. But Stirling cryocoolers still have two moving parts : the pressure oscillator piston and the cold finger displacer. In the pulse tube refrigerator the moving displacer has been eliminated conferring a potential for lower cost, higher reliability and less vibration.

In a preliminary attempt to validate these assumptions we have developed a miniature pulse tube associated with a pressure oscillator adapted from a commercial oil free piston air compressor.

Some characteristics and performances of this prototype operating in an orifice pulse tube configuration are summarised in the following table.

Oscillator swept volume :	8 cm^3
Pulse tube length/diameter :	10 cm / 10 mm
Regenerator length/diameter :	10 cm / 16 mm
Mean pressure :	1.5 MPa
Frequency of operation :	25 Hz
Ultimate temperature :	80 K
Typical cooling power :	1 W @ 110 K

Mean pressure and pressure oscillations were limited by the rotating motor torque and the by pass flow in the piston/cylinder clearance, but high frequency operation was demonstrated.

Later on, in the framework of a collaboration with CRYOTECHNOLOGIES S.A., different miniature double inlet pulse tubes have been designed and tested with a standard Stirling cooler pressure oscillator.

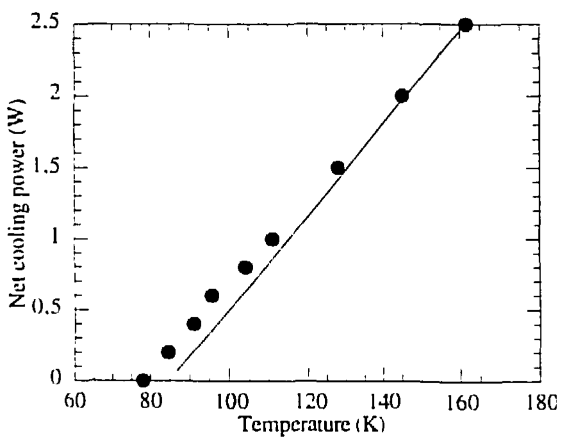

Figure 5: Experimental cooling power of a miniature pulse tube prototype. The full line corresponds to the calculated performance

Figure 5 shows as an example the net cooling power versus the cold tip temperature for one of these prototypes. The good agreement between experimental and theoretical data validates our previously described model in a very different frequency and cooling power range. The main characteristics of the pulse tube configuration corresponding to these results are summarised here after.

Pressure oscillator swept volume :	2.1 cm^3
Pulse tube/regenerator length/diameter :	7 cm / 5 mm
Mean pressure :	3.5 MPa
Operation frequency :	20 Hz
Ultimate temperature :	76 K

Further work is being done with this pressure oscillator to estimate the performances of a coaxial geometry for the pulse tube and regenerator. Efficiency measurements are also planned recording PV diagram with a modified oscillator.

FUTURE WORK

The preliminary results presently available and reported here gives us a great confidence in the potentialities of pulse tube refrigeration. New developments are currently underway in our laboratory.

The further step in the improvement of the "Stirling like" pulse tube refrigerator is the use of a linear pressure oscillator. The necessary resonant operation will induce a constraint on the sizing and optimisation of the system which will be included in our model. This work is presently underway and a double inlet pulse tube will be soon associated with a commercial linear Stirling pressure oscillator. The use of highly efficient compressor developed for space borne applications is also to be considered.

"Gifford Mac Mahon like" (i.e. using helium compressor associated with a distributor) pulse tube are also under development. In the low frequency range the goal is to achieve large cooling power at intermediate temperature or low ultimate temperature with multistaged systems. Recently in a preliminary test, a multistaged pulse tube using magnetic material for regeneration has reached a 5 K temperature. Concurrently a theoretical work has been initiated to modelise the thermal behaviour and efficiency of the regenerator in a temperature range [20 K - 4 K] where specific heat of the regenerative material exhibits a sharp anomaly and the helium no more behaves as a perfect gas.

CONCLUSION

A development program on pulse tube refrigeration has been undertaken at CEA/SBT. A numerical model has been developed and successfully validated against experimental data obtained with large capacity/low frequency (16 W/80 K at 3 Hz) and miniature high frequency (1 W/100 K at 25 Hz) prototypes. This numerical tool now available for design and optimisation is currently being used for new developments. These developments include coaxial geometry pulse tube associated with resonant linear pressure oscillators, or low temperature multistaged systems.

The association of a pulse tube with a pressure oscillator developed for long life space cryocooler will certainly contribute to simplify and improve the integration and reliability aspects.

REFERENCES

1 Gifford W.E. and Longsworth R.C. Trans. ASME J. Eng. Ind. (1964) 63, p. 264.
2 Wheatly J., Hoffer J., Swift G.W., Migliori A. Am. J. Phys. (1985) 53, p. 147.
3 Richardson R.N. Cryogenics (1986) 26, p. 331.
4 Longsworth R.C. Adv. Cryo. Eng. (1967) 12, p. 608.
5 Mikulin E.I., Tarasov; A.A. and Shkrebyonock M.P. Adv. Cryo. Eng. (1984) 29, p.629
6 Radebaugh R., Zimmermann J. Smith D.R., Louie B. Adv. Cryo. Eng. (1986) 31, p. 779.
7 Storch P.J. and Radebaugh R. Adv. Cryo. Eng. (1988) 33, p. 851.
8 Zhu Shaowei, Wu Peiyi and Chen Zhungti. Cryogenics (1990) 30, p. 514.
9 Zhu S.W., Wu. P.Y. and Chen Z.Q. Cryogenics (1990‡ 30, p. 257.
10 Tward E., Chan C.K. and Burt W.W. Adv. Cryo. Eng. (1990) 35, p. 1207.
11 Liang J. Development and Experimental Verification of a Theoretical Model for Pulse Tube Refrigeration. Thesis (1993), Grenoble.
12 Kays and London. Compact Heat Exchangers. Mc Graw Hill Series in Mechanical Engineering.