*Paper No: 07-IAGT-2.4*

INDUSTRIAL APPLICATION OF
GAS TURBINES COMMITTEE

# SUSTAINABLE THERMOACOUSTIC REFRIGERATION SYSTEM FOR GAS TURBINE POWER PLANTS

**by**

**Hadi Babaei, Kamran Siddiqui**
**of**
**Department of Mechanical and Industrial Engineering, Concordia University, Montreal**

**and**

**Wajid A. Chishty**
**of**
**Gas Turbine Laboratory, Institute for Aerospace Research, National Research Council, Ottawa**

---

**Presented at the 17th Symposium on Industrial Application of Gas Turbines (IAGT)**
**Banff, Alberta, Canada – October 2007**

# Sustainable Thermoacoustic Refrigeration System for Gas Turbine Power Plants

Hadi Babaei, Kamran Siddiqui[*]
*Department of Mechanical and Industrial Engineering, Concordia University, Montreal*

and

Wajid A. Chishty
*Gas Turbine Laboratory, Institute for Aerospace Research, National Research Council, Ottawa*

## Abstract

Trigeneration is a process to simultaneously produce power, heat and cooling from a single consumable fuel source. A typical gas turbine-operated hybrid trigeneration cycle produces refrigeration by either using electric power via a compressor or using low grade waste heat in an absorption chiller. A new sustainable refrigeration system based on the thermoacoustic technology is proposed for the trigeneration system applications. This refrigeration system involves no hazardous refrigerants and utilizes waste heat from the gas turbine engine to produce acoustic power, which is then utilized as the refrigerator work input. A thermoacoustic refrigeration system is designed and numerically evaluated. Preliminary results show that the thermoacoustic refrigeration system is capable of achieving an overall theoretical efficiency of 36% with a refrigeration COP of 2.35.

---

[*] Corresponding author: Department of Mechanical and Industrial Engineering, Concordia University, Montreal. Tel: (514) 848-2424 x 7940. Email: siddiqui@encs.concordia.ca

## Introduction

With the increasing need for energy, there is a growing concern about its over-use and impact on the environment and natural resources. As a result, there is an enhanced interest in seeking alternative and sustainable energy sources as well as improving the efficiency of energy producing systems. While the debate on the net benefits of alternative energy sources on environment continues between many governmental and industrial quarters, there seems no disagreement on developing energy efficient and sustainable energy systems to conserve depleting energy resources.

The performance of energy utilizing systems plays a vital role in the consumption of energy as well as emission of hazardous pollutants. Conventional power plants convert only 30% of the fuel energy into electric power. The rest is lost in the form of heat. In order to enhance the efficiency of power production, cogeneration of power and process heat has been adopted in many industries, where up to 80% of fuel energy can be converted into usable forms. Trigeneration is an extension to a cogeneration process and implies the simultaneous production of power (mechanical/electric), heat and cooling from a single consumable fuel source. It is also referred to as combined heat, cooling and power (CHCP) production. In a trigeneration system, a portion of power or heat production is used to produce refrigeration.

Figure 1 shows a typical gas turbine-operated hybrid trigeneration topping cycle process. Refrigeration may be obtained by either using electric power via a compressor or using low grade waste heat in an absorption chiller. Hybrid refrigeration systems can change thermal and electrical load profiles of a plant by shifting refrigeration from an electrical load to heating load during high cooling seasons and when high time-of-day electrical rates are enforced. This effective utilization of process energy reduces the energy production cost, reduces energy consumption and cuts down net emissions.

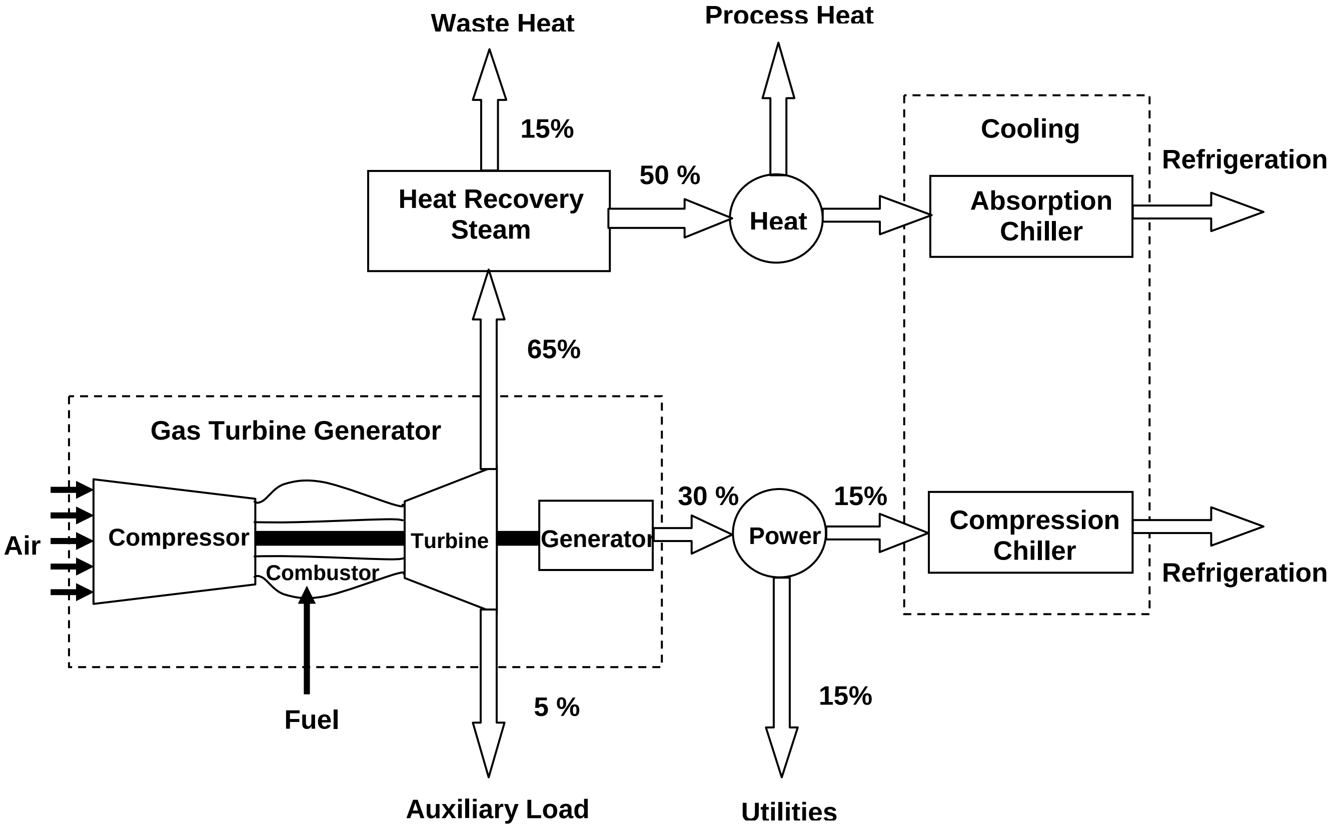

**Figure 1: Gas turbine operated hybrid trigeneration topping cycle process**

Compression and absorption refrigerators are both vapor cycle systems, where refrigeration takes place by evaporating a liquid (refrigerant) with a sub-zero boiling point. In both cases, when the liquid evaporates it takes away heat from its surrounding. The difference between the two systems is in the method used to liquefy the vapor refrigerant. A conventional refrigerator uses an electrically driven compressor to pressurize the vapors into liquid. An absorption refrigerator on the other hand uses heat energy to accomplish the same task in an absorber. In some cases limited mechanical energy may also be consumed for pumping refrigerant.

Compression refrigeration is three to five times more efficient than absorption system. However, as shown in Fig 1, compression process may consume 50% of the produced electric power. On the other hand absorption refrigeration is obtained using heat which might otherwise be wasted. The drawback in this case is the complex processes involved in the recovery of liquid refrigerant.

Both compression and absorption refrigeration systems use refrigerants that are hazardous to environment. CFC losses from compression refrigeration are about 25% annually and produce greenhouse effects 3000 times greater than that of carbon dioxide (equivalent to 1.4 megatonnes of $CO_2$ emissions in Canada). Similarly ammonia which is used as the refrigerant in absorption refrigeration system is highly toxic.

Due to the limitations of the cooling technologies presently being utilized in trigeneration systems, there is a need to develop a sustainable and environmentally benign refrigeration system that can be easily integrated in the existing CHCP cycles. Thermoacoustic refrigeration possesses the potential to fulfill this mandate.

Recent advancement in the field of thermoacoustics has revolutionalized the way many conventional devices operate. Thermoacoustics deal with the conversion of heat energy to sound energy and vice versa. The device that converts heat energy into sound energy is called thermoacoustic heat engine and the device that converts sound energy into heat energy is called thermoacoustic refrigerator. A complete thermoacoustic refrigeration system can be created by combining the thermoacoustic heat engine and thermoacoustic refrigerator. The heat engine takes heat as an input from a thermal source and delivers sound as an output. This sound is then taken as an input by the refrigerator, which pumps heat from a cold reservoir to a hot reservoir. If the heat source to the thermoacoustic engine is waste heat e.g., from the gas turbine power plant, the complete system may then be a source of “sustainable” refrigeration.

There are many advantages of this refrigeration system as compared to the conventional refrigeration systems. One advantage of these devices is low fabrication cost which is due to their inherent simplicity, in spite of some technical design challenges [1]. These devices have no moving parts. As a result there are no chances of mechanical failures and hence, they are more reliable. Furthermore, no harmful refrigerants are required. Air or any inert gas can be used as the working fluid. Thus, these devices are also environmentally friendly. From an overall plant perspective, since the sustainable system utilizes waste heat to generate the required refrigeration power, the overall plant efficiency would also be enhanced.

The discovery of the thermoacoustic phenomenon dates back to more than a century, but the significant work in this area was started about two decades ago at the Los Alamos National Laboratory by Swift and co workers. They developed different types of thermoacoustic refrigerators and heat engines [1, 2]. Since then a few other research groups in US, Netherlands, UK, Japan and China are also working in this area and have developed thermoacoustic refrigerators mostly for cryogenic applications. For example, Garret *et al.* [3] developed a new spacecraft cryocooler, which used resonant high-amplitude sound waves. This cryocooler was used in the space shuttle Discovery. Tijani *et al*. [4] achieved temperature as low as -65 $^{o}$C in their thermoacoustic refrigerator. Jin *et al.* [5] used a thermoacoustic engine to drive a thermoacoustic refrigerator and obtained the refrigeration temperature as low as 120 K.

In the absence of a thermoacoustic heat engine, an acoustic driver may be used to provide work input to the thermoacoustic refrigerator. However, in this configuration, the energy input will be in the form of electrical power. Almost all of the thermoacoustic refrigerators previously developed either used a speaker as an acoustic input or used electric heating as an input for the heat engine. Thus, these devices could not be considered as sustainable. Recently, some efforts have been made to develop heat engines that operate on waste heat. Symko *et al*. [6] designed and developed a thermoacoustic heat engine that utilizes heat from a microcircuit to produce sound. Hatazawa *et al.* [7] proposed a heat engine that utilizes waste heat from a four-stroke automobile gasoline engine. More recently, Zoontjens *et al.* [8] theoretically studied the feasibility of using thermoacoustic refrigerator for automotive air conditioning that operates on the waste heat from the automobile engine. They argued that the replacement of automotive vapor-compression refrigeration system with the thermoacoustic refrigeration system has economical and environmental benefits. Their analysis showed that the device could produce cooling power equal to 10% of the waste heat input.

## Working Mechanism

A thermoacoustic heat engine consists of a tube (resonator) with a gas medium and a stack of thin and closely spaced plates of specific length, which is placed at a specific location inside the resonator. An electric heating coil or a heat exchanger is attached to the stack to transfer heat to the stack. Another heat exchanger is attached to the other end of the stack to create a temperature gradient across the stack (see Fig. 2a). The principle of thermoacoustic heat engine is relatively simple. This temperature gradient across the stack interacts with the Brownian motion of the gas molecules inside the stack, thus causing them to emit sharp pressure pulses at the other end of the stack, which excite a sound wave in the resonator [9]. Thus, if a temperature gradient is continuously maintained across the stack, sound will be continuously generated inside the tube. It is important to note that the temperature gradient of the gas parcels must be less than the temperature gradient across the stack to produce acoustic power (see Fig. 2d).

In a similar manner, a stand-alone thermoacoustic refrigerator consists of a gas-filled resonator tube, a stack of thin and closely spaced plates and heat exchangers at both ends of the stack (see Fig. 2b). A half-wavelength standing acoustic wave is created inside the tube when sound input is provided from one end, with the other end kept closed. The length of the tube is set corresponding to a preferred resonance frequency and the speed of sound in the given gas. The acoustic wave generates pressure differential and causes oscillations in the gas parcel displacement with the maximum pressure amplitude at the closed ends of the resonator and minimum pressure amplitude in the middle of the resonator. When the stack is placed inside the tube, the gas parcels will be at a higher pressure near that edge of the stack, which is closer to the closed-end of the resonator (i.e. left end of the stack in Fig. 2b). When the gas parcels move towards the other end of the stack (i.e. away from the closed-end of the tube) under the action of the standing acoustic wave, their pressure decreases and thus they cool off. The temperature of the gas parcels is lower than that of the stack, as a result, heat is transferred from the stack to the gas. The temperature at this end (i.e. the right end) of the stack is imposed by the cold heat exchanger attached to this end. When the gas parcels move back, their pressure and thus, temperature increase. The temperature of gas parcels is now higher than that of the stack and the heat is transferred to the stack. The temperature at this end (i.e. the left end) of the stack is imposed by the hot heat exchanger attached to this end (see Fig. 2c). Thus, by back and forth motion (i.e. oscillation) of the gas parcels, heat is pumped from the cold-end heat exchanger fluid to the hot-end heat exchanger fluid. [1]. Thus, the heat is transferred from a cold medium to a hot medium and the device works as a refrigerator. It should be noted that for a thermoacoustic device to work as a refrigerator and for the gas parcels to transfer heat from one end of the stack to the other, the temperature gradient of the gas parcels must be greater than that across the stack (see Fig. 2e).

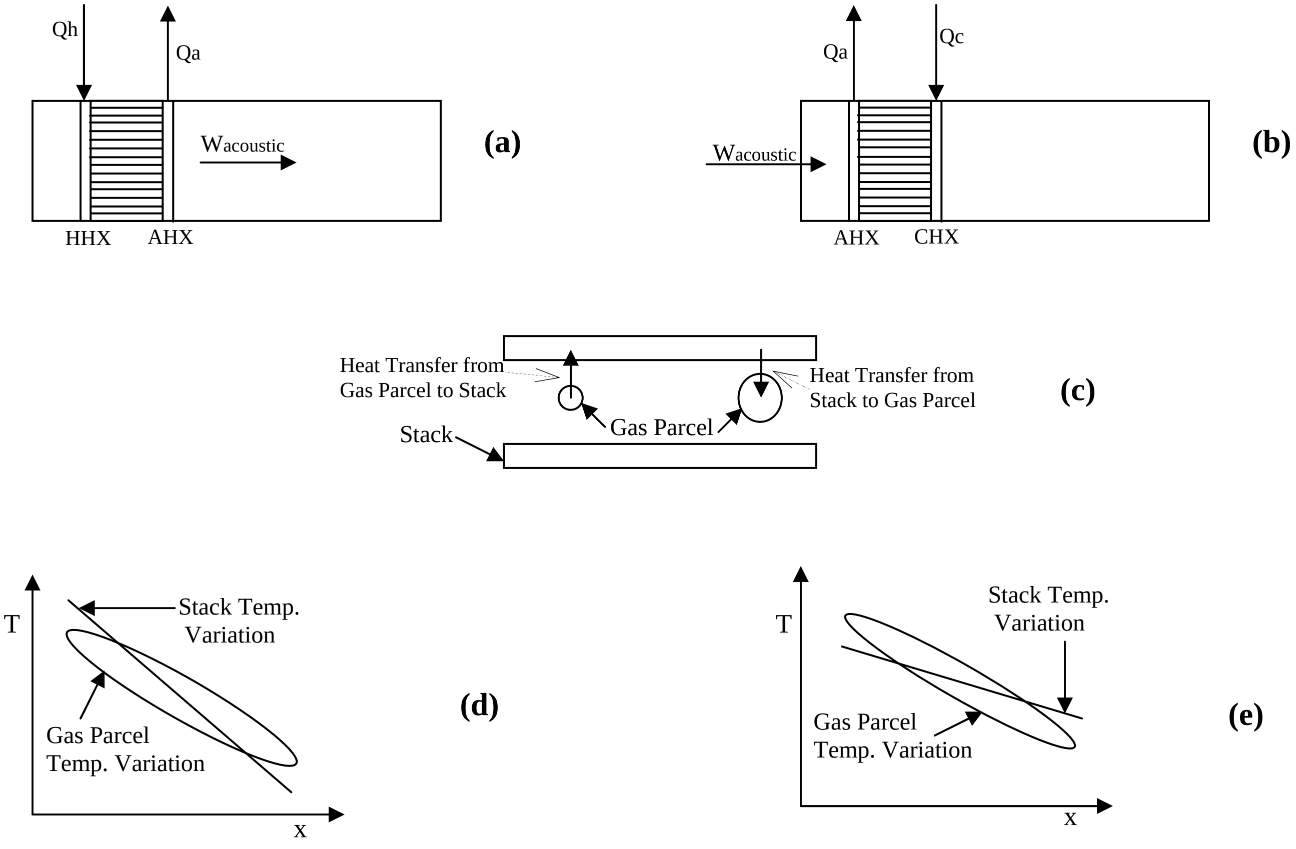

**Figure 2: Schematic of thermoacoustic (a) heat engine, (b) refrigerator, (c) heat transfer process in the refrigerator stack. The temperature variation in the stack of (d) heat engine, (e) refrigerator.**

A complete thermoacoustic refrigeration system may be created by combining the thermoacoustic heat engine and thermoacoustic refrigerator (dash-lined box in Fig. 3). That is, the heat engine takes heat as an input from a thermal source and delivers sound as an output. This sound is then taken as an input by the refrigerator, which pumps heat from a cold reservoir to a hot reservoir.

The most important components of a thermoacoustic device are the stacks and the heat exchangers, which facilitate heat transfer to and from the working fluid. The design and characteristics of these components therefore, have a significant impact on the performance of a thermoacoustic device.

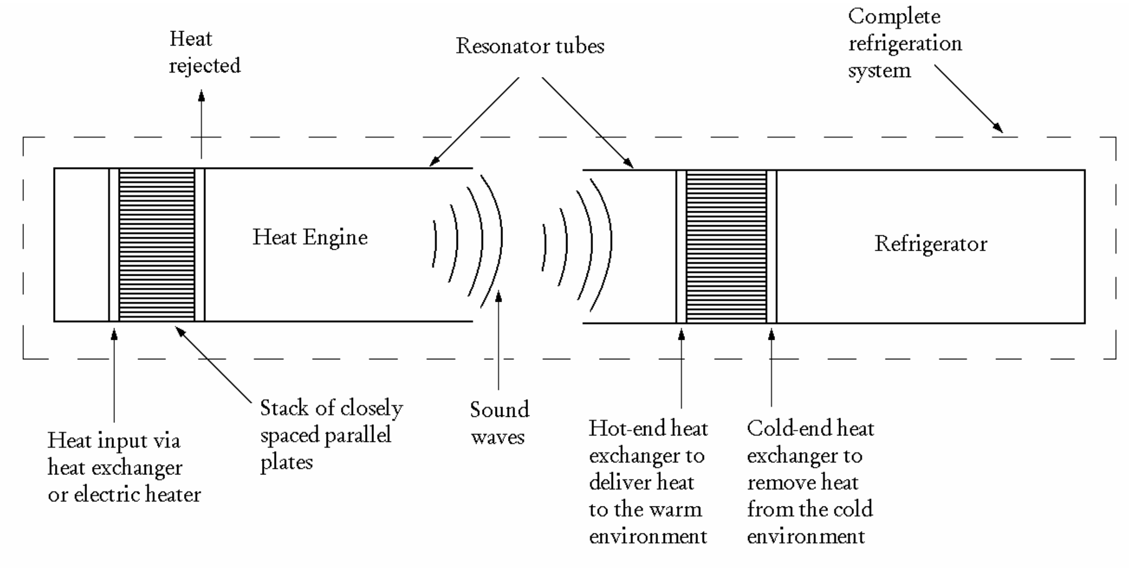

**Figure 3: Schematic of a combined thermoacoustic heat engine and thermoacoustic refrigerator.**

## Numerical Modeling

The theory of thermoacoustics is primarily based on a low amplitude linearization of the Navier-Stokes equation with sinusoidal oscillations of all variables [10]. The governing thermoacoustic equations are derived from the linearized Navier-Stokes, continuity and energy equations. The first governing equation is the wave equation for the pressure when the temperature gradient exists along the stack. The second governing equation is the time-average energy flux equation and the third equation is called the acoustic power equation [1]. These equations indicate that there are different sets of parameters that play important roles towards the performance of a thermoacoustic system. These parameters include thermophysical properties of the working gas and stack, resonance frequency, mean pressure and pressure variation of the working gas, position and length of the stack, temperature difference across the stack, stack plate spacing and plate thickness. Nineteen independent parameters could be identified from these equations when designing a thermoacoustic refrigerator. When coupled to a thermoacoustic engine the number of independent parameters increases further, thus making the design task even more challenging.

For the linear range of acoustic waves, the research group of Gregg Swift at Los Alamos National Laboratory has developed a computer code called “Design Environment for Low Amplitude Thermoacoustic Engines” or more commonly “DeltaE”, to numerically optimize the design of a thermoacoustic device. It is a powerful tool to design and optimize a thermoacoustic device and to predict its performance. The code solves the one dimensional wave equation in a gas or liquid based on the low amplitude acoustic approximation in a user defined geometry [11].

## Design Optimization and Results

Meeting the required cooling power at the desired cooling temperature is the main goal of any refrigeration system. To optimize a heat-driven thermoacoustic refrigeration device, the main components that must be designed and optimized are, the resonator, the refrigerator stack and its associated heat exchangers, and the engine stack and its associated heat exchangers.

In the present study, a thermoacoustically-driven thermoacoustic refrigerator is designed considering a refrigeration baseload of 1 kW. All previously reported thermoacoustic refrigerator designs (theoretical and experimental) are based on the cooling power less than 100 W with the exception of one by Wollan *et al.* [12], where a cooling load of 7 kW was considered with the intentions for cryogenic applications. The present system requires 2.8 kW of heat input from a waste heat source for providing 1 kW of cooling power. DeltaE was used to optimize the design and to simulate the performance. Helium was used as the working gas since it has the highest thermal conductivity and highest speed of sound amongst all inert gases. The resonance frequency of the acoustic wave is an important design parameter. Higher resonance frequency results in lower penetration depth i.e. small plate spacing in the stack which increases the manufacturing challenge, however, higher resonance frequency increases the acoustic power and reduces the length of the resonator. Different resonance frequencies were considered in the range 100-500 Hz. The resonance frequency of 400 Hz was selected where the stack plate spacing is considered relatively simple from manufacturing stand point. The corresponding length of the resonator is therefore 1.27 m (i.e. half of standing wavelength). Simulations were performed at different resonator tube pressures. Results shown in the paper are for a mean pressure of 1 MPa. The acoustic waves are assumed to be in the linear range. The drive ratio, which is defined as the ratio of the pressure oscillation amplitude to the mean pressure, was set equal to 0.03. The temperature of ambient heat exchangers was assumed to be 305 K. The axial length of the cold heat exchanger was considered to be 2 mm while the width of hot heat exchangers was considered to be 3 mm. The geometry of the stack was chosen to be of parallel plate type, as it is relatively easy to fabricate compared to other stack geometries and it has the best theoretical efficiency compared to all geometries except the pin array [13]. By considering the stack plate spacing equal to three times the thermal penetration depth and taking into account the blockage ratio and mean temperature of the stacks; the engine stack plate spacing was found to be 0.57 mm with the plate thickness of 0.2 mm and the refrigerator stack plate spacing was 0.36 mm with the plate thickness of 0.12 mm.

To achieve the cooling power of 1 KW, the lengths and positions of both stacks were also optimized. The optimum length and position of both stacks were obtained using an optimization algorithm similar to that reported by Wetzel and Herman [14]. The optimum stack length of the refrigerator was found to be 1.6 cm with the center position of the stack located at 4 cm from the right end of the resonator. The optimum stack length of the engine was 4 cm and the center position of stack was located at 4 cm form the left end of the resonator. The optimized cross sectional area of the resonator was found to be 0.25 m$^2$. The detailed design of the system is presented in Fig. 4. The figure shows that the thermoacoustic heat engine received 2.765 kW of heat from the waste heat source and rejected 2.240 kW of heat to the ambient. The heat rejected through the ambient heat exchanger includes the heat associated with the dissipation of acoustic power in both heat exchangers of the engine. The thermoacoustic refrigerator removed 1 kW from the cold environment and delivered 1.424 kW to the warm ambient environment.

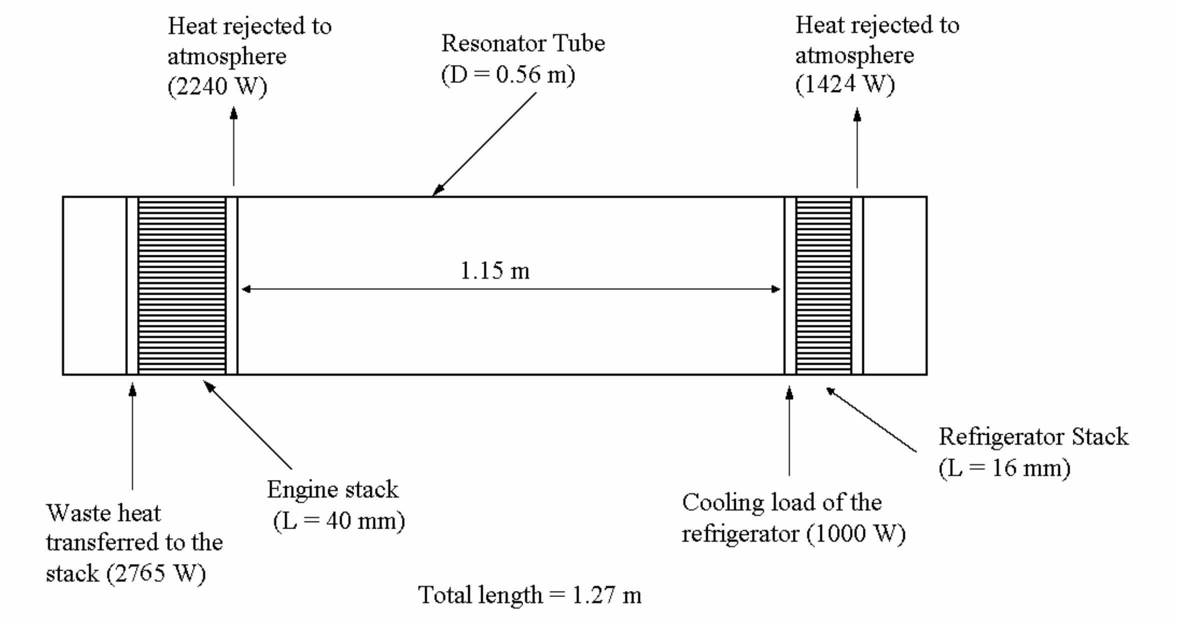

**Figure 4: Schematic of the designed thermoacoustic refrigeration system.**

DeltaE was also used to predict the temperature, pressure and velocity variations and the acoustic power along the length of the resonator. The variation in the acoustic power is shown in

Fig. 5. The dashed lines in the left section marks the location of the heat engine stack and the dashed lines in the right section marks the location of the refrigerator stack. The plot shows that acoustic power is generated within the stack of the heat engine and is then utilized in the stack of the refrigerator. The detailed analysis shows that the designed thermoacoustic heat engines produces 686 W of the acoustic power in the heat engine stack. However, out of 686 W of the available acoustic power, 161 W was dissipated in the two heat exchangers attached to the stack of the heat engine. About 72 W of acoustic power was also dissipated within the resonator tube between the two stacks and 433 W of acoustic power was delivered to the refrigerator. Inside the thermoacoustic refrigerator, 169 W of the acoustic power was dissipated in the two heat exchangers attached to the stack of the refrigerator and 255 W was utilized as the work input to the refrigerator to provide the cooling power of 1 kW.

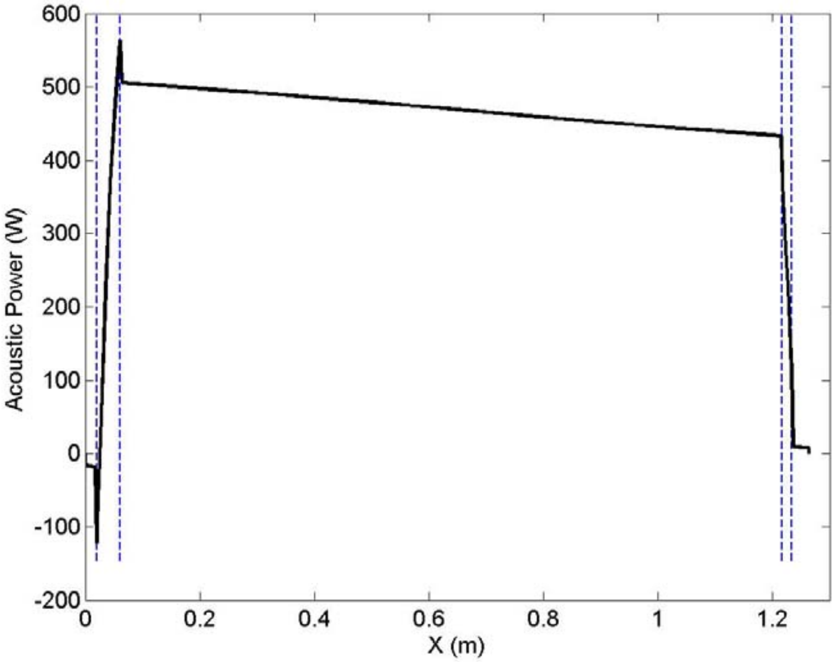

**Figure 5: The variation of acoustic power along the resonator. Dashed lines in the left section marks the location of the heat engine stack and the dashed lines in the right section marks the location of the refrigerator stack.**

The temperature distribution in the resonator is depicted in Fig. 6. In the preliminary design, the temperature at which the waste heat is delivered to the stack is considered to be 818 K. The ambient air is considered to be at 305 K. The plot shows that the temperature of the gas in the region from the heat engine stack to the left end of the resonator is almost constant at 818 K. As the hot and ambient heat exchangers attached to the stack of the heat engine are maintained at their set temperatures, the temperature difference generated across the stack is 515 K. At the stack of the refrigerator, the cold end is maintained at 278 K while the hot end is at the ambient temperature of 305 K. The plot also shows that in the section between the two stacks, the temperature gradually decreased from 305 K at the ambient end of the heat engine stack to 278 K at the cold end of the refrigerator stack.

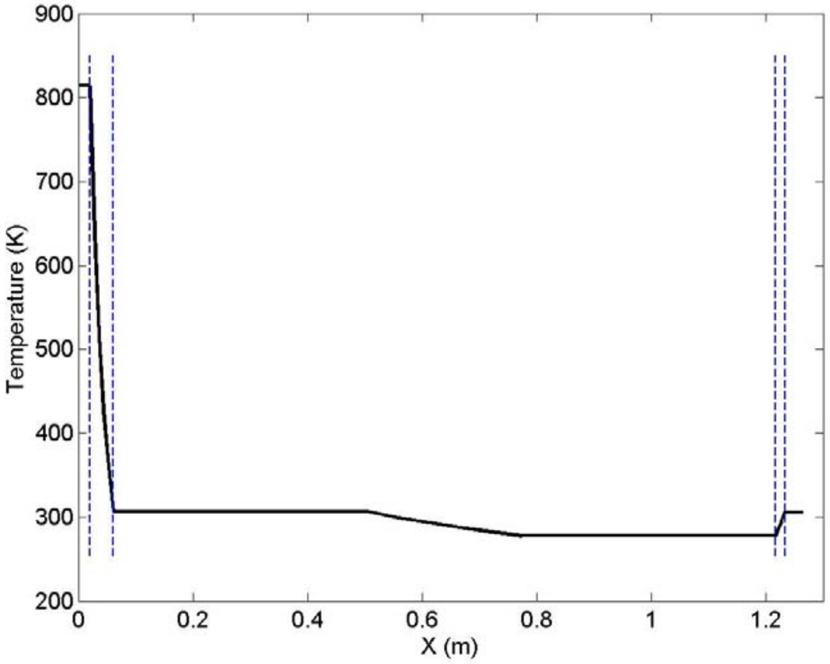

**Figure 6: Variation of the temperature along the resonator.**

The magnitudes of pressure and velocity variations along the resonator are shown in Fig. 7 which is similar to the typical pressure and velocity profiles of a half-wavelength acoustic standing wave. At the two ends of the resonator tube, the acoustic velocity is zero (i.e. the velocity node) and the pressure is maximum (i.e. the pressure antinode). In the middle of the channel, the pressure fluctuations are minimum and the acoustic velocity is maximum. The maximum pressure amplitude is about 32 kPa which is around 3% of the mean pressure. It should also be noted that the hot end of the stack in both heat engine and refrigerator are at higher pressure than the other end of the stack.

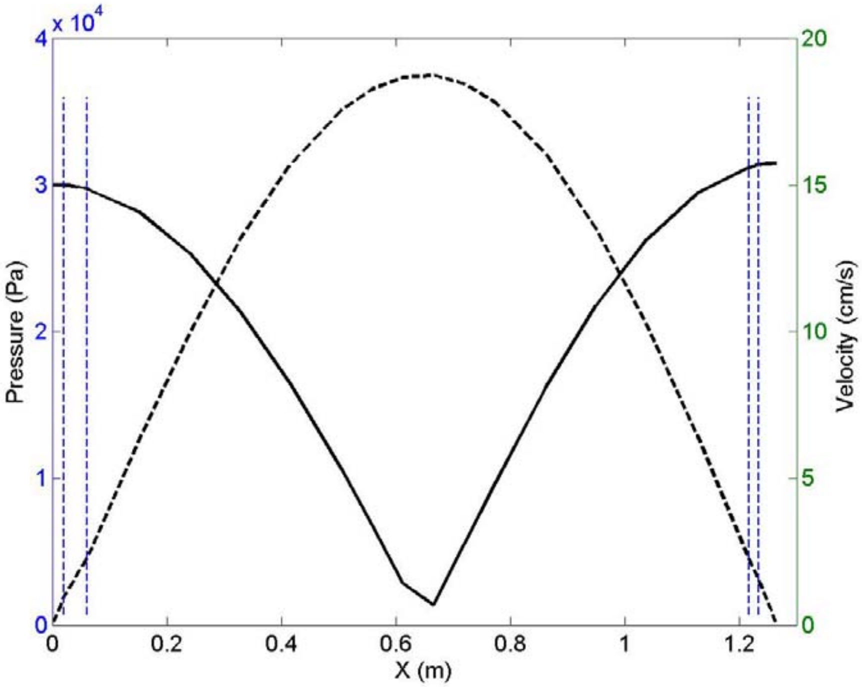

**Figure 7: Variations of pressure and velocity along the resonator. Solid line, pressure; dashed line, velocity.**

The overall efficiency of the complete thermoacoustic system can be evaluated based on the amount of waste heat input to the system (i.e. 2.765 kW) and the produced cooling power (i.e. 1 kW) which in the present case is 36.2%. Taking into account the power dissipated by the heat exchangers, the thermal efficiency of the engine and the coefficient of performance of the refrigerator are estimated to be 19% and 2.35, respectively.

The presented parametric analysis demonstrates that a thermoacoustic refrigeration system can be incorporated in gas turbine-operated trigeneration systems. Operating on the waste heat, the overall efficiency of the plant may be enhanced by 5%. In addition, the system is sustainable, cost effective and environmentally friendly.

## Concept Validation

Although the results of theoretical feasibility studies are very promising, the thermoacoustic refrigeration technology however, is in the early stages of its development and significant experimental and additional analytical efforts are needed to develop efficient systems. Collaborative efforts between academia and industry are also required to bring this promising technology closer to maturity.

The research group at the Concordia University is actively involved in the research activities in the field of thermoacoustics/acoustics. An experimental facility has been established for acoustic velocity and pressure measurements using state-of-the-art measurement techniques including Particle Image Velocimetry (PIV). A low power thermoacoustic refrigerator has been fabricated. The images of the device are shown in Fig. 8. The refrigerator is being investigated to determine the effects of heat exchanger geometry on the cooling load [15]. In addition, experimental and numerical investigations of nonlinear acoustic wave dynamics are being conducted, which are very crucial in the design of thermoacoustic devices operating at high-amplitude pressure oscillations [16, 17, 18]. A project has also been initiated in collaboration with National Research Council Canada to design and fabricate a prototype sustainable thermoacoustic refrigeration system based on waste heat from different sources.

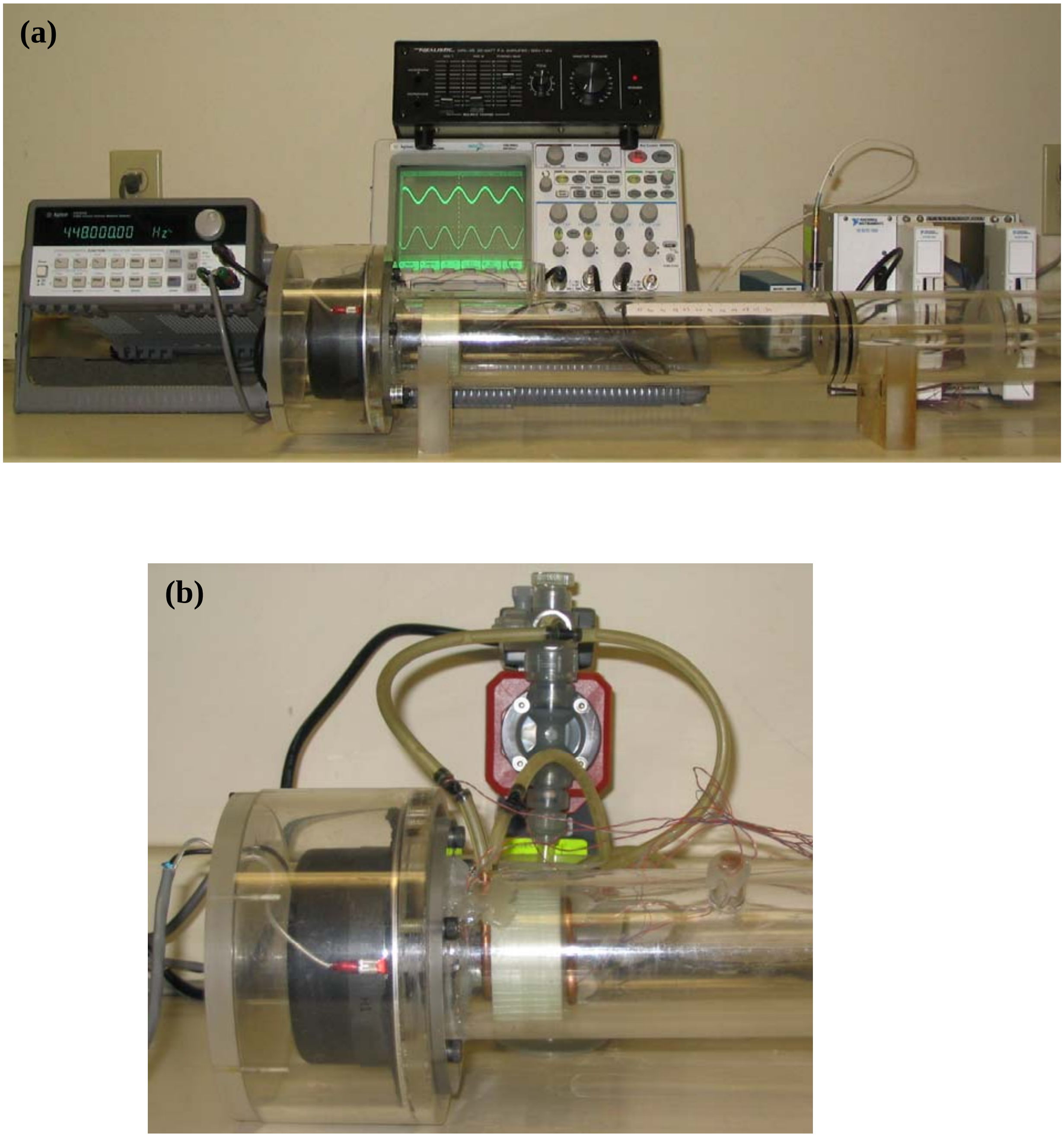

**Figure 8: Thermoacoustic refrigerator developed at the Concordia University (a) complete view without heat exchangers, (b) partial view with the heat exchangers attached to the stack.**

## Conclusion

A new refrigeration system is proposed for a gas turbine-operated trigeneration system that is based on the thermoacoustic technology. The thermoacoustic refrigeration system utilizes waste heat from the plant to produce acoustic power. This acoustic power is then utilized as the work input to the refrigerator. The thermoacoustic refrigeration system is simple in design and reliable. Furthermore, it is environmentally friendly as no hazardous gases are used in the system, unlike the compression and absorption refrigeration systems currently used in the trigeneration systems.

A thermoacoustic refrigeration system was designed and analytically evaluated using DeltaE code from Los Alamos National Laboratory. The parametric analysis shows that the designed system receives 2.765 kW of waste heat and produces 1 kW of the cooling power with the overall efficiency of 36%. The coefficient of performance of the designed system was found to be 2.35. The present study demonstrates that the thermoacoustic refrigeration system has the ability to enhance the overall efficiency of a trigeneration system by 5%.

## References

1. Swift, G.W. “Thermoacoustics: A unifying perspective for some engines and refrigerators”, The Acoustical Society of America, Melville, NY, 2002.
2. Swift, G.W. “Thermoacoustic engines and refrigerators”, *Physics Today* **48**, 22-28, 1995.
3. Garrett, S.L., Adeff, J.A. and Hofler, T.J. “Thermoacoustic refrigerator for space applications”, *J. Thermophysics Heat Transfer* **7,** 595-599, 1993.
4. Tijani, M.E.H., Zeegers, J.C.H. and De Waele, A.T.A.M. “Construction and performance of a thermoacoustic refrigerator”, *Cryogenics* **42,** 59-65, 2002.
5. Jin, T., Chen,G.B., Wang, B.R. and Zhang, S.Y. “Application of Thermoacoustic Effect to Refrigeration”, *Review of Scientific Instruments* **74**, 677-679, 2003.
6. Symko, O.G., Abdel-Rahman, E., Kwon, Y.S., Emmi, M. and Behunin, R., “Design and development of high-frequency thermoacoustic engines for thermal management of microelectronics.” *Microelectronics J.*, **35**, 185-191, 2004.

7. Hatazawa, M., Sugita, H., Ogawa, T. and Seo, Y., "Performance of a thermoacoustic sound wave generator driven with waste heat of automobile gasoline engine." *Trans. Japan Soc. Mech. Eng.* **70**. 292-299, 2004.
8. Zoontjens, L., Howard, C., Zander, A. and Cazzolate, B. "Feasibility Study of an Automotive Thermoacoustic Refrigerator", Proceedings of *Acoustics*, Busselton, Australia, November 9-11, 2005.
9. Symko, O.G. "Energy Conversion Using Thermoacoustic Devices", Proceedings 18th International Conference on Thermoelectric, Baltimore, Maryland, August 29 – September 2, 1999.
10. Rott, N. "Thermoacoustics", *Adv. Appl. Mech.* **20**, 1235-175, 1980.
11. Ward, W.C. and Swift, G.W. "Design environment for low-amplitude thermoacoustic engine." *J. Acoustic Soc Am.* **95**, 3671-3674, 1994.
12. Wollan, J.J., Swift, G.W., Backhaus, S. and Gardner, D.L. "Development of a thermoacoustic natural gas liquefier", Proceedings of AIChE Meeting, New Orleans, LA, March 11-14, 2002.
13. Pat-Arnot, W., Bass, H.E. and Raspeect, R. "General formulation of thermoacoustics for stack having arbitrary shaped pore cross sections." *J. Acoust. Soc Am.* **90**, 3228-3237, 1991.
14. Wetzel, M. and Herman, C. "Design optimization of thermoacoustic refrigerators." *Int. J. Refrig.* **20**, 3-21, 1997.
15. Akhavanbazaz, M., Siddiqui, K., and Bhat, R.B., "The Impact of Gas Blockage on the Performance of a Thermoacoustic Refrigerator", *Exp. Thermal Fluid Sci* (In press), 2007.
16. Nabavi, M., Siddiqui, K. and Dargahi, J. "Simultaneous measurements of acoustic and streaming velocities using synchronized PIV technique." *Meas. Sci Technol.* **18**, 1811-1817, 2007.
17. Nabavi, M., Siddiqui, K. and Dargahi, J. "Measurement of the acoustic velocity field of nonlinear standing wave using synchronized PIV technique." *Exp. Fluids* (Submitted), 2007.
18. Nabavi, M., Siddiqui, K. and Dargahi, J. "A fourth-order accurate scheme for solving highly nonlinear standing wave equation in different thermoviscous fluids." *J. Comp. Acoustics* (Submitted), 2006.