THERMOAKOESTISCHE SYSTEMEN

4-stage thermo acoustic power generator

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Contents

1 Introduction

This document describes the design and construction of a prototype of a thermoacoustic water heater and power generator intended for rural areas, as commissioned by FACT Foundation. Chapter 2 gives a brief introduction to the thermoacoustic principles. The design and construction of the prototype is detailed in chapter 3. Experiments and test are described in chronological order in chapter 4 followed by the conclusions and recommendations in chapter 5.

2 Thermoacoustic energy conversion

This chapter briefly addresses the principles and basics of multistage travelling wave thermoacoustic engines.

2.1 Principle

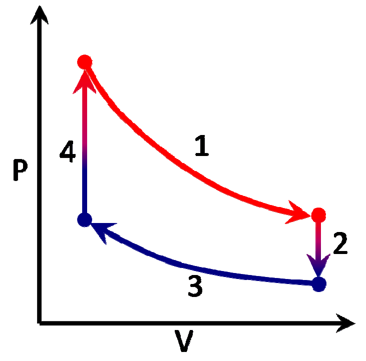
In thermoacoustic systems compression, expansion and displacement of the working gas is done by an acoustic wave rather than by pistons and displacers.

A sound wave is associated with changes in pressure, temperature, and density of the gas through which the sound wave propagates. In addition, the gas itself is moved around an equilibrium position. At the sound levels we experience in our daily lives, these fluctuations are small. By contrast, the pressure amplitude of the acoustic waves in thermoacoustic systems are extreme in magnitude and result in appreciable gas oscillations up to 10% of the mean pressure. If such acoustic waves interact with near isotherm porous structures with a much higher heat capacity than the working medium (gas), the acoustic wave will force the gas into a thermodynamic cycle. In an acoustic travelling wave this cycle is close to the Stirling cycle which potentially has a high conversion efficiency.

The idealized Stirling cycle consists of four thermodynamic processes acting on the working fluid (see diagram to right):

1. Isothermal Expansion. The expansion-space is heated externally, and the gas undergoes near-isothermal expansion.
2. Constant-Volume (known as isovolumetric or isochoric) heat-removal. The gas is passed through the regenerator, thus cooling the gas, and transferring heat to the regenerator for use in the next cycle.
3. Isothermal Compression. The compression space is intercooled, so the gas undergoes near-isothermal compression.
4. Constant-Volume (known as isovolumetric or isochoric) heat-addition. The compressed air flows back through the regenerator and picks-up heat on the way to the heated expansion space.

source: http://en.wikipedia.org/wiki/Stirling_cycle



In practice these four stages are overlapping and due to the sinusoidal movement of pistons and replacers the diagram follows more an circle or oval. While in “mechanical” engines the cycle is defined by the movement of the piston and displacer[1], in thermoacoustic devices the timing (or phase) between displacement and pressure variation is realized by gas movement in a passive acoustic resonance or feedback network in such a way that gas in the regenerator is successively being compressed, at the same time displaced, expanded, and displaced again. If a temperature gradient is applied across the regenerator, the gas will be heated at compression, and cooled at expansion as is depicted in Figure 1.

[1] having a 90° mutual rotational shift

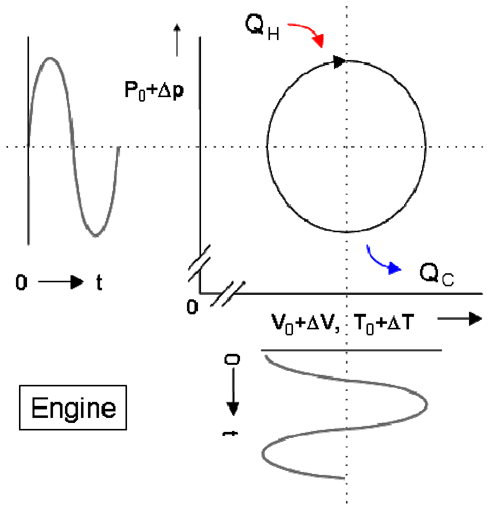


Figure 1 Thermodynamic cycle in a thermoacoustic engine.

As said, due to the acoustic wave motion gas is cycled along two isotherms (compression, expansion) and two isochors (heating, cooling). For a real acoustic impedance[2] this is similar to the Stirling cycle for which the (idealized) PV-diagram is shown before. The area within this shape is the amount of (acoustic) power that is produced from this cycle. As a result heat is converted into acoustic energy (engine). If the cycle is forced to run in the reverse way, heat is pumped from a low temperature to a high temperature (heat pump)

2.2 The thermoacoustic engine

Basically a thermoacoustic engine consist of a regenerator clamped between two heat exchangers. This so called "regenerator unit" is placed inside an acoustic resonance or feedback circuit. The acoustic feedback circuit also acts as a pressure vessel, as thermoacoustic devices are generally pressurized to increase output power and/or reduce size.

As seen in the acoustic propagation direction heat is supplied at a high temperature by the output heat exchanger (hex) and rejected at low (e.g. ambient) temperature by the input hex.

The acoustic circuit also includes an acousto-electric transducer or linear alternator for converting part of the acoustic energy into electricity. This is depicted schematically in Figure 2.

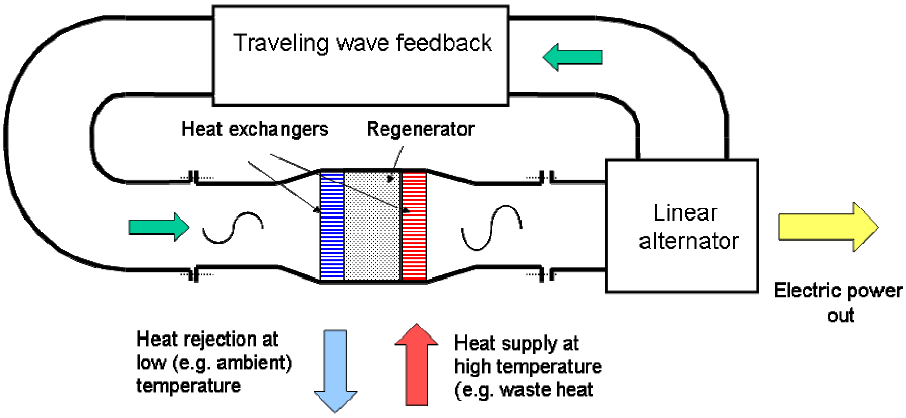


Figure 2 Basic configuration of a thermoacoustic engine with electric output

[2] A real impedance means that pressure amplitude and gas velocity are in phase. Gas displacement in that case is 90° out of phase.

The temperature difference between both hex will create a positive temperature gradient in the regenerator. As a result the regenerator unit will act as an acoustic power amplifier. The acoustic power gain of this "amplifier" (G_{TA}) is equal to the ratio of absolute in- and output temperatures (T_{High} and T_{Low}, respectively)

$$G_{\_TA} = \frac{T_{High}}{T_{Low}} \qquad (1)$$

When, at increasing input temperature, the thermoacoustic gain equals or exceeds the acoustic losses in the acoustic feedback circuit, oscillation will start at a frequency for which the acoustic length of the circuit corresponds with 2π phase shift. Note that due to the additonal volumes the physical loop length will be less than the acoustic wavelength (λ). If the input temperature is raised above this onset temperature the surplus of acoustic power can be extracted by the linear alternator as useful output. Expressed in terms of the temperature difference applied to the engine the net acoustic output power P_{ac_out} is:

$$P_{ac\_out} = P_{ac\_loop} \cdot \left(\frac{T_H - T_0}{T_H} \right) \qquad (2)$$

In which T_H is the engine input temperature and T_0 the engine heat rejection temperature. For an ideal travelling wave thermoacoustic engine this equals the Carnot factor. Real thermoacoustic engines could reach 40 to 50% of this values which is also called "second order" or exergetic efficiency.

For a travelling wave propagating in a wave guide (e.g. a feed back loop), the acoustic loop power (P_{ac_loop}) is given by:

$$P_{ac\_loop} = 0.5.A_0 \cdot \left(\frac{\hat{p}_a^2}{\rho.c} \right) \qquad (3)$$

In which A_0 is the feedback loop cross-sectional area, p_a the pressure amplitude, ρ the gas density and c the propagation velocity. The term $\rho.c$ is also called the acoustic characteristic impedance (Z_0) of the gas[3] which gives the "natural" ratio between pressure and velocity amplitudes. For a travelling wave this velocity amplitude (v_a) equals:

$$v_a = \frac{p_a}{\rho.c} \qquad (4)$$

As is the case for continuous gas flow, acoustic gas velocity is associated with boundary and turbulent losses. While there is not a sharp cut-off, in the end these acoustic losses set an upper limit to the acoustic amplitude in the engine. Typical values used in thermoacoustics are 5 to 10 % of the mean pressure (P_0).

In travelling wave configurations, velocity reduction in the regenerator is required to reduce viscous losses. This is done by enlarging the regenerator cross-sectional with respect to the feedback tube diameter. Doing so the volume flow rate, and with that acoustic power, is not altered but local velocity in the regenerator is reduced proportionally. The associated change in cross-sectional area will introduce some minor losses but with a proper rounding these losses are found to be less severe than the acoustic dissipation in a standing wave resonator yielding a more efficient coupling between engine and load.

Travelling wave feedback is found not only to have less acoustic losses but, even more important, this type of (loop) resonator allows inserting an arbitrary number of regenerator units. By this means, acoustic power gain could be increased proportionally as will be explained in the next section.

[3] for air at atmospheric pressure $Z_0 \approx 405$ N.s.m^{-3}

2.3 Multistage thermoacoustic engines

At low operating temperature differences and/or high heat rejection temperatures, the acoustic power gain (equation 1) is limited and net acoustic output power of the engine is only a small part of the acoustic loop power (equation 2). The other way around, a high acoustic loop power is required to get a certain output power. However, for a fixed drive ratio, high loop power requires large feedback tube diameters which results in higher acoustic losses. A way out is to increase the acoustic loop gain.

A well-known option to increase thermo acoustic power gain at low operating temperatures is to place multiple thermoacoustic units[4] in cascade. In order to benefit from using multiple regenerator units in series it is obvious that in all connected units the acoustic conditions (high and real impedance) should be maintained. Unfortunately, in the classic torus configurations[5] with standing wave resonator this is hard to maintain in more than 2 regenerator units without adding additional loops or branches, which increases the acoustic losses.

In a travelling wave feedback circuit, the preferred acoustic condition in all regenerator units can be set individually by adapting the length and diameter of the mutual tube sections and increasing the regenerator cross-sectional area relative to the feed back tube diameter. When carefully dimensioned, in theory an arbitrary number of regenerator units (with increasing cross-sectional area in the propagation direction) can be connected in series. The consequence of this approach is that in general all regenerator units and mutual tube sections are different in size. A “special case” however, in which all regenerator units and mutual tube sections are identical, is when four regenerator units are placed on a mutual distance of a quarter wavelength (¼ λ).

2.4 Special case, the 4-stage engine

In case the mutual distance between the regenerator units is ¼ λ, reflections due to impedance anomalies tend to compensate each other[6]. If, in addition, an acoustic load is added per stage the device is acoustically completely symmetric and therefore will be “self matching”, requiring no adjustment or tuning at all.
Recently, such a novel 4-stage “self matching” travelling wave engine has been developed and experimentally validated by Aster. Based on the promising results so far, this symmetric 4-stage configuration will be the base for further deployment and commercialisation of low temperature thermoacoustic engines for utilizing solar and waste heat.

2.5 Thermoacoustic water heater and power generator

For FACT Foundation, Aster has implemented the concept of the 4-stage travelling wave engine into an atmospheric pressure operated thermoacoustic water heater and power generator intended for developing countries. Beside hot water, the device is intended to supply about 50 W electric power. The first prototype and proof of concept of this extremely thin (50 mm) thermoacoustic engine is described in the next chapter. The basic configuration of the device is depicted in Figure 3.

[4] regenerator clamped between the in- and output heat exchanger
[5] a loop mucht shorter than the wave lentgth
[6] Similar to the ¼ λ transformer applied in microwave and anti reflection coatings in laser optics.

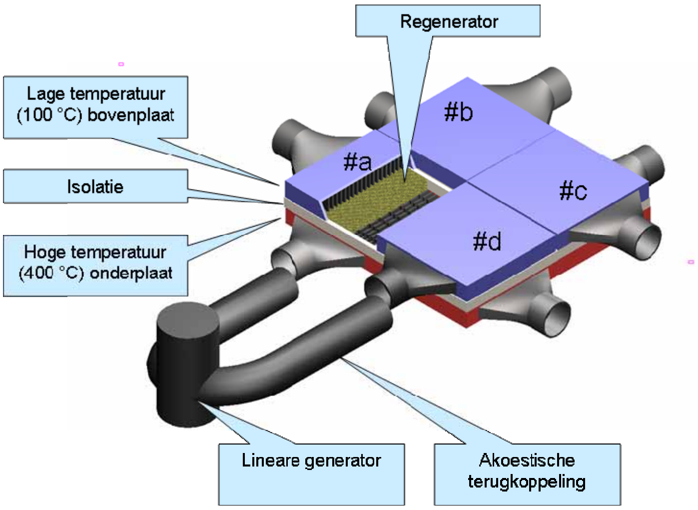


Figure 3 4-stage thermoacoustic water heater and power generator (only one feedback loop / alternator drawn)

The idea is to position the device between an arbitrary heat source (e.g. wood fire) heating the bottom hex plates (red) and a water reservoir attached to the top hex plates (blue) keeping the temperature below 100°C. Linear alternators will be connected to the feedback loops connecting the four engine stages.

3 Construction of the test set-up

This chapter describes the characterisation and construction of the engine and alternator test set-up.

3.1 4-stage engine

Thermoacoustic devices have a large freedom of implementation. The construction described here is only one of them, keeping in mind that the device has to be placed between an arbitrary heat source and a water reservoir resulting in a remarkably flat thermoacoustic engine.

3.1.1 Heat exchangers

The high and low temperature hex are identical and are made from commercially available aluminium heat sinks measuring 200 x 200 x 25 mm with fins at one side. Fin height and fin distance are respectively 20 mm and 7 mm. At the finned side, a 145 x 145 x 3.5 mm chamber is milled out for inserting the regenerator. At the back, the open side of the fins is closed by an aluminium strip. The fins at the input are provided with an appropriately shaped connection equipped with 1¼" gas thread flanges. Between the fins and regenerator, a sheet of nickel foam was inserted to get a more uniform heat distribution.

The hot (bottom) hex plate will be directed towards an arbitrary heat source and could be heated up to 360 °C. The top hex plate should be kept at a low temperature (40-90 °C) by a water reservoir or other heat removing provision.

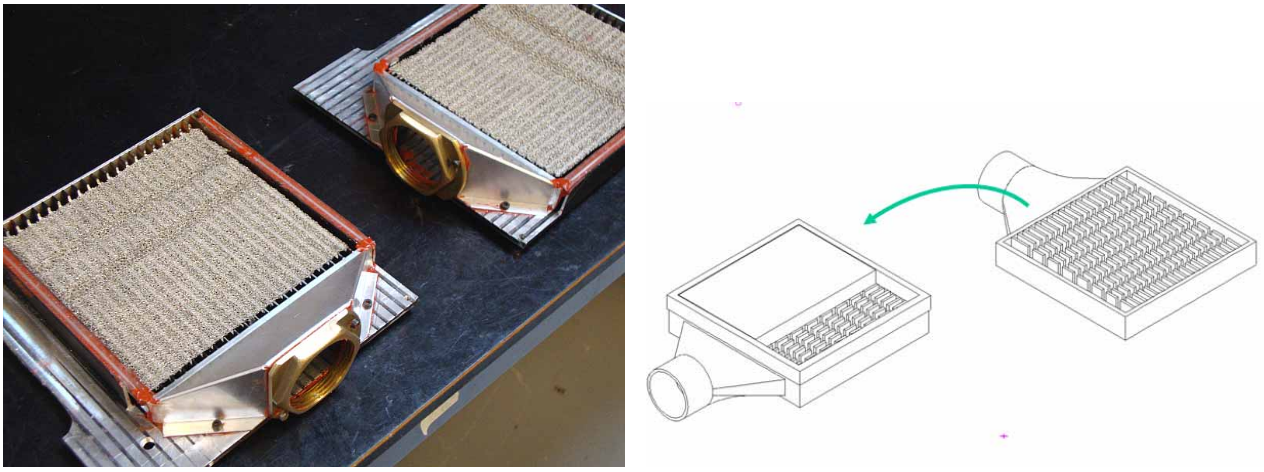


Figure 4 Hot and cold hex made from aluminium heat sink

Both hex will be assembled with two appropriately shaped silicon gaskets (1.2 mm thick) in between as a thermal isolation layer. Note that the acoustic hex input connections have a mutual 90° shift.

3.1.2 Regenerator

The regenerator is about 9 mm thick and placed inside the milled chambers of the hex. The regenerator is made up of 35 sheets (145 x 145 mm) stainless steel gauze of 80 μm wire diameter, wire distance of 120 μm and has a volume porosity of about 70%.

Regenerator temperatures in #a and #b (see figure 3) are measured with 0.5 mm type K thermocouples positioned in between the regenerator and both hex's.

Static heat conduction is found to dependent on mechanical pressure on the regenerator packet. Initially, static heat conduction (k) per regenerator was measured in the order of $0.5 W.m^{-1}.K^{-1}$. Releasing this pressure by adding an additional gasket finally reduced the k value to $0.24 W.m^{-1}K^{-1}$.

3.1.3 Acoustic feedback and assembly

The first section of the feedback tubes are made of 1¼" steel gas tubing screwed into the hex flanges. The remaining of the feedback tubes (which stay below 80°C) are made of standard 50 mm (43mm inner diameter) pvc tubing. Figure 5 shows the implementation of the extremely flat thermoacoustic engine (without alternators).

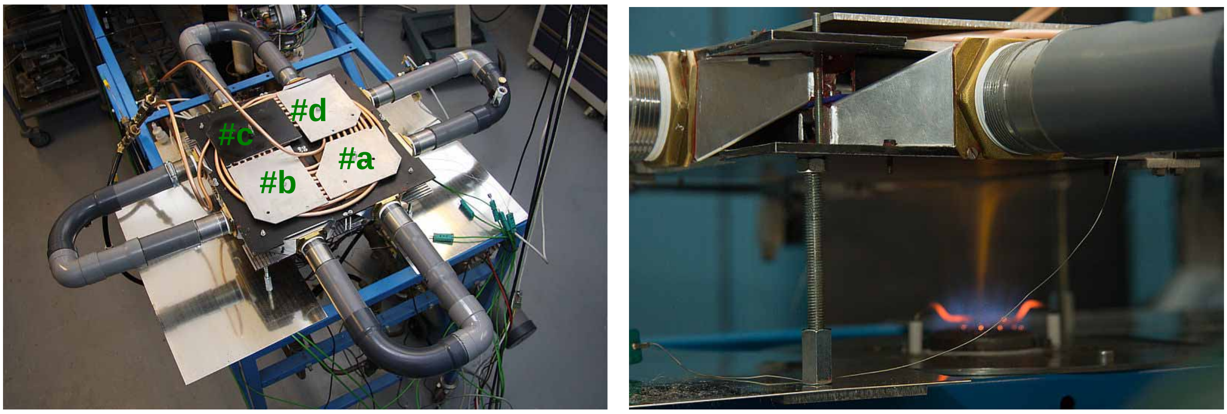


Figure 5 test set-up of the flat (50 mm) thermoacoustic engine with the water reservoir replaced by a spiralized water tube

The engine is positioned above a controllable gas burner. A spiralized water tube is connected to the upper plates (cold hex) allowing for measuring the process heat rejected at low temperature (40-90°C).

Acoustic power is measured in the feedback loop between #a and #d by the so called "dpdx method". This method uses two pressure sensors spaced 120 mm apart. From the complex pressure difference between both sensors, the gas density and frequency the acoustic velocity (v_a) can be derived and from that the acoustic loop power.

3.1.4 Characterizing TA engines

A thermoacoustic engine is characterized by its onset temperature difference and by the slope of the $\Delta P_{ac\_loop} / \Delta T$ curve. To be able to connect a useful load to the engine, the onset temperature must be far below the maximum operating temperature and the slope must be as steep as possible. The measured curve for the test engine is shown below.

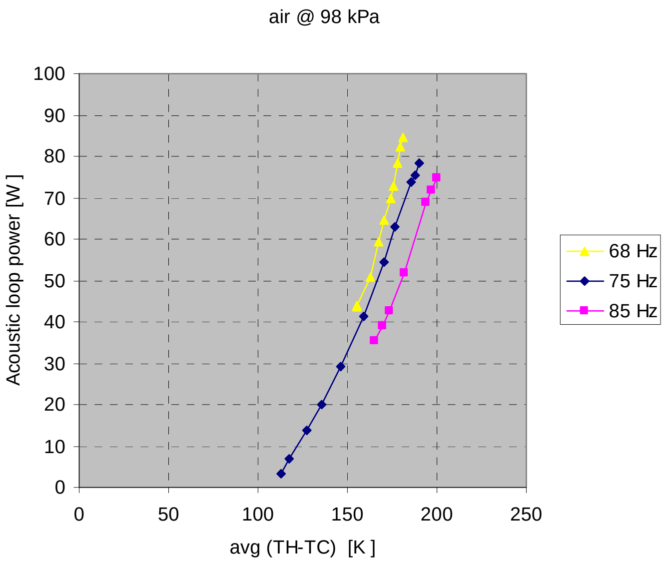


Figure 6 Loop power versus temperature difference across the regenerator for the unloaded engine.

Figure 6 shows an onset temperature difference of 105 K at 75 Hz. At 190 K the acoustic loop power is nearly 80 W at a pressure amplitude of 3700 Pa. This high acoustic power level at relatively low pressure amplitude is typical for the near travelling wave in the feedback loop.

The same test is done with longer and shorter feedback loops yielding lower and higher frequencies. Note that the slope is nearly identical, indicating similar (turbulent) losses, but that onset and operating temperatures increase with frequency. This increase is caused by acoustic anomalies due to the open space of the inlet section of the hex which increase at diminishing feedback length c.q volume.

3.2 Alternator basics

The periodic pressure c.q. velocity variation is converted in electricity by a so called linear alternator attached to the feedback loop. In this alternator a magnet is moved longitudinally in a set of coils. The magnet is connected to a low friction piston or membrane. This is schematically depicted in Figure 7.

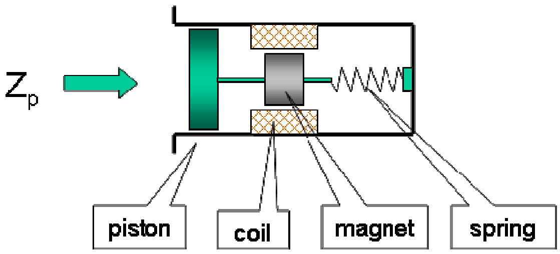


Figure 7 Linear alternator.

The resonance frequency of the moving mass of piston and magnet, together with the spring, is set equal to the oscillation frequency of the thermoacoustic engine. The acoustic impedance of the piston (Z_p) in that case is near real and acoustic power absorbed by the piston is given by:

$$P_{ac\_alt} = 0.5.A_p.p_a.\omega.u_p \qquad (5)$$

In which A_p is the piston cross-sectional area, p_a the pressure amplitude (peak value), ω the radial frequency ($2.\pi.f$) and u_p the piston stroke amplitude. At increasing power levels the moving mass of the alternator could be quite high causing external vibrations. This can be avoided by using two units with pistons moving in opposite phase, counterbalancing external vibrations. Note that for the same power extracted, cross-sectional area or stroke per unit is halved.

The alternator acts as an acoustic load to the engine and, similar to normal engines, this load has to be set to a proper value to get maximum useful output power. Exact dimensioning however depends on many parameters requiring a thorough analysis of acoustic impedances, temperatures etc.

As an educated guess for the for the 4-stage engine (high gain) is that in total about half the loop power may be extracted as acoustic output power. In case of four alternators each piston pair may extract 1/8 of the loop power. Equation 5 may be used in that case to find the physical measures of the individual alternators like stroke and piston diameter. The acoustic power extracted is also the input parameter for dimensioning the electro-magnetic part.

3.3 Balanced linear alternator

The linear alternator is built around actuators (model 6033) purchased from Magnetic Innovations B.V. (M.I.) The highest measured mechanical to electric conversion efficiency is 63% at an electric load of 2.5 Ω. The moving magnet of the actuator is mounted on two blade springs (supplied by FACT) allowing a longitudinal movement while at the same time providing the radial stiffness required to maintain the gap between magnet and coil. The (moving) mass of the magnet (130 g) together with the stiffness of both blade springs yield a resonance frequency of 47 Hz. This frequency adapted to the engine oscillation frequency of 75 Hz by adding a separate compression spring (S ≈ 30 $N.mm^{-1}$).

For the first experiment the cone of a standard speaker (75mm ∅) is used as a piston. This is done by removing the magnet and gluing a pvc plug at the position of the voice coil, which in turn is connected to the moving magnet. This is a quick and cheap solution but life time expectation is low and stiffness and sealing of the cone is not optimal as was found from reflection measurements. In case of a mechanical blocked cone the reflection coefficient should be one, but for a single cone is was measured to be 0.8.

The alternator is balanced in the sense that the magnets move in opposite phase cancelling out external vibrations[7]. Mass and spring stiffness of both units are equal. From the assembled balanced alternator the reflection coefficient was measured as a function of the load resistor (R_e). The result is given in Figure 8.

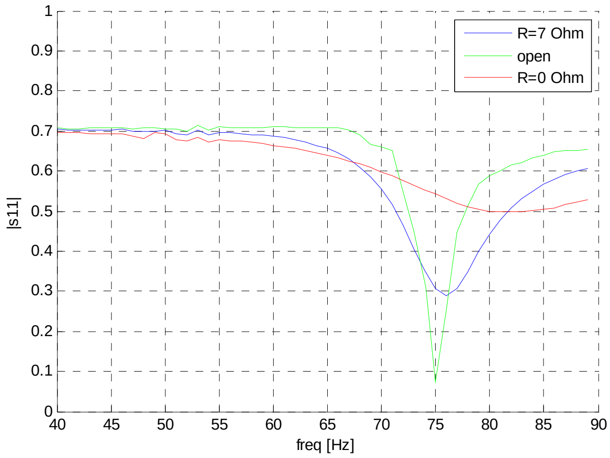


Figure 8 Acoustic reflection of the balance alternator for various load resistors (D_{ref} = 67 mm)

Figure 8 shows that, at the mechanical resonance frequency (75 Hz), acoustic reflection and acoustic alternator impedance are related. Outside the resonance range, acoustic reflection is hardly dependent from R_e and approaches a value of about 0.7. As mentioned, reflection should be close to 1 and a value of 0.7 implies that from the acoustic power applied to the alternator piston only 0.49 is converted into mechanical power.

Measurements on the actuators show that conversion efficiency depends on the load resistor. For the actuators used the optimum is around 2.5 Ω at which 63% of the mechanical is converted into electric power.

Summarizing, using the current alternator components, only 100*0.49*0.63 = 31% of the engine net acoustic output power is converted into electricity (for R_e = 2 x 2.5 Ω).

The mechanical resonance frequency of the alternator should equal the engine oscillation frequency (or vice verse) to maximize acoustic power transfer from engine to alternator. The impact of de-tuning between engine and alternator is given in Figure 9.

[7] Using a single alternator result in a (too) high noise and vibration level

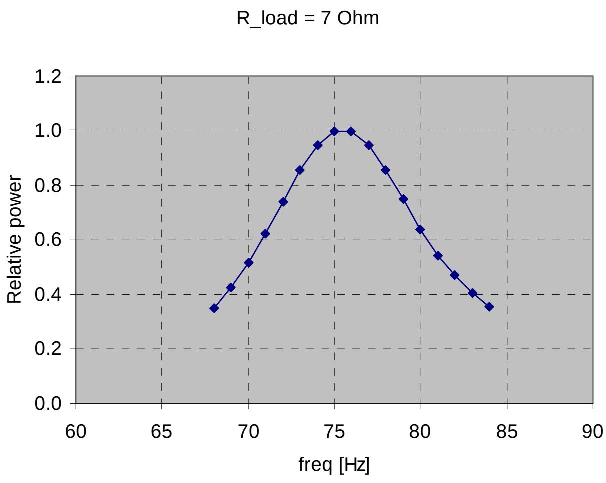


Figure 9 Power transfer between engine and alternator versus frequency

As the gas temperature in the acoustic circuit raises during operation, acoustic load to the engine will vary according to Figure 9. This effect has to be taken into account by setting the oscillation frequency of a “cold” system to a 3-5 Hz lower value. In that case the oscillation frequency will shift towards the alternator frequency during operation.

4 Experiments

This chapter describes a series of the experiments to determine the characteristics of the set-up and to optimise its performance.

4.1 Coupling the alternator to the engine

The balanced alternator is inserted into the feedback loop between #a and #b at a distance of about 17 cm from the cold hex. The load resistor is set to 5 Ω (2 actuators in series). The complete set-up is depicted in Figure 10.

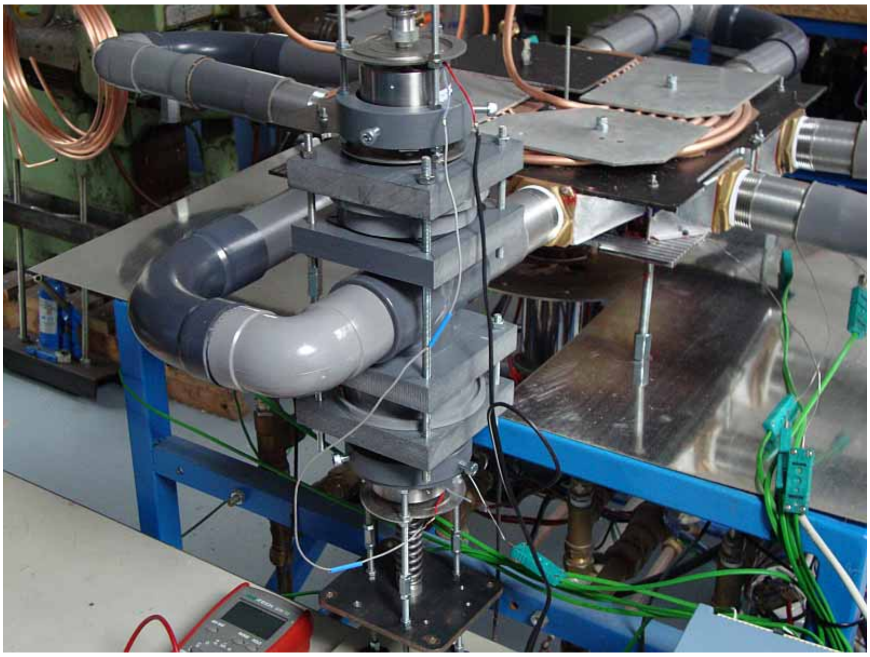


Figure 10 Balanced linear alternator coupled to the 4-stage engine

As compared with the unloaded engine, onset temperature is higher and acoustic loop power reaches no more than 24 W. The electric power dissipated in 5 Ω is 3.6 W which corresponds with an net engine acoustic output power of 3.6 / 0.31 is 11.6 W. For the measured pressure amplitude of 2450 Pa this is close to the simulation.

During the first experiment some thermal imperfections were observed:

- a high temperature difference between the each hex plate and regenerator
- high static heat losses. With the burner at minimum power (≈ 2kW), the temperature difference between top and bottom hex's does not exceed 93 K while the set-up as a whole is heated to more than 100°C
- more than 60 K temperature difference between the cold hex plate and the water circuit while this should be less than 10 K

4.2 Reduction of static heat loss

Thermal conduction between both hex is reduced by inserting an additional gasket reducing the mechanical pressure on the regenerator sheets, consequently reducing static heat conductance (k_{reg}) from 0.48 to 0.24 $W.m^{-1}.K^{-1}$.

The water circuit (copper spiral) is inserted in a thermal conducting paste reducing the average temperature of the device by 50 K which is beneficial for both gas density[8] and construction.

Due to the reduced heat losses, more heat is available for the thermo-acoustic process. This is confirmed by the fact that for the same burner power, acoustic loop power increases from 24 to 34 W and the electric output power to 5.4 W, corresponding to an acoustic loop power of 5.4 / 0.31 = 17.4 W. Pressure amplitude is now 3000 Pa and heat rejected at the cold hex is about 1100 W of which 580 W account for static heat conduction.

4.3 Increasing hex heat transfer rate

The initial measurements indicate a large temperature drop between regenerator and hex (plate) temperature. An attempt in made to increase the hex heat transfer rate by pressing strips of aluminium foam between the fins (20 pores per inch). This is shown in Figure 11.

Figure 11 Aluminium foam pressed between the hex fins

Heat transfer measurements on the hex plate show nearly a factor 3 increase in heat transfer. All eight hex in the system are modified this way. The effect on the system is measured on the unloaded engine.

[8] Gas density (ρ) is inverse proportional with temperature

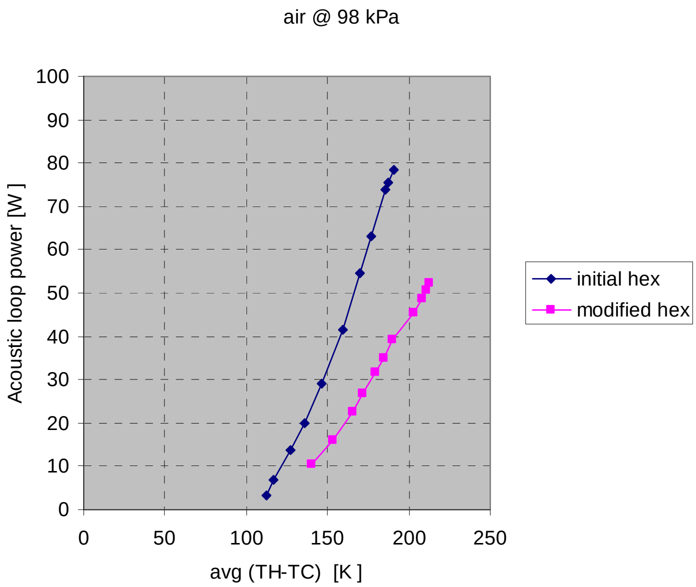


Figure 12 Impact of the eight modified hex on the engine performance

Figure 12 shows that the onset temperature is slightly higher and that the ΔP_{ac_loop} / Δ T becomes less steep compared to the initial hex, indicating more dissipation.

From the heat transfer measurements it was found that the flow resistance of the modified hex was also a factor of 3 higher, yielding more acoustic dissipation which is not compensated by a better heat transfer rate. Therefore, on system level, the initial finned hex plates performs best.

4.4 Alternator improvement

In order to reduce the observed acoustic losses of the "speaker cones" it was decided to replace them by a cylinder-piston combination with clearance seal. For cost considerations this assembly was made of pvc. Despite the coefficient of expansion and mechanical restrains during shaping it was feasible to obtain a clearance between piston and cylinder in the order of 130 μm. This is still to a too wide gap but the results are already better than the speaker cones as can be seen from Figure 13.

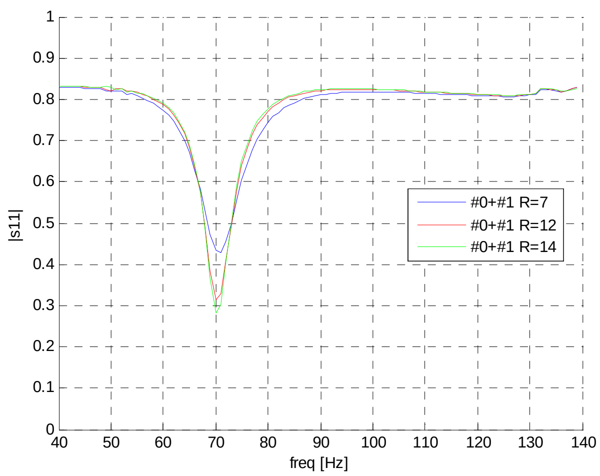


Figure 13 Measured reflection coefficient for balanced alternator equipped with clearance seal pistons

Figure 13 shows that the reflection coefficient beside the resonance frequency is higher (> 0.8) compared to the set-up with the speaker cones (0.7, see Figure 8). The output is expected to be a factor 0.8^2 / 0.7^2 = 1.3 higher. Due to the piston mass, the resonance frequency of the alternator is reduced slightly from 75 to 70 Hz. The engine set-up is adapted to this frequency by extending the feedback loops. Compared to the previous set-up, the measured electric output for the new configuration is raised from 5.4 to 6.3 W, an increase of 17%.

The diameter of the piston (82 mm) is somewhat larger than the effective diameter of the speaker cone (73 mm). As a consequence, for the same electric load of 5 Ω, the engine experiences a higher load, resulting in an increase of regenerator temperature difference from 215 K up to 249 K yielding a lower overall efficiency.

The ratio between measured electric output power and measured acoustic loop power for this configuration is 0.18 which is higher than for the previous case (0.16) corresponding with a more strict coupling between engine and load. These and previous experiments indicate that careful matching between load and engine is a prerequisite for maximum performance.

During this measurement session a degradation of performance was observed. After disassembly it was found that both surface of the pvc piston and cylinder were affected by mechanical contact because the springs used do not have enough radial stiffness to maintain the air gap, in particular at higher amplitudes. Better springs (more radial stiffness) and shaping of the piston-cylinder combination is required. However, the basic idea has proven to be feasible.

4.5 Reduction of acoustic feedback loss

Due to constructive restrains, the bends of the feedback loops comprise a sharp bend. It is expected that this kind of bends will introduce some minor losses. Therefore new symmetric bends have been manufactured. The effect on the (unloaded) engine is shown in Figure 14

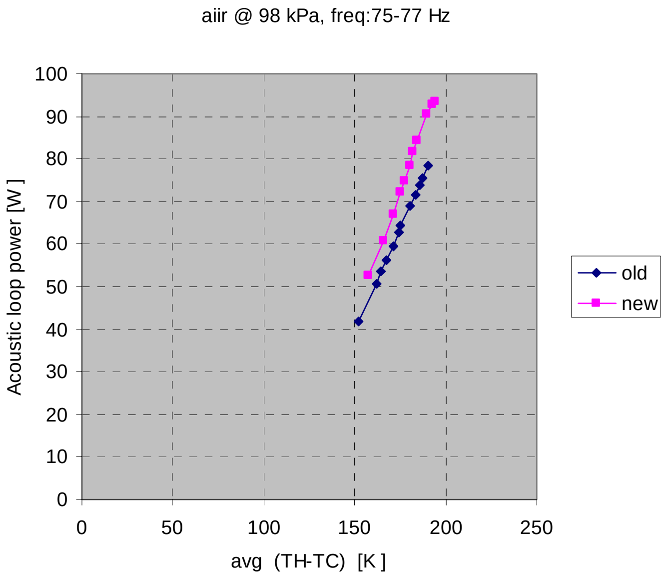


Figure 14 Impact of replacing the sharp bend (old) in the initial set-up (see Figure 5) by a more appropriate one (new)

Figure 14 shows that compared with the old bends, the slope of the ΔP_{ac_loop} / Δ T curve is somewhat more steep indicating less acoustic feedback loss. As a result, at high amplitude the acoustic loop power is raised from 80 W up to 95 W for the same temperature difference applied.

4.6 Impact of a single load on the multi stage engine

The net electric output of the set-up is found to be less than expected. As shown in previous sections one cause is the low efficiency of the simple alternator. Another issue not addressed before is the number of loads. The concept is based on symmetry in which each engine has its own load in order to maintain near identical acoustic conditions in all stages, as is shown in the left graph of Figure 15. In this test set-up however, the load is concentrated on a single position. From simulation and measurements it is known that net engine acoustic output power is close to the acoustic loop power (P_{ac_out} / $P_{ac\_loop} \approx 0.8$) resulting in significant deviation of the local acoustic impedance but also affecting the impedance in the other engine stages. For a single load attached to the set-up the acoustic impedance per stage is given at the right graph.

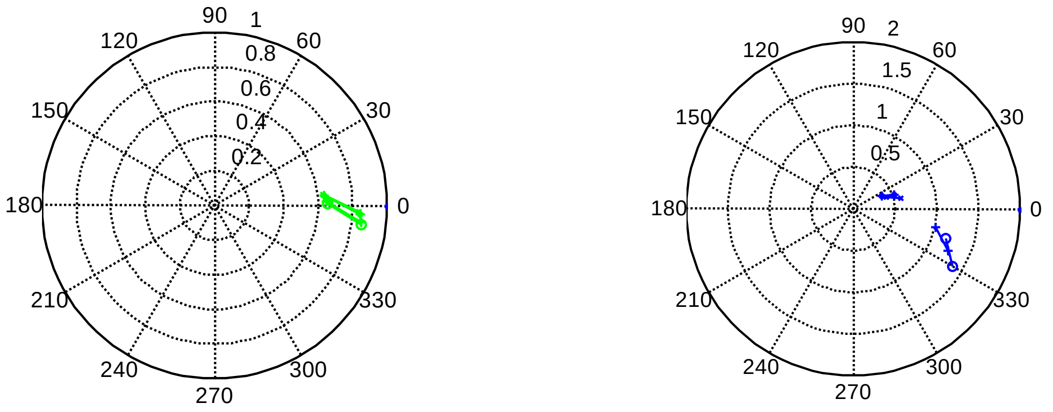
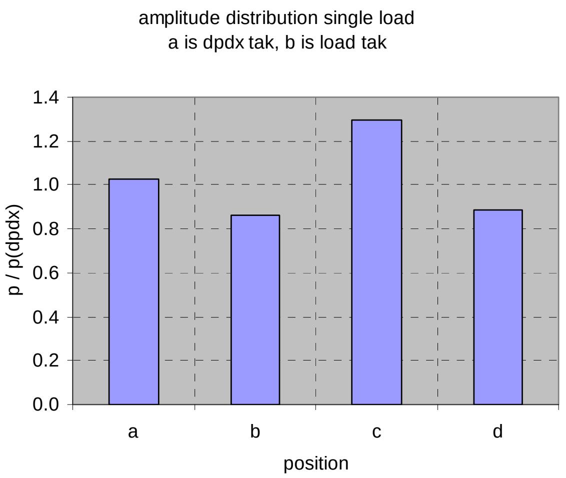


Figure 15 Regenerator impedance (Z / ρ.c) in case of a symmetric load (left) and in case of a load at one position (right)

Figure 15 clearly indicates that a load at only one position modifies the impedance in all stages, affecting the contribution per stage and thus reducing output power and efficiency. Simulation shows that exergetic efficiency[9] with a single load as compared with the same absolute load value but distributed over four stages drops from 0.41 down to 0.29.

The detrimental effect of the single load is also confirmed by the non-uniform distribution of measured pressure amplitude per engine stage. The variation in pressure amplitude in the set-up is shown in Figure 16.



[9] Acoustic output power over heat supplied times the Carnot factor

Figure 16   Measured pressure amplitude distribution per engine stage

For a symmetric load the pressure amplitude per stage should be identical. Figure 16 however shows a significant difference between stages which is in line with the variation in impedance as shown in the right graph of Figure 15.

4.7 double load measurements

From pressure distribution and simulation, it is expected that performance will improve when the 4 stage engine is loaded with multiple (but at least two) alternators spaced ¼ λ apart. Unfortunately, currently only one balanced alternator has been built and available. To be able to perform this measurement, the balanced alternator is split up and each unit is fixed to a brick in order to increase the fixed alternator mass and reduce vibrations. The measuring set-up is depicted in Figure 17

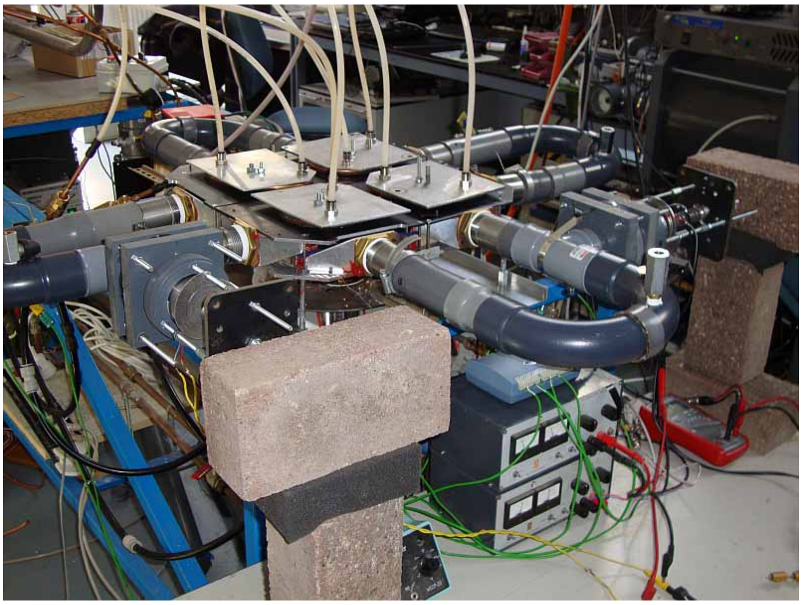


Figure 17   4-stage engine with two (unbalanced) alternators attached to #a and #b

The measured electric output power versus temperature difference is plotted in Figure 18.

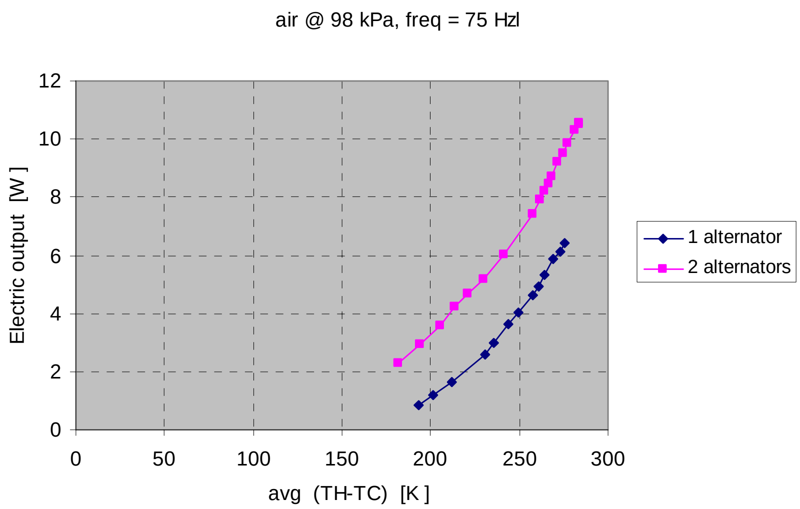


Figure 18 Electric output versus the temperature difference for one and two alternators connected to the 4-stage engine

Connecting two alternators to the 4-stage engine improves the performance, as predicted by theory and simulations (see section 3.6). At the maximum allowable temperature of the set-up, the electric output power is raised from 6 W_e for the single (balanced) alternator up to 10 W_e for two alternators spaced ¼ λ apart. Taking into account that the conversion efficiency of the alternators used is less than 0.5, the net acoustic output power of the engine is estimated to be more than 20 W which is close to the theoretical value found in simulation for a device using air at atmospheric pressure (21W). Figure 18 shows also a lower onset temperature in case of the double load, which indicates a better acoustic impedance matching.

4.8 2-stage engine

The thermoacoustic engine concept can be implemented also as a 2-stage engine, in which the two engines are positioned on a mutual spacing of ¼ λ and for which the (travelling wave) feedback loop has an acoustic length of ¾ λ. This reduces the number of components which is beneficial from manufacturing point of view. However, reducing the number of regenerator units (acoustic amplifiers) will halve the loop gain and therefore double the acoustic loop power required for the same net acoustic output power at the same operating temperatures.

Using the existing set-up a 2-stage engine has been configured by disconnecting #B and #C as shown in Figure 19.

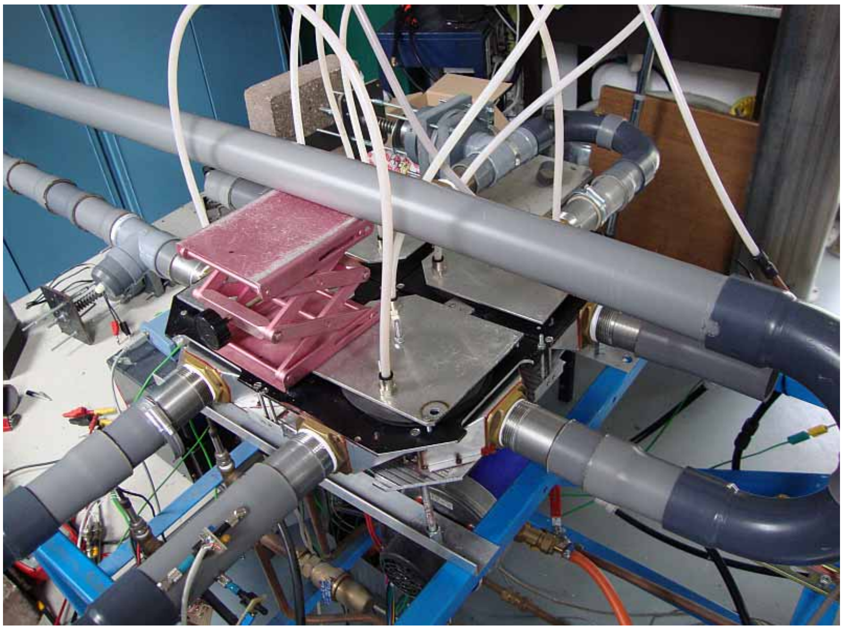


Figure 19 2-stage engine with ¾ λ feedback loop

The length of the ¾ λ feedback loop is set to such a value that the oscillation frequency is 73 Hz, as in the previous cases. The engine is characterized the same way as explained in section 2.1.4. Figure 14 shows the performance of both the 2- and 4-stage engine.

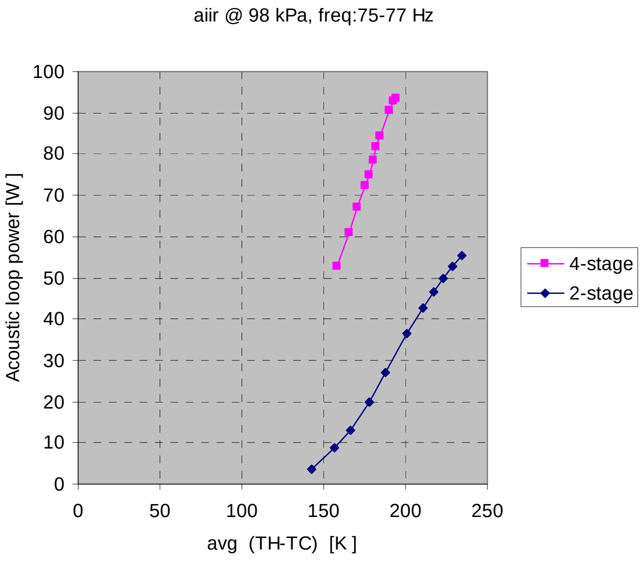


Figure 20 unloaded characteristic of the 2 and 4-stage engine

Figure 20 shows that the onset temperature has hardly changed. The slope of the ΔP_{ac_loop} / ΔT curve however is about half as steep as is expected from the lower gain. For the 2-stage engine, acoustic losses in the acoustic feedback circuitry are similar to the losses in the 4-stage configuration. Because thermoacoustic gain is halved, engine loading and acoustic losses will have proportionally more impact. In other words, to get the same net acoustic output power twice the loop power is required, which doubles the losses for the same output power and as such reduces engine performance.

Another issue, in particular for low mean pressure (near atmospheric) devices, is that static heat conduction is found to be a dominant factor. As can be seen from Figure 20, the 2-stage engine requires a significantly higher temperature for the same loop c.q. output power. Due to the increased static heat losses, the loop power of the 2-stage engine can reach no more than 60 W which of course further limits the useful output power that can be produced by the engine.

5 Conclusions and recommendations

A prototype of a thermoacoustic water heater and power generator was designed, built and tested by Aster as commissioned by FACT foundation. The aim was to develop a device that can be placed on an arbitrary heat source (e.g. wood fire) and can produce both hot water and 50 W of electric power.

The constructed prototype and the experiments have proven the full concept of the heat powered generator, but also show that the target of 50 W electric output power is difficult to reach for atmospheric devices within the limitations of size and weight, as explained below.

At a nominal pressure amplitude of 5% of the mean pressure (drive ratio), the travelling wave loop power for an internal feed back tube diameter of 43 mm is about 47 W. Due to the high gain of the 4-stage concept, in total nearly half this power can be extracted as useful output power (using 4 loads) which corresponds well with the net acoustic output power of 20 W measured in the test set-up and found from simulations.

Assuming that eventually a low cost alternator with 70% conversion efficiency can be applied, the targeted 50 W of electric output power requires a net acoustic output power of at least 2 x 50 / 0.7 = 143 W, in case of a 4-stage engine. Without changing gas type and mean pressure, 50 W net electric output requires an increase of all cross sectional areas (regenerator, hex and feedback loops) by a factor of 143 / 47$\approx$ 3.

For the current dimensions, static heat conduction through the regenerator and housing is found to be more than half the thermal power supplied. This will not improve when increasing the regenerator cross-sectional area, and in absolute sense three times more heat input is required to maintain the operating temperature.

A common option to increase the power levels is to increase the system mean pressure[10] (P_0) . For the same drive ratio (e.g. 5%), acoustic loop power is proportional with P_0. Under the prevailing conditions, a mean pressure of 300 kPa (3 bara) will triple the acoustic power levels. Because static heat losses are hardly affected by mean pressure, the ratio between acoustic loop power and heat loss improves proportionally, leaving more thermal energy available to the thermoacoustic process. In addition, acoustic losses (boundary layer) are relatively low at elevated mean pressure, lowering the onset and operating temperature and therefore further reducing static heat losses and improving overall engine performance.

The 4-stage engine requires numerous (identical) components so from manufacturing (cost) point of view it would be desirable to reduce the number of stages. For a 2-stage engine however, acoustic power gain is halved as compared with the 4-stage engine and consequently only a quarter (in stead of half) of the loop power can be extracted as useful output power at the same temperature and acoustic conditions. For a 2-stage engine this raises the required loop power up to 286 W, requiring an increase of all cross sectional areas (regenerator, hex and feedback loops) by a factor of 6 compared with the current test set-up, or by an increase of mean pressure up to 600 kPa (6 bara).

Summarizing, increasing the mean pressure is the best way to cope with the targeted output levels. Experiments at elevated mean pressures however require construction of a new test set-up, because due to mechanical constrains the current set-up can not be pressurized.

Experience with the current set-up has also led to some other requirements. The shape and length of feedback loops was found to be impractical; in a next version the feedback loops should be folded someway and integrated in the water reservoir get a more compact device. One option to reduce loop length is to increase the frequency from 75 Hz up to +100 Hz. This will also increase the output power of the alternators or reduce their size and weight.

[10] Nominal mean pressure for practical high power thermoacoustic systems is 6 to 10 bar but even mean pressures up to 40 bar are reported