Coaxial thermoacoustic Stirling integral system: Cooler driven by a two-stage engine

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Abstract

Environmental implications and increasing energy cost demand improvement of the energy efficiency of the industrial processes. Significant reductions in energy consumption are expected by using heatpumps which can operate at waste heat temperature and upgrade waste heat to process heat temperature. At ECN, thermoacoustics is one of the technologies being investigated to develop such a heatpump. This paper addresses a thermoacoustic integral Stirling system consisting of an engine and a heatpump. The engine consists of two stages in order to reduce the operating temperature so as to facilitate use of waste heat at lower temperatures. The two stage engine drives a heatpump which functions as a cooler. The engine and the cooler are housed in a half wavelength resonator. The objective of this investigation is to study the interaction between the engine and the cooler. Measurements and analysis of the performance of the two stage engine and the integral system are presented. The no-load, lowest temperature reached by the cooler is about -40 °C. This is obtained when the maximum permissible heat (about 200 W for first unit and 240 W for second unit) is applied to the electric heaters in the engine. The maximum efficiency of the engine relative to Carnot is 20.7 % when operating at a drive ratio of 5.0 % and a temperature of 396 °C. The maximum COP relative to Carnot of the cooler is 30.0 % operating at a temperature of 278 K and a drive ratio of 3.0 %. Gedeon streaming in the engine and the cooler is suppressed using an elastic latex membrane near the ambient heat exchangers.

1. Introduction

In the Dutch industry, significant demand for energy is heat in the form of steam at several pressure levels. In 2005, the heat used in the industry was about 581 PJ. At the same time, a large part of the heat is released into the environment as "industrial waste heat". Clearly the reuse of this waste heat will result in large energy savings and contribute to both benchmark agreements and the Kyoto protocol.

The problem with the reuse of waste heat is the mismatch between demand and supply in terms of temperature, time and location of heat. Solving the mismatch in temperature will lead to the most economic benefits because the heat is available at the same location and can be applied directly without the need for expensive infrastructure. The technology needed for this solution is a heat pump, so that the temperature level of heat can be raised above the pinch and the 'upgraded' heat can be useful again. Existing heat pumps are not capable of operating at the desired temperature level nor generate the desired temperature lift. Systems based on thermo-acoustic technology are in principle capable of fulfilling the above criteria.

Thermoacoustics refer to the technology of converting heat into acoustic power and vice versa. This technology is used to contruct engines and heatpumps. The working principle of a thermoacoustic Stirling system is similar to the conventional Stirling system. The essential difference being the mechanical moving parts in a conventional Stirling system are replaced by acoustic components. The absence of moving parts in thermoacoustic system is attractive because of the lower maintenance costs. At ECN, thermoacoustic engines and heat pumps are being developed in order to upgrade the waste heat to useful process heat temperature. A thermoacoustic system for upgrading waste heat essentially consists of a thermoacoustic engine which produces acoustic power from waste heat, a heatpump which consumes acoustic power to upgrade waste heat to process heat, and a resonator which contains the acoustic wave. A thermoacoustic engine operating at an efficiency of 49 % relative to Carnot value has been recently achieved in our laboratory [1]. In this engine, the produced acoustic power is dissipated in a dummy acoustic load. Simultaneously, a cooler which is driven by a linear motor is also developed that operates at an efficiency of 45 % relative to Carnot [2]. The next step is to integrate the engine and the heatpump.

A two-stage coaxial thermoacoustic Stirling engine driving a coaxial cooler is described in this paper. High performance thermoacoustic engines reported by Backhaus [3], Tijani [4] and others usually have their feedback inertance engineered in a torus configuration. The difference in the coaxial and torus engine is the manner in which the feedback inertance is engineered. In the coaxial configuration, the regenerator unit is mounted inside the resonator. The annular gap between the regenerator unit and the resonator wall forms the feedback inertance. In a torus configuration a separate duct acts as the feedback inertance. This facilitates easy insulation of hot parts of the engine. Also the flow losses in the feedback inertance in a torus configuration are smaller compared to a coaxial configuration. However, we choose coaxial configuration because of the ease with which adaptations can be made to the size of the engine and the cooler. The objective of this study is to understand the interaction between the engine and the cooler and not necessarily the thermal performance. Further, a two-stage engine is proposed because it lowers the operating temperature of the engine. In the next section, the design of the integral system is presented followed by the description of the experimental setup. The following section details the experimental results of the engine and the cooler.

2. Design

Figure 1 is the schematic of the proposed heat driven thermoacoustic integral system. The thermodynamic components of the engine and the cooler consists of an ambient heat exchanger, a regenerator, a hot heat exchanger or cold heat exchanger and a thermal buffer tube. A pressure sensor, mounted on the resonator near the cooler, is used to determine the drive ratio (P_1/P_0, P_1 is the amplitude of the oscillating pressure and P_0 is the mean pressure) in the system. The pressure sensors mounted near the centre of the resonator are used to make two-microphone measurements of the acoustic power flowing past their midpoint [5]. The ambient heat exchangers are of shell and tube type and water is used as the coolant. Heat is applied at the hot heat exchangers in the engine using an electric heater. An electric heater is also used at the cold heat exchanger in order to simulate a cooling load.

The integral system is designed using the DELTAEC code developed at LANL [6], which solves one dimensional thermoacoustic equations. In this model, the hot temperature of the two stages of the engine is forced to remain identical. Acoustic power losses in the feedback inertance are included in the model but heat leaks are not included. We anticipate substantial heat losses due to improper insulation of the hot parts which is inherent to coaxial systems (it is rather difficult to place insulation material around the hot parts because it might partially block the feedback inertance or absorb acoustic waves). The optimized dimensions of the integral system are given in Table 1. The diameter of the regenerator holders are chosen to be of equal size to simplify the construction. A 230 mesh wire screen material having a porosity of 78.6 % and a hydraulic radius of 27.6 μm is used as the regenerator material. Argon gas at a pressure of 15 bar is used because of the low sound velocity so that the length of the 1/2-wavelength is reasonably short. The resonator is designed in order to achieve an acoustic resonance of 60 Hz. The total length of the resonator is 1.82 m and it is made of stainless steel. The resonator consists of three sections as shown in Fig.1. The first and third section is a pipe of length 44 cm and an inside diameter of 9.55 cm. The center section is 79.4 cm long and 5.47 cm in diameter. Conical sections 7 cm long form the transition between the tubes.

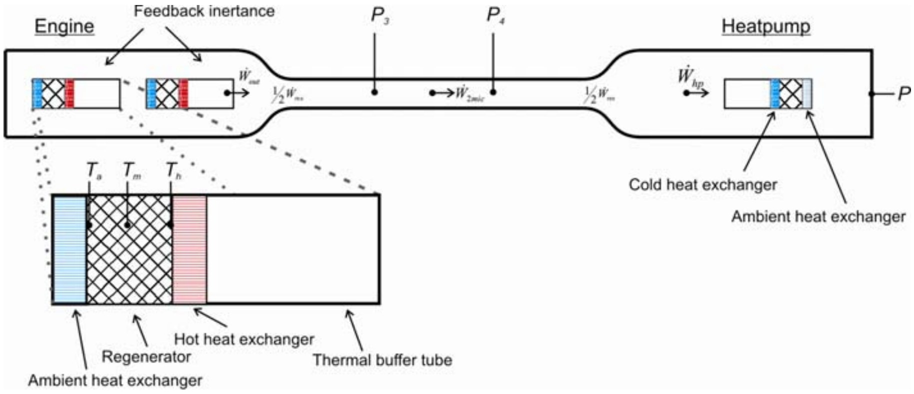

Figure 1 Schematic of the integral system.

3. Construction

The regenerator, thermal buffer tube and heat exchanger holders are made from stainless steel tubes. Figure 2 shows the assembled engine and the components used. A shell and tube heat exchanger with a diameter of 55 mm is used as an ambient heat exchanger. The tubes, which are parallel to the acoustic displacement, carry argon gas. The tubes have a diameter of 2 mm and a length of 15 mm. They are cooled by chilled water passing through channels perpendicular to the tubes. The inner diameter of the tubes for the gas flow is larger than optimal(the radius should be less than thermal penetration depth of gas which is about 0.1 mm). We anticipate degradation of the performance of the engine due to larger than acceptable temperature difference between gas and the water in the heat exchanger. To suppress DC or so-called Gedeon mass streaming [7], a thin latex membrane is attached as shown in figure 2. Two layers of nickel foam are used on either side of the heater (not shown in figure 2) to enable uniform flow of gas. Nickel foam is used at the end of the thermal buffer tube to provide smooth transition of flow at the junction and to avoid jet streaming. Thermocouples are inserted at the centre and either ends of the regenerator. Care is taken to place the tip of each thermocouple in the circular centre of the regenerator holder. Aluminium tape is used generously to plug tiny holes and at the joints of the components.

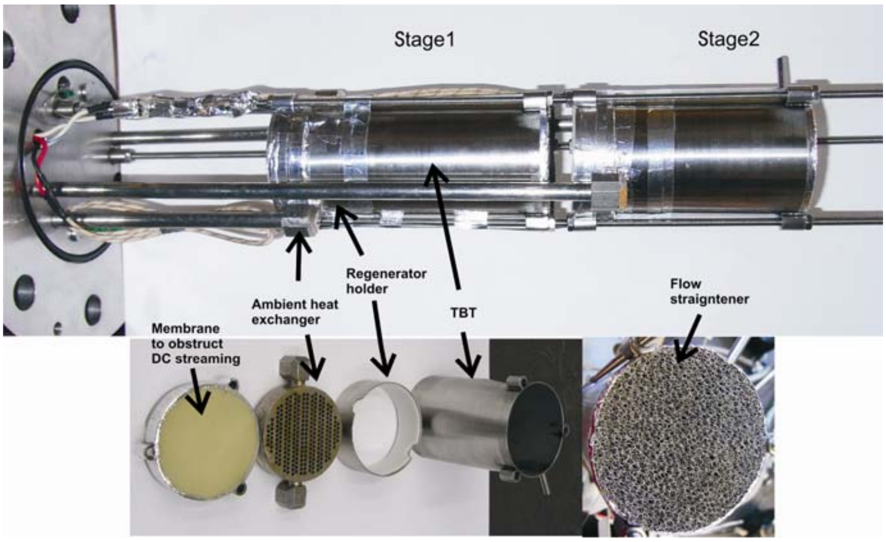

Figure 2. Two stage engine with assembled components

Table 1 Dimensions of some of the components of the two-stage engine and the cooler.

		Engine		Cooler
		Stage 1	Stage 2	
Regenerator:				
	Diameter (cm)	5.5	5.5	5.5
	Length (cm)	1.5	1.3	2.0
Thermal buffer tube:				
	Diameter (cm)	5.7	5.7	5.7
	Length (cm)	8.1	5.3	7.5
Feedback inertance:				
	Area (cm^2)	43.1	43.1	43.1
	Length (cm)	13.0	10.0	14.0

4. Experimental results

In this section, the performance of the engine and the cooler in the integral system is presented. At first the procedure used to determine the resonator losses is described followed by the engine and cooler performance. In figure 1, the two microphones located at the centre of the resonator are used to determine the acoustic power $\dot{W}_{2mic}$ flowing through the midpoints of the two microphones.

$$\dot{W}_{2mic} = \frac{A}{2\omega\rho_g \Delta x}\left[\left(1-\frac{\delta_v}{r}\right)p_3 p_4 \sin\alpha + \frac{\delta_v}{2r}\left(p_3^{\,2} - p_4^{\,2}\right)\right] \tag{1}$$

where P_3 and P_4 are the amplitudes of the dynamic pressures measured by the two pressure sensors, Δx is the distance between the two transducers along the resonator, α is the time phase difference, ω is the angular frequency, ρ_g is the density of the gas, δ_v is the viscous penetration depth, r the duct radius and A the gas cross-sectional area.

In order to determine the resonator loss $\dot{W}_{res}$ the cooler is removed from the system shown in figure 1. For several values of heat input to the engine , the drive ratio and the 2-microphone power are measured. Assuming acoustic losses are same on both sides of the resonator from the centre location, the resonator loss is equal to two times the measured 2-microphone power,

$$\dot{W}_{res} = 2\left(\dot{W}_{2mic}\right) \tag{2}$$

Figure 3 is the plot of $\dot{W}_{res}$, as a function of the square of the drive ratio for a mean pressure of 15 bar argon. The data is clearly not linear indicating enhanced dissipation at higher drive ratios. This is corroborated in experiments at higher drive ratios when the middle and the conical sections of the resonator are noticed to be warm. The data in figure 3 is used to determine the acoustic power input to the cooler and the acoustic power output of the engine.

The performance of the cooler is given by the coefficient of performance, *COP* which is the ratio of the cooling power to the input acoustic power

$$COP = \dot{Q}_c / \dot{W}_{hp} \tag{3}$$

The maximum *COP* of the cooler operating between cold temperature T_c and ambient temperature T_a is given by the Carnot value which is equal to $COP_{CARNOT} = T_c / (T_a - T_c)$. The performance of the cooler relative to the maximum Carnot value is equal to

$$COP_r = COP / COP_{CARNOT} \tag{4}$$

The thermal efficiency of the engine η_{engine} is the ratio of the acoustic power output of the engine $\dot{W}_{engine}$ to the total heat input $\dot{Q}_h$ to the engine.

$$\eta_{engine} = \dot{W}_{engine} \big/ \dot{Q}_h \tag{5}$$

The total heat $\dot{Q}_h$ is equal to the sum of the heat input to the two stages of the engine. The efficiency of the engine relative to the Carnot efficiency is,

$$\eta_r = \eta_{engine} \big/ \eta_{Carnot} \tag{6}$$

The Carnot efficiency η_{Carnot} is equal to $1-T_a/T_h$, where T_a is the ambient temperature of the regenerator and T_h is the hot temperature of the regenerator. The acoustic power input to the cooler $\dot{W}_{hp}$ is determined from the 2-microphone power and resonator losses and is equal to,

$$\dot{W}_{hp} = \dot{W}_{2mic} - \frac{1}{2}\dot{W}_{res} \qquad 7$$

Similarly, the acoustic power out of the engine $\dot{W}_{engine}$ is equal to,

$$\dot{W}_{engine} = \dot{W}_{2mic} + \frac{1}{2}\dot{W}_{res} \qquad 8$$

Results

At first, the static heat leak in the engine and the cooler is determined by supplying heat with the electrical heater. A heating power of 18.6 W is consumed by the cooler to maintain a temperature difference of 75 °C. The cooling water temperature and the ambient temperature are both about 22 °C. Assuming the variation of regenerator thermal properties is negligible, a heat of 18.6 W leaks to the cooler when operating at -53 °C. The static heat leak in the engine (both stages) is about 61 W at a temperature of 100 °C. The heat leak in the engine and the cooler is rather large for such a system because the warm or cold parts could not be insulated.

In all the dynamic experiments described below, the electrical heater voltage is regulated to maintain the same temperature at the hot end of the two stages of the engine. The second stage of the engine always consumes higher electrical power and hence more heat than the first stage because the acoustic impedance at the second stage is lower than the first stage (the second stage is farther away from the acoustic velocity node than the first).

The cooling experiments are carried out for several drive ratios. For each drive ratio, the cooler is loaded by supplying heat with the electrical heater at the cold heat exchanger. The drive ratio in the system is maintained by regulating heat to the engines. Figure 4 (a) show the cooling power and (b) acoustic power input to the cooler as a function of the cold temperature for several drive ratios. For a certain cold temperature, the cooling power (heat applied) and acoustic power increase with drive ratio because they are approximately proportional to the square of the drive ratio. Figure 4 (c) shows the calculated COP as a function of the cold temperature. The slope of the lines in the figure decrease with drive ratio.

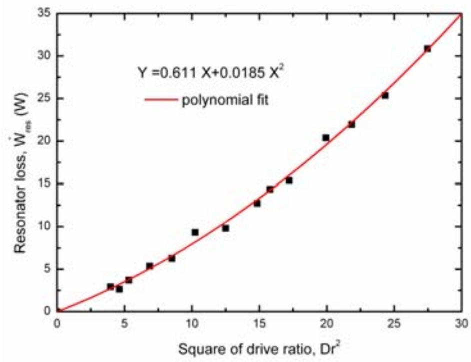

Figure 3. Resonator losses as a function of square of drive ratio for a mean pressure of 15 bar.

Comparison to figure 4 (b) it suggests that the acoustic losses would increase with drive ratio in the cooler. The minor losses in the feedback inertance and the sharp changes in gas motion inherent in coaxial systems could be a possible explanation for high acoustic losses. Figure 4 (d) shows the calculated relative COP_r as a function of cold temperature. The COP_r has a maximum value of about 30 % at a temperature of about 278 K and a drive ratio of 3 %. Figure 5 (a) gives the efficiency of the engine as a function of the temperature ratio between the hot and cold ends of the engine. The thermal efficiency is approximately flat for a certain drive ratio which is expected because the losses in the system are only proportional to the drive ratio. Figure 5 (b) gives the relative to Carnot efficiency of the engine as a function of the temperature ratio along the engine. The engine has a maximum efficiency of 20.7 % relative to Carnot when operating at a temperature of 396 °C, temperature ratio of 2.0 and a drive ratio of 5.0 %. At these operating conditions the no-load temperature of the cooler is -28.5 °C. For a drive ratio of 4.0 %, the engine operating temperature is 330 °C and has a relative to Carnot efficiency of 17.7 %. The no-load temperature of the cooler at these conditions is -32.7 °C. The maximum COP relative to Carnot of the cooler is 30.0 % operating at a temperature of 278 K and a drive ratio of 3.0 %. The no-load, lowest temperature measured at the cooler is about -40 °C. This is obtained when the maximum permissible heat (about 200 W for first unit and 240 W for second unit) is applied to the electric heaters in the engine.

Conclusion

A coaxial thermoacoustic Stirling engine driving a coaxial thermoacoustic cooler is constructed and tested. The engine has two stages in order to reduce the operating temperature. A thin membrane at the ambient heat exchanger is used to suppress Gedeon streaming. The coaxial configuration of the engine and the cooler allowed easy exchange of components and relatively simple design of the thermodynamic parts because they are housed inside the resonator which is the only pressure vessel. However, the heat and acoustic power losses in the coaxial configuration are considerable because the thermodynamic parts could not be well insulated and the gas undergoes several changes in the direction of flow. The no-load, lowest temperature reached by the cooler is about -40 °C. The maximum measured efficiency of the engine relative to Carnot is 20.7 % when operating at a drive ratio of 5.0 % and a temperature of 396 °C. The maximum COP relative to Carnot of the cooler measured is 30.0 % operating at a temperature of 278 K and a drive ratio of 3.0 %. Although the measured performance of the integral system is much lower than the high temperature engine [1] and linear motor driven cooler [2], the coaxial integral system performance is still acceptable for the purpose it is designed for: to study the interaction between the engine and the cooler. Higher performance is possible with the engine and the cooler having torus configuration, so that the hot and cold parts can be well insulated and the acoustic power losses are further reduced.

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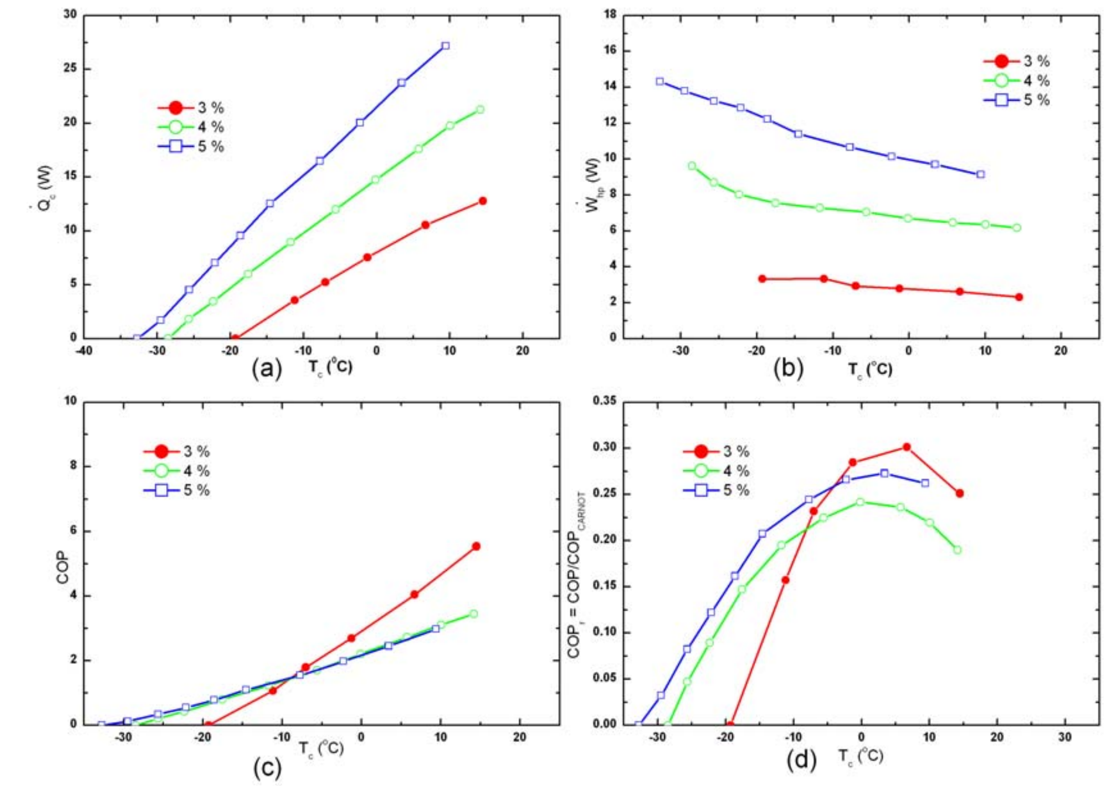

Figure 4 (a) Cooling power vs. the cold temperature of the regenerator. The no-load cold temperature decreases with drive ratio. (b) Acoustic power input $\dot{W}_{hp}$ to the cooler vs. the temperature at the cold end of the regenerator. $\dot{W}_{hp}$ increases with drive ratio for a certain temperature which is consistent with the model. (c) Calculated COP vs. the cold temperature. (d) The relative to Carnot COP vs. the cold temperature.

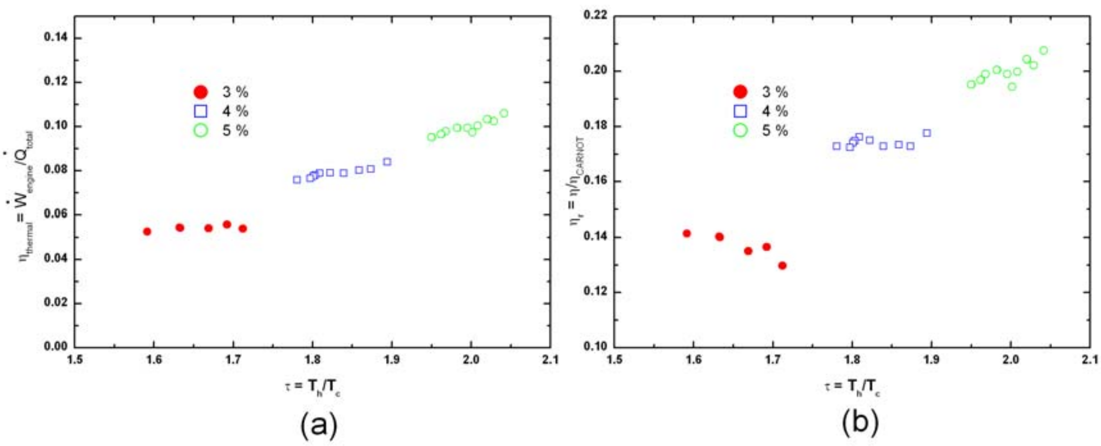

Figure 5 (a) The efficiency of the two stage engine vs. the temperature ratio at the ends of the regenerator. (b) The efficiency of the engine relative to Carnot vs. the temperature ratio at the ends of the regenerator.