

CFD Simulation of a Thermoacoustic Cooler with RHVT Effect

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ABSTRACT

High frequency and pressure operation conditions are the usual approach for thermoacoustic devices of reduced size and practical feasibility. As a result, heat is generated by viscous losses and vorticity, mainly around the stacks. Many different topologies of stacks have been analyzed to reduce these effects, and also helium or mixtures He/Ar are employed for this purpose.

This article evaluates the effect of incorporating a Ranque-Hilsch Vortex Tube (RHVT) into a thermoacoustic cooler device as a way to reduce the dependence on the stack and mimic the behavior of the traveling wave arrangement in a reduced space.

INTRODUCTION

Thermoacoustic

Numerical simulation has been the first approach to design thermoacoustic devices, as it is practical, inexpensive and reliable to predict results with an accuracy close to experiments. Wang gives a detailed explanation of how to use ANSYS in the design and analysis of thermoacoustic refrigeration devices [1].

Among the most difficult elements to simulate are the stacks, because of the big difference in the length scale of reference. They are considered as critical elements, as they are located where the interchange between thermal and acoustical energies takes place. Many numerical, theoretical and experimental efforts have been dedicated to analyzing the length of the stack, the position in the resonant cavity, the diameter and geometry of the pores, fibers, homogeneous and heterogeneous porous media, even the materials from which they are made [2-7].

Two main characteristic lengths have been determined,

$$\delta_k^2 = \frac{2k_f}{\rho_m c_p \omega} \quad , \quad \delta_v^2 = \frac{2\nu}{\omega} \tag{1}$$

known as the thermal penetration depth and the viscous penetration depth, respectively. They are important design parameters for stacks and optimize the performance of any thermoacoustic device. By example, it has been found that a plate spacing of $3\delta_k$ in the stack gives the optimum performance for thermoacoustic devices [4]; the heat conduction is relevant if the channels have lengths between 10-20 times δ_v [5,6].

The magnitude $\sigma = \delta_v^2 / \delta_k^2$, known as the Prandtl number, should be low to increase the COP performance, and the lowest value is presented for monoatomic gases with 2/3, but even lower values can be realized with mixtures such as helium-argon, which present a minimum for pressures under 4.5 MPa at a ratio 4:6; for pressures over 4.5 MPa, mixtures of helium-argon-xenon work better [8,9,10].

Concerning the position of the stack, it has been determined that the optimum matches with the zeros of the derivative function ΔT evaluated at the resonance of the system [11]. But also there are proposals for devices with no-stack; this means the heat exchangers are close enough, between the average wavelength, to allow the parcels of gas to get into contact with both exchangers. Although some have found higher efficiencies than in stack-based systems, viscous losses at exchangers are increased compared with systems that consider stacks and exchangers [12,13].

Ejection

Ejectors were first used in 1858 as water pumps and to refill the reservoir of steam engine boilers. The use of ejectors for refrigeration and air conditioning goes back to 1910, allowing one to use low-grade thermal energy coming from steam engines [14]. Good improvements were realized with two-phase ejectors; however, ejectors were displaced when compressors came into being.

By the 1980's, ejectors were used to produce clusters of particles at very low temperatures, on the order of 50 K [15]. In recent years, a new interest has developed for applications in air conditioning and refrigeration systems driven by solar energy and waste heat [16].

Key factors of these systems is the geometry of the ejector and the properties of the mixture fluid. The first one determines the suction vacuum and the ratio gas/vapor, while the second one determines the rate of condensation and cooling.

RHVT Effect

Qualitatively, the working principle is to induce a strong whirlpool in a confined space, then the two opposite extremes of the tornado exit with hotter and colder temperatures than the entrance, respectively. While at the beginning, the key factor to realize significant cold temperatures was a good compressor, recently, with improved closed systems, lower pressures at the entrance are needed [17, 18]. Actually, RHVT is considered an active part in some pulse tube refrigerators [19, 20].

The present work considers a combination of these three effects to realize a compact refrigeration system (25 cm in diameter and 10 cm height) for use in freezers and water condensers. The designed operating temperatures are 25°C outside temperature of discharge and -10°C at the lower temperature. As the ejector is considered an acoustic diode in pulse tube refrigerators, its operation is explored under both continuous DC and oscillating AC conditions.

DESCRIPTION

Figure 1 shows a flow diagram of the proposed device. At the beginning, T_{mix} is the ambient temperature, but later, it is the average temperature of the discharges of the RHVT. Also, it is evident that the performance depends on the efficiency of the heat interchangers.

The compressor is a peripheral fan (not shown), running at about 750 rpm. It increases the pressure at the entrance of the nozzle up to 1.5 Bar over the reference pressure (T_o , P_o). At the same time it creates a light vacuum, reducing the ratio P_n/P_o , and then leading to a lower temperature (T_n , P_n), according to

$$T_n = T_o \left(\frac{P}{P_o} \right)^{(Y-1)/Y} = T_o \left(1 + \frac{Y-1}{2} M^2 \right)^{-1} \quad (2)$$

$$\text{where } T_o = \frac{P_o}{\rho_o R} \quad (3)$$

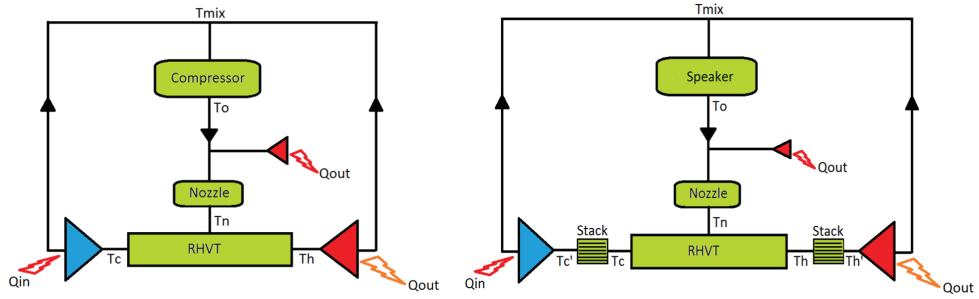


Figure 1. Flow diagram of the proposed device. Working on continuous operation (left), two effects are considered: Ejection and RHVT. Under oscillating flow (right), thermoacoustic is introduced.

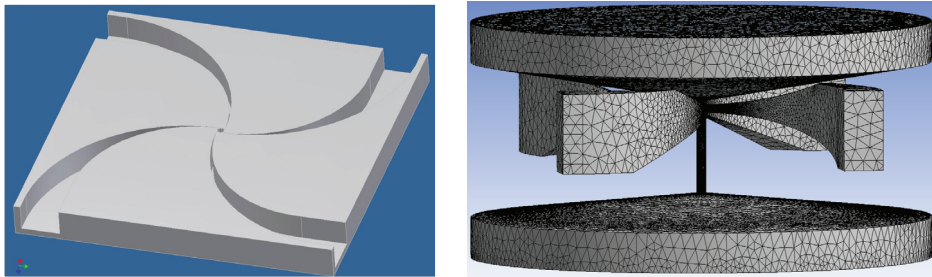


Figure 2. CNC sketch of the nozzle section (left), and mesh of the assembled proposed prototype (right).

If the temperature T_o is kept constant, then the temperature at the entrance of the RHVT is a few degrees below T_{mix} , while velocity reaches up to 110 m/s. This requires a compression at constant temperature and a heat rejection of

$$Q_{out1} \sim -RT_o \ln \frac{P_o}{P_{mix}} \quad (4)$$

A sketch in CNC of the nozzle is shown in Fig. 2a. Four nozzles at all converge to form the whirlpool.

At the RHVT section, the distribution of temperatures is given approximately by [17]

$$T_n = \mu_c T_c + (1 - \mu_c) T_h \quad (5)$$

$$\mu_c = \frac{\dot{m}_c}{\dot{m}_n} \quad (6)$$

where \dot{m}_n and \dot{m}_c are the mass flow at the entrance and the cold side, respectively. It was chosen $\mu_c \sim 0.4$, which is the optimum for maximum $\Delta T = T_h - T_c$ [17].

At the exit of the RHVT, the flow is directed to both heat exchangers and sent to the peripheral, where is suctioned by the compressor and the cycle initiates again. Figure 2b shows the mesh and an assembled view of the nozzle, RHVT and heat exchangers.

The fluid is a mixture of Ar and N_2 at a rate 9:1.

Under continuous flow, the coefficient of performance is approximately given by

$$COP = \frac{Q_{out1} \frac{T_c}{T_o} \left(1 + \frac{T_o}{T_h} \frac{Q_{out2}}{Q_{out1}} \right)}{RT_o \ln \left(\frac{P_o}{P_{mix}} \right)} \quad (7)$$

For oscillating conditions, the compressor is replaced by speakers and stacks are introduced, although the right position and dimensions are still under analysis.

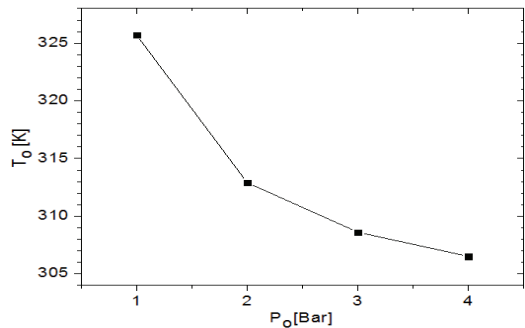


Figure 3. Rate of increasing temperature by compression before the entrance to the nozzle. It corresponds to the heat Q_{out1} that must be rejected.

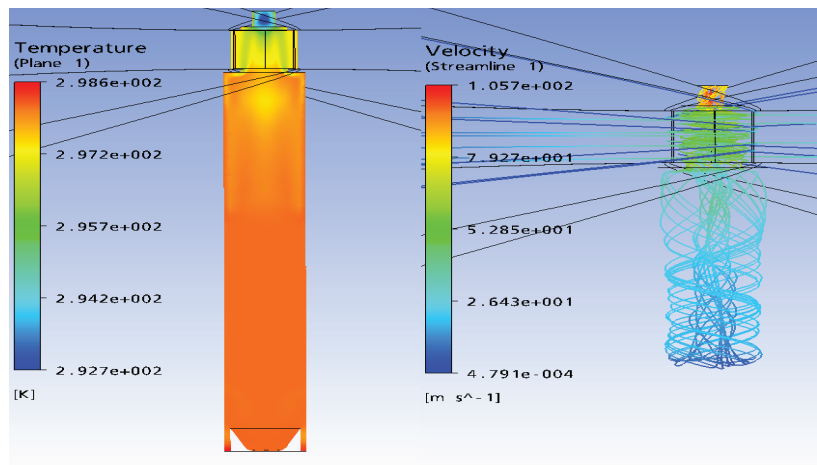


Figure 4. CFD results for the RHVT section of the prototype. It was assumed $T_{amb} = 25^{\circ}C$, $V_{in}=0.1m/s$ and $P_{ref} = 1Bar$.

RESULTS AND DISCUSION

So far, simulations show a light decreasing dependency of T_o on the pressure P_o (Fig.3); this means that when we compress from $(T_{amb}, P_{ref}) \rightarrow (T_o, P_o)$, the heating is lower. This is due to the fact that T_o depends on the ratio P_o / P_{ref} and not on the magnitude. But this indeed implies a high power compressor.

Something similar happens with T_n ; it depends on the ratio P_n / P_o , and is on the order of 13K in all cases.

At the RHVT section, the developed velocity and T_n at the entrance determine the values of T_c and T_h , almost entirely. Figure 4 shows the results of a CFD analysis with a velocity of 0.1m/s at the entrance.

CONCLUSIONS

There is still a need to analyze many variables to correctly optimize the prototype, but so far there is confidence it is reliable. Our better results show it is possible to reach 243 K at T_c with reasonable operation conditions.

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REFERENCES

1. Jian-Xin W., Z. Xiao-Yan, "Research of thermoacoustic refrigerator design based on thermoacoustic effect numerical simulation," *Proc. of AIMSEC-IEEE* (2011), pp. 4216 - 4219.
2. Hendrikus P., *Mathematical Aspects of Thermoacoustics*, PhD thesis, Technische Universiteit Eindhoven, Netherlands, 2009.
3. Humaira-Tasnim S., *Porous Media Thermoacoustic Stacks: Measurements and Models*, PhD thesis, University of Waterloo, 2011.
4. Wei-Hsin L., *Investigation of edge effects in thermoacoustic couple measurements*, MSc thesis, Naval Postgraduate School, 1990.
5. Tijani M.E.H., Zeegers J.C. and de Waele A.T., "The optimal stack spacing for thermoacoustic refrigeration," *Journal of the Acoustical Society of America*, vol.112, no.1 (2002), pp. 128-133.
6. Hamilton M.F., Ilinskii Y.A. and Zabolotskaya E.A., "Thermal effects on acoustic streaming in standing waves," *Journal of the Acoustical Society of America*, vol.114, no.3092 (2003).
7. Irwan-Shah A., Normah M.G., Jamaluddin A.R. Aminullah A.R.M., and Dairobi G.A., "Stack geometry effects on flow pattern with particle image velocimetry (PIV)," *Jurnal Mekanikal*, vol.12, No. 33 (2011), pp. 82-88.
8. Tijani M.E.H., Zeegers J.C.H. and de Waele A.T.A.M., "Prandtl number and thermoacoustic refrigerators," *Journal of the Acoustical Society of America*, vol. 112, no.1 (2002), pp. 134-143.
9. Belcher J.R., Slaton W.V., Raspet R., Bass H.E. and Lightfoot J., "Working gases in thermoacoustic engines," *Journal of the Acoustical Society of America*, vol. 105, no.5 (1999), pp. 2677-2684.
10. Hanbing K., Yaling H., Yingwen L. and Fuqing C., "Mixture working gases in thermoacoustic engines for different applications," *International Journal of Thermophysics*, vol.33 (2012), pp. 1143-1163.
11. Tae Kim Y. and Gon Kim M., "Optimum positions of a stack in a thermoacoustic heat pump," *Journal of the Korean Physical Society*, vol.36, no.5 (2000), pp. 279-286.
12. Wakeland R.S. and Keolian R.M., "Thermoacoustics with idealized heat exchangers and no-stack," *Journal of the Acoustical Society of America*, vol.111, no.6 (2002), pp. 2645-2664.
13. Behrang A. and Matveev K.I., "Numerical modeling of thermoacoustic no-stack heat pumps and prime movers," *Engineering Applications of Computational Fluid Mechanics*, vol. 6, no.3 (2012), pp. 346-355.
14. Elbel S. and Hrnjak P., "Ejector refrigeration: An overview of historical and present developments with an emphasis on air-conditioning applications," *Proc. of International Refrigeration and Air Conditioning Conference*, Purdue e-Pubs (2008).
15. Sattler K., Reyes Flotte A., Echt O. and Recknagel E., "Dimerization of Zn vapor in a supersonic carrier gas," *Journal of Applied Physics*, vol. 53, no.3 (1982), pp. 1317 - 1319.
16. Jianlin Y., Hua Ch., Yunfeng R. and Yanzhong L., "A new ejector refrigeration system with an additional jet pump," *Applied Thermal Engineering*, vol.26, (2006), pp. 312-319.
17. Oliver R., *Numerical Prediction of Primary and Secondary Flows in a Ranque-Hilsch Vortex Tube*, PhM thesis, Dublin Institute of Engineering, 2008.
18. Sullivan Sh., *Refrigerator*, US Patent No. 7565808 B2 (2009).
19. Matthew P.M., *Pulse Tube Refrigerator*, US Patent No. 6109041 (2000).
20. Matthew P.M., *Double inlet arrangement for pulse tube refrigerator with vortex heat exchanger*, US Patent No. 6442947 B1 (2002).
21. Chistensen K.G., Heiredal M., Kauffeld M. and Schneider P., "Energy savings in refrigeration by means of a new expansion device," Journal No. 1223/99-0006, Internal report for the Energy Research Programme (2001).