14th IEA Heat Pump Conference
15-18 May 2023, Chicago, Illinois

Thermoacoustic heat pump for very high temperature applications

M.E.H. Tijani*, J.A. Lycklama à Nijeholt

Energy & Materials Transition TNO
P.O. Box 1, 1755 ZG, Petten, The Netherlands

Abstract

Industrial heat pumps are gaining increasing interest as a way to decarbonize the industrial heat system. Waste heat is upgraded to useable temperature levels, leading to lower energy demand, lower energy costs and lower CO_2 emissions. Industry needs heat pumps which can operate at high temperatures (>200°C) and can achieve large temperature lifts. Conventional heat pumps do not have the capability to accomplish these requirements. The thermoacoustic heat pump (TAHP) is an auspicious innovative technology which uses acoustic power to lift heat from a low-temperature source to a high-temperature sink. High temperatures and large temperature lifts can be achieved by making use of Stirling cycle with a gas (helium) as working fluid. This paper presents the design, development, and test of a TAHP driven by a piston compressor which produced steam at 170°C with a high temperature lift. The heat pump has the potential to produce steam at a high temperature of at least 250°C.

Keywords: Heat pump, thermoacoustics, Stirling;

1. Introduction

Industrial heat pumps are rising in popularity as a measure to decarbonize the industrial heat system. Heat pumps are highly efficient systems to upgrade low temperature industrial waste heat to useable temperature levels, resulting in lower energy demand, lower energy costs, and lower CO_2-emissions, especially if renewable electricity is used [1]. Heat pump development is aimed at to achieving higher output temperatures to match the required high process temperatures in industry. Industry needs heat pumps which can operate up to temperatures of 200℃ and can achieve large temperature lifts. Conventional heat pumps currently do not have the capability to accomplish these requirements [2]. The temperature that can be delivered by closed-cycle compression heat pumps under development for example is limited to 150°C due to the lack of appropriate refrigerants and compressors for high temperature applications [2].

Different types of heat pump technologies are conceivable for an industrial application [3]. Conventional compression heat pumps are limited in their operating range by the working medium they employ. In contrast, heat pump technology that is based on a gas cycle can be operated at a variety of temperatures, not limited by either condensation or evaporation temperatures/pressures. These include the reverse Brayton cycle heat pump, Stirling heat pump, and TAHP. The TAHP use a Stirling thermodynamic cycle which has the highest COP of all heat pump cycles. The difference between a conventional Stirling and thermoacoustic systems is that thermoacoustic systems have fewer moving parts. The electrically driven TAHP is an auspicious new technology which can operate over a large range of temperatures, up to high temperatures (> 200℃) and can achieve large temperature lifts (e.g. 100 K) using preferably pressurized Helium as working medium. This enables the development of flexible systems that can be applied for large variety of temperature conditions, based on the same design and same working fluid

* Corresponding author. Tel.: +31-665-000-9629
E-mail address: hassan.tijani@tno.nl.

The study presented in this paper is the next step in the development of TAHPs for industrial applications. The first step was a laboratory heat pump setup designed and tested successfully for high temperature applications [4-5]. Secondly, a bench scale TAHP using an acoustic resonator has been developed and tested [6]. The current step in the development is to obtain a more compact and cost-effective heat pump by a TAHP re-design without an acoustic resonator. Another objective is to check the technical feasibility of using a commercial piston compressor as a driver for such a system. The possibility to use a commercially available piston compressors is of crucial importance to pave the path for the upscaling of TAHP to large capacities. Laboratory scale TAHPs use pistons driven by linear motors. However, linear motors are difficult to scale up and are very expensive. Piston compressors are already used in the gas industry and are commercially available at different scales up to MW’s which makes upscaling of TAHPs feasible. The last objective is to study the capability of the TAHP to produce steam at temperatures of 200 °C.

The remaining of this paper is organized as follows: section 2 discusses the working principle of the TAHP. Section 3 presents the design and construction of the TAHP. Section 4 discusses the instrumentation. Section 5 presents the experimental results. In section 6 conclusions are drawn.

2. Thermoacoustic heat pump

An illustration of a compact electrically driven TAHP is shown in Figure *1*. TAHP uses acoustic power (W) to pump heat (Q_L) from a low temperature heat exchanger (LHX) and to deliver heat Q_H to a high temperature heat exchanger (HHX). The heat exchangers are linked to an external heat source and heat sink. The heat pump consists of a regenerator (REG) placed between the two heat exchangers which are placed inside a pressure vessel filled with helium gas at high pressure. The regenerator is a porous structure with very small channels where the heat pumping process takes place. An oscillating piston driven by an electrical motor supplies the acoustic power required for the heat pumping process. The acoustic wave generated by the piston forces the helium gas in the regenerator to execute a thermodynamic cycle similar to the Stirling cycle. The acoustic wave performs the compression, displacement, expansion, and the timing necessary for the Stirling cycle. The right phasing between the acoustic pressure and the gas velocity in the regenerator is realized by an acoustic circuit formed by the flow resistance of the regenerator, the acoustic inductance L formed by the annular space around the regenerator and heat exchangers, and an acoustic compliance C (volume of gas). It is worth noting that the functionality of the displacer piston in a conventional Stirling system is replaced by the combination of the inertance (fluidic inertia) and the compliance. This avoids the use of a second piston as mechanical piston displacer [7].

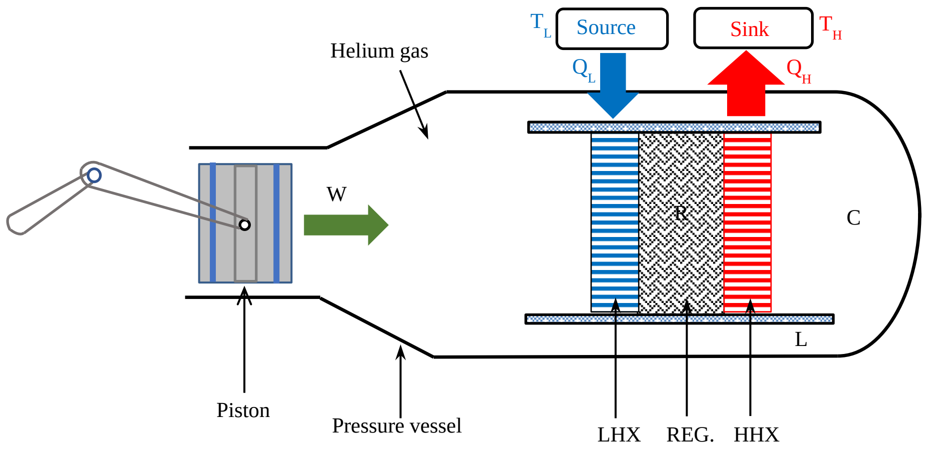

Figure 1 Schematic illustration of the electrically driven thermoacoustic heat pump

The operation of a TAHP will be explained by using an electric analog of the acoustic network, as shown in **Error! Reference source not found.**. The acoustic pressure and velocity are analog to the electrical voltage and current, respectively. The fluidic resistance of the regenerator, the acoustic compliance, and the acoustic inertance are represented by an electric resistance R, electric capacitance C, and electric inductance L, respectively. The function of the network is to create the traveling-wave timing necessary for the Stirling cycle in the regenerator. The ideal phase difference between the acoustic pressure (P) and the acoustic flow (U) in the regenerator is zero (traveling acoustic wave).

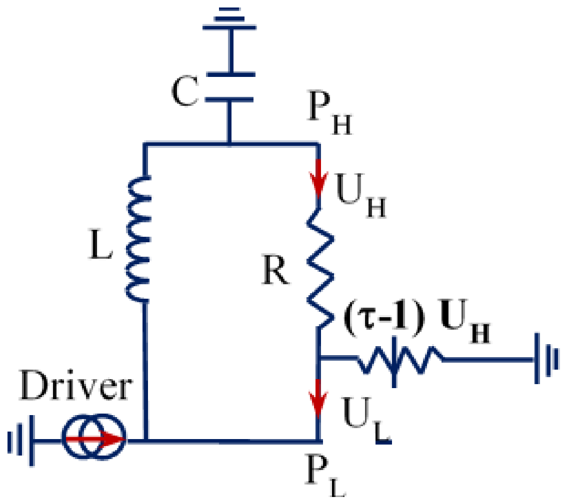

Figure 2 Simplified lumped-element electrical analog of the thermoacoustic heat pump

An analysis of the simplified lumped-element electrical analog of the heat pump can be used to show how the acoustic flows, powers, and timing are determined by the different parameters and components in the circuit. The acoustic flow entering the regenerator at the hot side U_H is damped by the temperature ratio across the regenerator $\tau=T_L/T_H$. An acoustic flow sink [(1-τ) U_H] is used to fulfil the condition of damping. U_H is damped because acoustic power is used to pump heat. The parallel branches of the inertance and regenerator have the same voltage [8]

$$i\omega L U_H = R U_H \tag{1}$$

The flows U_H and U_L are determined by the flow to the compliance

$$U_L + U_H = -i\omega C p_L \tag{2}$$

Combining expressions (1) and (2), U_H is given by

$$U_H = \frac{\omega^2 LC}{R} \frac{P_H}{1+\frac{i\omega L}{R}} \tag{3}$$

Expression (1) shows that the flow in the inertance is 90° out of phase from that in the regenerator. Expression (2) shows that the flows into the regenerator and inertance is controlled by the compliance C. Increasing the compliance volume will results in an increase of the flow through the regenerator. Expression (3) shows that if the impedance of the inertance ωL is much smaller than R, then U_C and p_c will be in phase corresponding to the traveling-wave phasing necessary for the Stirling cycle. Expression (3) shows also that the magnitude of acoustic flow is controlled by R, L, and C. In general R, L, and C are designed to create an impedance in the regenerator which is 15-30 times the traveling wave impedance $\rho_m c/A$ to avoid large losses in the regenerator.

The acoustic power at the hot side of the regenerator in the case Lω << R (pressure in phase with flow) is given by

$$W_H = \frac{1}{2} p_H U_H \tag{4}$$

Where p_H is the magnitude of the acoustic pressure at the hot side. The acoustic power flowing out the cold end of the regenerator is given by

$$W_C = \frac{T_C}{T_H} W_H \tag{5}$$

An qualitative explanation of the working priciple of the thermoacoustic effect is shown in Figure 3. As mentioned in the foregoing the thermoacoustic thermodynamic cycle is similar to Stirling cycle. **Error! Reference source not found.**a illustrates an acosutic wave treveling through the core of the heat pump. The wave enters at HHX and it is used to pump heat from a heat source to heat sink. As acoustic power is used to pump heat the wave is damped. **Error! Reference source not found.**b illustrates the thermoacoustic cycle which consists of four steps. Step1: Isothermal expansion (1→2), the gas expands isothermally in LHX and heat is absorbed by the gas. Step 2: Heating, the gas moves through the regenerator from LHX to HHX and it is heated at constant volume. Step 3: Isothermal compression (3→4), the gas is compressed isothermally in HHX and heat is rejected to HHX. Step 4: Cooling, the gas moves back through the regenerator from HHX to LHX and it is cooled at constant volume. **Error! Reference source not found.**c shows the pressure-volume diagram of the cycle which is elleptic due to the overlap between the different steps. The area of the ellipse correpnds to the acoustic power (W) used in the cycle.

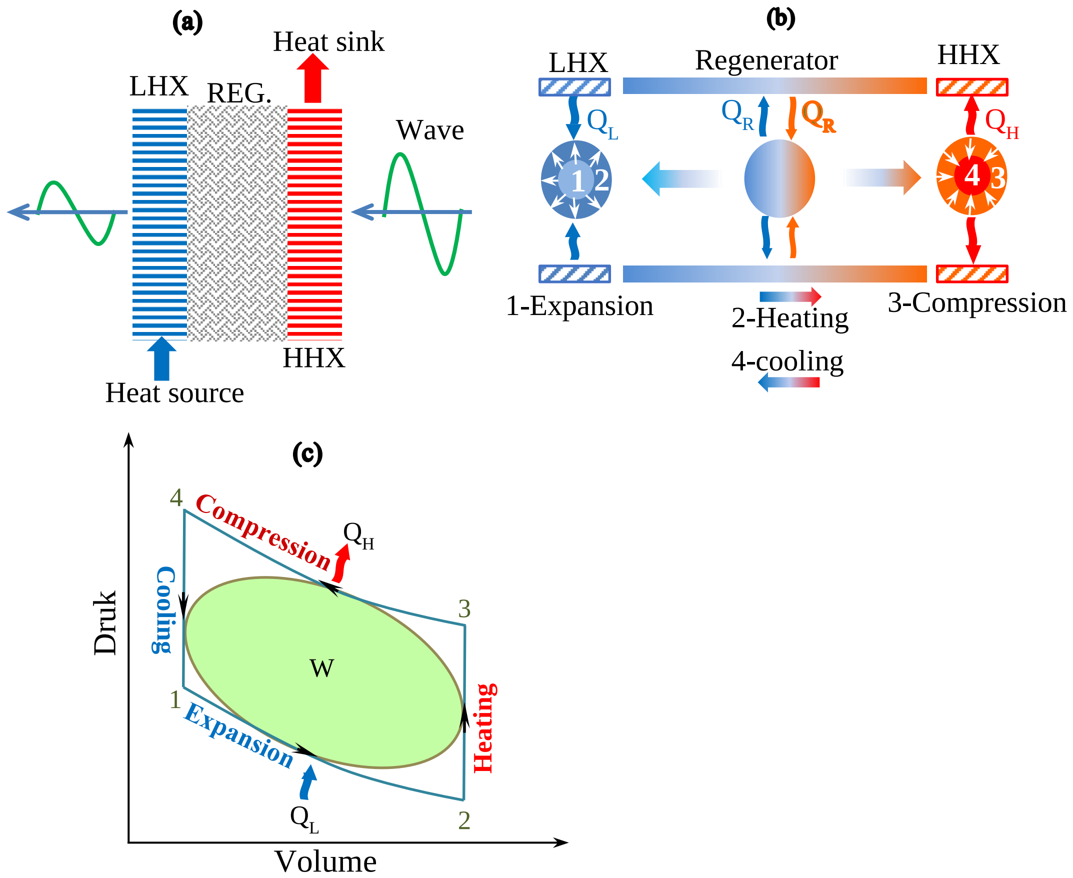

Figure 3 Illustration of the thermoacoustic cycle. (a) Acoustic wave travelling through the core. (b) illustration of the four steps of the cycle. (c) Pressure-volume diagram.

The performance of the heat pump is characterized by the coefficient of performance (COP) which is the ratio of the heat delivered to the heat sink and the acoustic work used by the heat pump.

$$COP = \frac{Q_H}{W} \tag{6}$$

The maximal theoretical limit for the COP of a heat pump is the Carnot coefficient of performance which is determined by the temperatures of the heat sink and the heat source and is given by

$$COPC = \frac{T_H}{T_H - T_L} \tag{7}$$

In practice the COP is always lower than the COPC due to the different losses in the heat pump. It is worth noting that helium is used because of its high power density, thermodynamic performance and safe use. However, it becomes rare and hence expensive. Hydrogen is a better working medium than helium and it will be very abundant in near feature and thus much less expensive. For industrial applications hydrogen could be the working medium for thermoacoustic systems.

3. Design and construction

As discussed in the foregoing, the aim of the study is to evaluate the feasibility of this compact system, of using a commercial piston compressor as a driver and the capability of the heat pump to produce steam at high temperature (~ 200℃). The starting point for the design is the piston compressor whose operation frequency and stroke will determine the operation conditions for the heat pump. A compressor is selected with a maximal speed of 1200 RPM, a stroke of 230 mm, and the piston diameter is 320 mm. Helium at a mean pressure of 50 bar is used as working fluid. This leads to the operation conditions for TAHP as given in Table 1. The drive ratio is the ratio of the acoustic pressure amplitude at the piston and the mean pressure of the gas.

Table 1 Operation conditions for TAHP

Working gas	Helium
Average pressure (bar)	50
Frequency (Hz)	20
Sink temperatures (℃)	100-200
Thermal power at 180 ℃ (kW)	6.8
Drive ratio (%)	10.4

The compressor is adapted by the supplier to comply with the expected load on the pin of the crosshead considering the thermoacoustic operation conditions [9]. An illustration of the TAHP as coupled to the compressor is shown in Figure 4. The piston compressor consists of a piston driven by an electric motor via a flywheel and a crankshaft. To limit the load on the pin of the cross-head when starting the compressor, a bypass with a starting valve is used in combination with a back volume. The back volume is a volume of gas at the crank-side of the piston which is used to balance the forces on the piston and hence to limit the load on the pin. When starting the compressor, the forces on the cross-head pin are too high and the valve must be open to limit the load on the pin. The bypass connects the back volume of the piston to the compliance to avoid large loading forces on the pin which might damage the driving system. Once the compressor is started, the valve is automatically closed and the measurements can proceed.

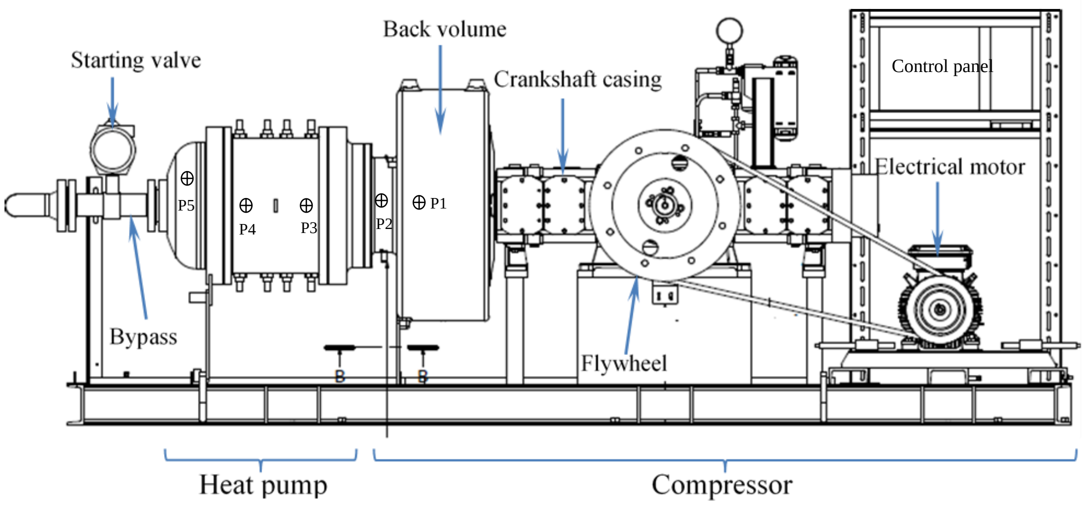

Figure 4 Illustration of the TAHP. The “P’s” are pressure sensors

A cross-sectional view of the TAHP is shown in Figure 5. The driver system includes a balance piston connected to the crank which is used to minimize the vibrations due to the oscillating motion of the driving piston.

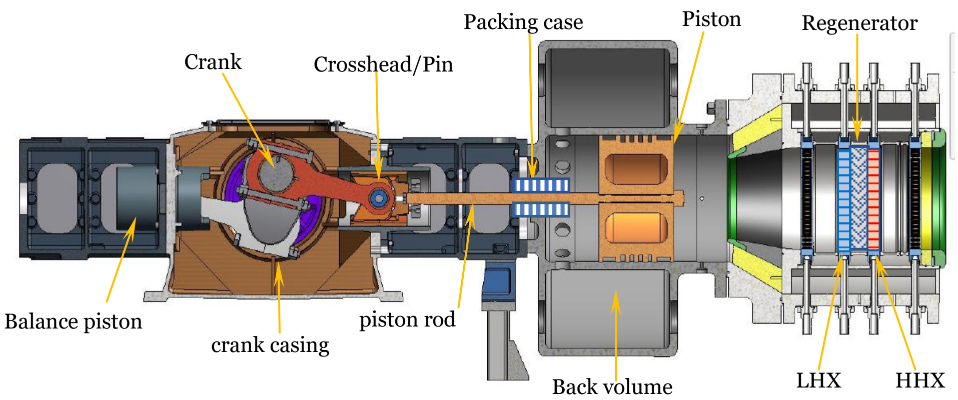

Figure 5 Cross-sectional view of the TAHP

The main parts of the TAHP are the regenerator, heat exchangers, and the acoustic circuit. These parts are discussed shortly in the following. For the sealing, O-rings are used for the nozzles of the heat exchangers and for the connections of the different components of the heat pump. A pressure packing case is placed at the crank side of the piston to limit the helium leakage from the back volume to the housing of the crosshead, which is at atmospheric pressure. The packing is provided with cooling water to remove the heat generated by friction.

Regenerator

The regenerator consists of a stack of stainless steel screen with a diameter of 260 mm and length of 48 mm. The hydraulic radius of the regenerator is 43 μm and the volume porosity is 83 %. A picture of the regenerator is shown in Figure 6. This regenerator is designed for an operation frequency of 80 Hz and an operating pressure of 50 bar and is not optimal for the operational conditions summarized in Table 1. It is used as it is available in our laboratory and can be used directly for the test of the compressor as driver and for the production of steam at high temperature. The use of an optimal regenerator will result in higher thermal power and performance.

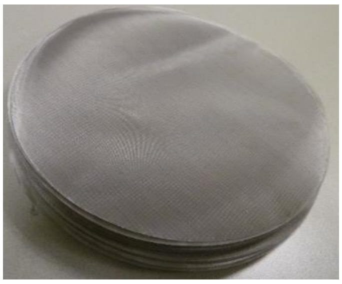

Figure 6 Stack of screen sheets used as regenerator

Heat exchangers

The heat exchangers LHX and HHX are finned, circular tube cross-flow heat exchangers as shown in Figure 7. These heat exchangers are made of copper fins and tubes and stainless steel headers. It consists of a single row of tubes with a diameter of 13.5 mm and the spacing between adjacent tubes (pitch) is 30 mm. The spacing between the fins is 0.88 mm and the fin thickness is 0.25 mm. Helium gas oscillates between the fins and a second fluid (water/steam/thermal oil) flows through the tubes. The maximum design temperature for the heat exchangers is 200℃.

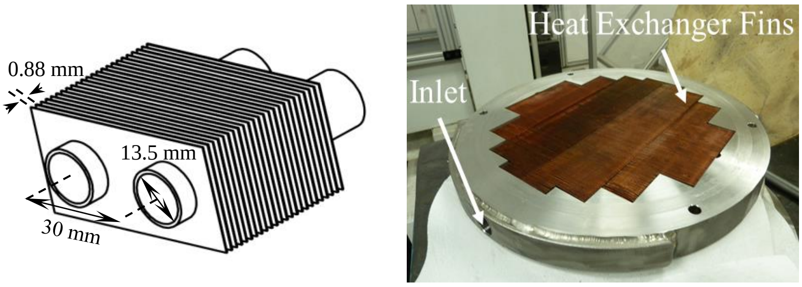

Figure 7 Illustration of the fin-tube structure (left) and picture of the heat exchangers (right)

Acoustic circuit

The feedback inertance, compliance, and regenerator flow resistance are designed to achieve the traveling-wave phasing in the regenerator and minimize viscous losses. The acoustic pressure and acoustic velocity have to be in phase at the regenerator midpoint (Stirling cycle). The feedback inertance L is formed by the annular space between the wall of the pressure vessel and the cylindrical holder of the regenerator and heat exchangers. The length of the annular space is 600 mm and the cross section is about 0.04 m^2. The volume of the compliance is 14 litres.

4. Test bench and instrumentation

A thermal bench using thermal oil has different heating and cooling circuits which can be used to simulate heat source and heat sink for the heat pump. The LHX of the TAHP is coupled to a circuit of the thermal bench that supplies heat between 50-150℃ by thermal oil. The HHX is coupled to a steam vessel where the steam produced by the TAHP can be condensed. The steam vessel contains a condenser which is cooled by the thermal bench. Pictures of the TAHP coupled to the thermal bench are shown in Figure *8*.

Various sensors are placed through the system to measure the operating parameters of the heat pump as indicated in Figure *4* [6]. The oil and water flow into the heat exchangers are measured with flow meters. The temperatures of the oil and water/steam at the inlet and the outlet of the heat exchangers are measured with thermocouples. Several pressure sensors are placed throughout the system to measure the acoustic pressure at different locations in the system. The signals from the thermocouples are read by a PLC-data acquisition system and sent to a computer. The pressure signals (magnitude and phase) are first measured by lock-in amplifiers and then send to a computer. The signals are recorded and displayed using Control Maestro. The acoustic power generated by the piston could not be measured.

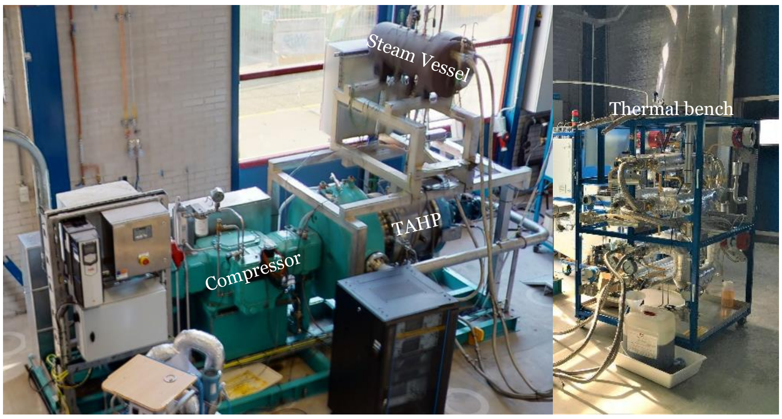

Figure 8 Picture of the TAHP coupled to the steam vessel and to the thermal bench

5. Experimental results

The heat pump is designed to operate at 50 bar helium. However, the electrical motor which drives the compressor reaches its maximum torque at about 32 bars. Therefore, measurements are done with the TAHP at reduced pressure of 30 bars. Steam is produced at different temperatures and the measurement data for the steam production at 150 and 170°C are summarized in Table 2.

Table 2 Measurements results for the production of steam at 150°C and 170℃

Dr (%)	T_{LHX}(°C)	T_{HHX}(°C)	T_{reg-H}(°C)	T_{reg-L}(°C)	Q_H(kW)
10.4	94	152	186	45	6.5
10.2	94	170	194	60	4.7

For the steam production at 170°C, the average temperature of the inlet and outlet temperatures of the oil at the LHX is 94°C and the temperature of the steam in the steam vessel is 170°C. The internal temperatures of the regenerator at the hot side and low temperature side are 194°C and 60°C, respectively. The external lift is 74 K while the internal lift is 134 K. The heat exchangers are performing poorly as the temperature difference across LHX between helium gas and oil is 33°C and across HHX between helium and steam is 26°C. The heat pump generates 4.7 kW of thermal power in the form of steam. This is lower than the design value due to the lower used mean pressure of 30 bar. The thermoacoustic computer code DeltaEC [10] predicts that operating at 50 bar, using an optimal regenerator and better heat exchangers with a temperature difference of 10 K, the heat pump would produce 16 kW of thermal power at 170 °C with a COP of about 4 and for an external source temperature of 100 °C.

As mentioned in the foregoing at the moment the acoustic power used by the TAHP can't be measured. An internal energy balance was used to calculate the acoustic power used in the heat pump. This results in an indicative COP of about 3.

6. Conclusions

An electrically driven thermoacoustic heat pump is built and tested and it can be concluded that a piston compressor can be used as acoustic driver for a TAHP. This leads to a compact heat pump without acoustic resonator. Piston compressors are commercially available at different scales up to MW's. This makes upscaling of thermoacoustic heat pumps feasible. The heat pump produces steam at different temperatures up to 170°C, supplying waste heat of 94°C. Higher temperatures can be achieved but not feasible with the current heat exchangers. The heat pump generates 4.7 kW of thermal power with a COP of about 3, excluding the losses in the compressor and internal heat losses. Large temperature differences are measured across the heat exchangers which indicates a poor heat transfer coefficient.

Simulation calculations using the thermoacoustic computer code DeltaEC [10] predicts that operating at 50 bar and using an optimal regenerator and better heat exchangers with a temperature difference of 10 K, the heat pump would produce 16 kW of thermal power at 170°C with a COP of about 4 for an external source temperature of 100°C.

In the near feature an optimal regenerator and optimized heat exchangers will be implemented to improve the performance of the heat pump and to generate steam at higher temperatures (>200°C). An acoustic power measurement system will be also incorporated enabling the measurement of acoustic power input and performance.

Acknowledgements

This work was funded by TNO and the Dutch Ministry of Economic Affairs and Climate Policy (EZK) under contract TEEI118003.

References

[1] Wolf S, Blesl M. Model-based quantification of the contribution of industrial heat pumps to the European climate change mitigation strategy. ECEEE Ind. Summer Study Proc. 2016:477–87.
[2] Arpagaus, C., Bless, F., Uhlmann, M., Schiffmann, J., & Bertsch, S. S. (2018). High temperature heat pumps: Market overview, state of the art, research status, refrigerants, and application potentials. Energy, 152, 985–1010.
[3] Nellissen P, Wolf S. Heat pumps in non-domestic applications in Europe: potential for an energy revolution. In: 8th EHPA European Heat Pump Forum, Brussels, Belgium; May 28 2015. p. 1-17.
[4] Tijani, M.E.H. Vanapalli, S. Spoelstra, S., Lycklama à Nijeholt, J.A. "Electrically driven thermoacosutic heat pump" 10th IEA Heat Pump Conference 2011, Tokyo, Japan.

[5] Tijani, M.E.H. Spoelstra, S., “High temperature thermoacoustic heat pump” 19th International Congress on Sound and Vibration (ICSV19), Vilinius, Litouwen, , 8-12 juli 2012.
[6] Tijani, M.E.H. Lycklama à Nijeholt, J.A, S. Spoelstra, S.,” Development of a thermoacoustic heat pump for distillation column”, 12th IEA Heat Pump Conference 2017, Rotterdam, Netherlands.
[7] Vanapalli, S, Tijani, M.E.H. Vanapalli, S. Spoelstra, “Thermoacoustic heat pump for domestic applications” 3rd Joint US-European Fluids Engineering summer meeting, ASME 2010, Montreal, Canada.
[8] Backhaus, S. and Swift, G. W. (2000): A thermoacoustic-Stirling heat engine: Detailed study. J.Acoust.Soc.Am, 107(6), pp. 3148-3166, (2000).
[9] https://www.howden.com/en-gb/contact-us/locations/europe/howden-thomassen-compressors-bv
[10] Ward, W. and Swift, G.W. “Design Environment for Low Amplitude Thermoacoustic Engine”, J. Acoust. Soc. Am, 95, 3671 (1994).