

Thermoacoustic compression based on alternating to direct gas flow conversion

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We present a remarkable thermoacoustically driven compression effect based on the conversion of gas flow from an alternating state to a direct state. The alternating gas flow is generated by the thermoacoustic effect in thermoacoustic engines, while the direct gas flow is achieved by the flow rectification effect of check valves. A demonstrative thermoacoustic compressor consisting of two standing-wave thermoacoustic engines, two reservoirs, and three check valves is constructed for experimental investigation. With nitrogen as a working gas and an initial pressure of 2.4 MPa in all components, a usable pressure difference of 0.4 MPa is achieved, with the average gas pumping rate reaching 2.85 Nm³/h during the first 3 seconds of the compression process. The simple mechanical structure and thermally-driven nature of the compressor show potential in gas compression, power generation, and refrigeration applications.

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I. INTRODUCTION

Conventional mechanical gas compressors, generally classified into positive displacement and dynamic types, are formed by complicated mechanical components and are generally driven by electric motors, suffering from high manufacturing and maintenance costs, high electrical energy consumption, high noise pollution, and strong electromagnetic interference. It would be a benefit to society if gas compressors of simple mechanical structure operating with low-grade thermal energy could be provided.

Thermoacoustic engines efficiently convert thermal energy into acoustic energy through a thermoacoustic effect with the significant advantages of simple structure, high reliability, and environmental friendliness.¹⁻⁴ The alternating gas flow in a thermoacoustic engine experiences periodic pressure oscillation simultaneously, demonstrating potential for continuous gas compression.

In addition, flow direction control components and methods have been applied in fluid transportation for quite some time. Recently, Gao et al.⁵ adopted a pair of check valves in a non-resonant self-circulating heat exchanger to transport alternating gas flow out of a thermoacoustic engine to exchange heat with a remote high-temperature or low-temperature heat source. Leslie et al.⁶ proposed and demonstrated an on-chip flow control method in microfluidic circuits by introducing fluidic diodes into the fluidic network. Selective fluidic transportation was realized in response to the frequency of a time-modulated pressure source.

We propose a type of thermoacoustic compressor on the basis of the thermoacoustic conversion effect of thermoacoustic engines and the flow rectification effect of flow direction control components. Fig. 1(a) shows the schematic of a proposed thermoacoustic compressor consisting of two standing-wave thermoacoustic engines and a check valve. As shown, the oscillating pressure of the low mean pressure engine can be higher than that of the high mean pressure engine at some moments. The dynamic pressure difference between the low and high mean pressure engines is the direct driving force of the gas compression effect. A thermoacoustic compressor system can be composed of one to a few thermoacoustic engines according to the demands in pressure ratio and pumping rate. Theoretically, if the pressure amplitudes of thermoacoustic engines are 10% of the mean pressures, a pressure ratio of 1.22 can be achieved by a single-stage thermoacoustic compressor composed of one thermoacoustic engine. A thermoacoustic compressor is advantageous especially in two aspects. Firstly, it intrinsically has a large operating range because the working conditions of thermoacoustic engines can vary greatly; for example, the mean pressure covers a range of 0.1 MPa-5.5 MPa^{3,7}, which can still be extended further if needed. Therefore, it is feasible to compress gas from ambient pressure to several MPa continuously by a thermoacoustic compressor with low manufacture cost, low maintenance cost, and low noise. Secondly, because of the thermally driven nature of the thermoacoustic engines, such thermoacoustic compressor can be driven directly by solar energy, thermal energy from combustion, and even industrial waste heat.

II. THEORETICAL ANALYSIS

As shown in Fig. 1(a), a thermoacoustic stack where the thermoacoustic effect occurs is the key component of thermoacoustic engines. With a sufficiently large temperature gradient along the stack that is exerted by the heater and cold end heat exchanger (CHX), the oscillating gas in the stack, experiencing thermal expansion when pressure is high and thermal contraction when pressure is low, converts heat into acoustic energy.⁸ The properly designed resonator keeps the working gas oscillating resonantly inside. On the basis of acoustic to electric analogy theory,¹ the dynamic pressure p_1 and volume flow rate U_1 are equivalent to the a.c. voltage and electrical current, respectively. The viscous resistance, inertia and compressibility of a working gas in acoustics can be modeled as a resistor, an

inductor and a capacitor in the equivalent electric circuit, respectively. The thermoacoustic conversion effect of the stack is represented by a current source gU_1 . These basic components (RLC) and the current source (gU_1) form the resonant acoustic system that together generate an acoustic oscillation. By merging the impedances, the networks of the thermoacoustic engines are simplified into the combinations of the current sources $g_L U_{1,L}$, $g_H U_{1,H}$ and the impedances Z_L , Z_H , as shown in Fig. 1(b). Two d.c. voltage offsets $p_{m,L}$ and $p_{m,H}$ denote the mean pressures of the engines, respectively. $Z_{con,H}$ and $Z_{con,L}$ are the complex impedances of the connecting tubes on the high mean pressure engine side and the low mean pressure engine side, respectively.

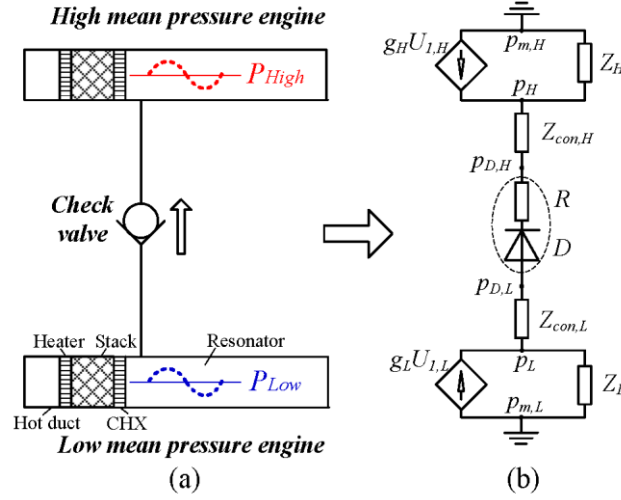


Fig. 1. Schematic and acoustic-electric analogy diagrams of a thermoacoustic compressor. Subscripts H and L are used to distinguish the high mean pressure engine and the low mean pressure engine, respectively.

However, the alternating gas flow in the thermoacoustic engines cannot turn into a direct gas flow spontaneously. As is well known, an alternating current can be converted into a direct current if a diode is introduced into the alternating current circuit. The check valve, which is modeled as a combination of a diode D and a variable resistor R , acts just as a “flow rectifier” that converts the alternating gas flow inside the thermoacoustic engines to the direct gas flow of the gas compression system. Whether the check valve is opened or closed is determined by the pressure difference between upstream oscillating pressure $p_{D,L}$ and downstream oscillating pressure $p_{D,H}$. The pressure difference $\Delta p = p_{D,L} - p_{D,H}$ and the volume flow rate U through the check valve obey the function below:⁹

$$\Delta p = \begin{cases} \Delta p_0 / R_a^2 & \text{if } \Delta p < \Delta p_c, \text{ fully closed} \\ \Delta p_0 + \Delta p_c & \text{if } \Delta p > (4R_a^2 + 1)\Delta p_c, \text{ fully open,} \\ \text{smooth transition} & \text{otherwise} \end{cases} \quad (1)$$

where Δp_0 is calculated from $\Delta p_0 = K F_{mult} \frac{\rho U^2}{2A^2}$. ρ , A , K and F_{mult} denote gas density, flow area of the check valve, local loss coefficient and empirical multiplier for viscous pressure drop, respectively. R_a , generally a positive number $\ll 1.0$, is the ratio of fully-closed to fully-opened flow area of the check valve. Δp_c , used to model the forward pressure drop of the diode D , is the pressure difference required to open the valve. Because of the oscillating nature of $p_{D,L}$ and $p_{D,H}$, the value of Δp varies periodically. When Δp is

smaller than Δp_c , there still exists an extremely small volume flow rate U due to valve leakage, which is also true for a diode, and Δp should be equal to $\Delta p_0 / R_a^2$ that is caused by the resistor R . Once Δp exceeds $(4R_a^2 + 1)\Delta p_c$ whose value is very close to Δp_c , the check valve is opened fully just as a diode conducts once the voltage exceeds the forward voltage, and Δp should be the sum of Δp_0 caused by the resistor R and the opening pressure difference Δp_c . When Δp is between Δp_c and $(4R_a^2 + 1)\Delta p_c$, flow resistance changes smoothly but rapidly to the fully-open value.

The compression process of the thermoacoustic compressor in Fig. 1 is simulated based on the basic theory mentioned above. The volumes of the high and low mean pressure engines are 6.3 L and 5.3 L, respectively. The initial mean pressures of the thermoacoustic engines are set to be 2.4 MPa, and the amplitudes of p_L and p_H are assumed to be a constant value of 0.1 MPa. The working frequencies of the high and low mean pressure engines are 40 Hz and 60 Hz, respectively. The maximal inner diameter of the check valve is 0.006 m. As shown by the variation trends of mean pressures in Fig. 2, the gas is compressed and transported from the low to the high mean pressure engine within several seconds. The highest instantaneous pumping rate reaches up to 150 Nm³/h at the beginning of the compression process and the average pumping rate during the 6 seconds is 2.8 Nm³/h, which is comparable with today's mechanical gas compressor. The pressure difference, which is 0.18 MPa now, can be increased by using more or improving the output capacity of thermoacoustic engines.

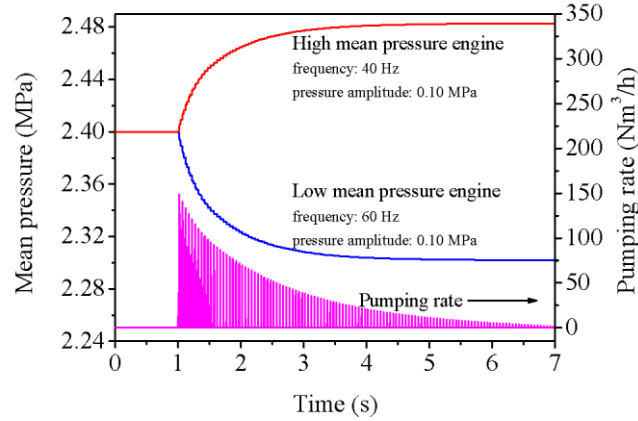


Fig. 2. Simulation of the compression process of the thermoacoustic compressor.

If the thermoacoustic compressor composed of two thermoacoustic engines shown in Fig. 1 is a part of a thermoacoustic compression system, the mean pressures of the thermoacoustic engines will keep constant in steady operation. The exergy efficiency of the thermoacoustic compressor can be defined as the theoretical minimum work requirement divided by the availability of the heat input,

$$\eta_e = \frac{\dot{W}_i}{\dot{Q} \left(1 - \frac{T_0}{T_h} \right)}, \quad (2)$$

where $\dot{Q}$ is the heat power input at temperature T_h , and T_0 is the ambient temperature, i.e. 300 K. The theoretical minimum work required to compress the gas from the low to the high mean pressure thermoacoustic engines is given by the isothermal compression process,

$$\dot{W}_i = \bar{m} R_g T_0 \ln \frac{p_{m,H}}{p_{m,L}}, \quad (3)$$

where $\bar{m} = \frac{1}{\tau} \int_0^\tau \dot{m} dt$ is the mean mass flow rate through the check valve, and can be calculated using Eq. (1) with given pressure amplitudes and mean pressures. Neglecting the acoustic power dissipation in the connecting tube, the heating power can be calculated by,

$$\dot{Q} = \frac{\dot{E}_{r,L} + \dot{E}_{r,H} + \dot{E}_{cv}}{\eta}, \quad (4)$$

where $\dot{E}_{r,L}$ and $\dot{E}_{r,H}$ are the acoustic powers entering the resonators of the low and high mean pressure engines, $\dot{E}_{cv}$ is the acoustic power dissipation due to the pressure drop through the check valve, and η is the thermal efficiency of the thermoacoustic engines. The acoustic power entering the resonator is dissipated in the resonator mainly in the forms of viscous and thermal relaxation losses within the linear thermoacoustics range.⁸ The dissipated acoustic power $\dot{E}_r$ in a half-wavelength standing-wave engine such as is shown in Fig. 1 can be estimated by,¹⁰

$$\dot{E}_r = \frac{1}{4} \left(\frac{p_1}{p_m} \right)^2 \frac{p_m a}{\gamma} \pi^2 R \left[\delta_\kappa (\gamma - 1) \left(1 + \frac{4Rf}{a} \right) + \delta_\nu \right], \quad (5)$$

where γ is the ratio of specific heats, a is the speed of sound, p_m is the mean pressure, f is the acoustic frequency, R is the radius of the resonator, and p_1/p_m is defined as the relative pressure amplitude. δ_ν and δ_κ are the viscous penetration depth and the thermal penetration depth, respectively. The acoustic power dissipation through the check valve $\dot{E}_{cv}$ can be calculated by,⁸

$$\dot{E}_{cv} = \overline{\Delta p \cdot \dot{U}} = \frac{1}{\tau} \int_0^\tau \frac{2R_g T \dot{m} \Delta p}{(p_{D,H} + p_{D,L})} dt \quad (6)$$

The dissipated acoustic power is balanced by the power production in the thermoacoustic stack from input heat power.

Consider a possible working condition with nitrogen as a working gas, where the relative pressure amplitudes p_1/p_m of the thermoacoustic engines are both 10%, and the thermal efficiencies η are both 15% with the heating temperature T_h at 900 K. The mean pressures of the low and high mean pressure thermoacoustic engines are 2.30 MPa and 2.50 MPa, respectively. The corresponding penetration depths ($\delta_\nu, \delta_\kappa$) of the 60 Hz and 40 Hz thermoacoustic engines are (61.06, 71.96) μm and (71.73, 84.54) μm , respectively. The radius of the resonators are 16 mm and 19.5 mm. According to Eq. (5), the acoustic powers dissipated in the resonators are $\dot{E}_{r,L} = 20.64$ W and $\dot{E}_{r,H} = 32.10$ W. The acoustic power dissipated in the check valve calculated by Eq. (6) is $\dot{E}_{cv} = 53.60$ W. Thus, the heating power needed is $\dot{Q} = 708.93$ W. The theoretical minimum work requirement is $\dot{W}_i = 67.33$ W with a mean flow rate of 26.12 Nm^3/h . According to Eq. (2), the exergy efficiency is 14.25%.

As analyzed above, by using efficient thermoacoustic engines, and check valves with low flow resistance, small opening pressure difference, and short response time, the efficiency of a thermoacoustic compressor can be improved greatly.

III. EXPERIMENTAL VERIFICATION

In order to verify the above ideas, a thermoacoustic compressor composed of two standing-wave thermoacoustic engines and two reservoirs was constructed and tested, as

shown schematically in Fig. 3. These two thermoacoustic engines, whose resonators are roughly of a quarter wavelength and 3/8 wavelength, operate at 60 Hz and 40 Hz and are chosen as the low and high mean pressure engines, respectively. The main components of the engines and their dimensions are listed in Table I. Both reservoirs have the same volume of 10 liters. Nitrogen is used as the working gas. Check valves V1 and V3 with an opening pressure of 0.01 MPa are supplied by HEROSE (type: 05011.X.0300), and check valve V2 with an opening pressure of 0.003-0.021 MPa is supplied by Swagelok (type: SS-6C-MM-1/3). Eight calibrated piezoresistive pressure sensors with an accuracy of $\pm 0.3\%$ and two calibrated NiCr-NiSi thermocouples are installed to measure pressures and heating temperatures, respectively. The ball valves are opened one by one when the thermoacoustic engines work steadily with heating temperature at approximately 650 °C, and gas flows in the direction as shown by arrows in Fig. 3.

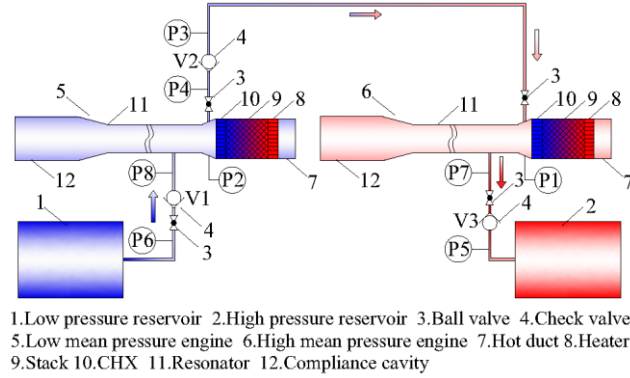


Fig. 3. Schematic of the experimental setup. P1, P2, P3, P4, P5, P6, P7, and P8 show pressure sensor locations.

TABLE I. Geometric parameters of the thermoacoustic engines. (Note: dimensions are listed as inner diameter $\times$ length in millimeter.)

Components	Low mean pressure engine	High mean pressure engine
Hot duct	50 $\times$ 70	59.5 $\times$ 60
Heater	50 $\times$ 80	59.5 $\times$ 50
Stack	50 $\times$ 60	59.5 $\times$ 110
CHX	50 $\times$ 60	59.5 $\times$ 40
Resonator	32 $\times$ 1050	39 $\times$ 2600
Compliance cavity	81 $\times$ 850	85 $\times$ 500

Simulation shows that increasing the pressure oscillation amplitudes at the two ends of a check valve is beneficial to the compression and pumping effect. According to the acoustic field of a standing-wave thermoacoustic engine, the pressure amplitudes are nearly the largest at the locations of P1 and P2. However, the performance of the thermoacoustic engines will be seriously affected if we set all the connection points here due to the disturbance of the mean gas flow to the acoustic oscillation. Furthermore, according to the transmission characteristics of acoustic transfer lines,^{11,12} the connecting tube between the thermoacoustic engines and check valves can amplify the pressure amplitude when it is of a specific length. Theoretically, the amplification factor is maximized when the length of the connecting tube is about 1/4 wavelength which is 2.2 m for the high mean pressure engine. However, when the length of the connecting tube is 2.2 m exactly, the pressure amplitude at P3 is only 0.02 MPa while the pressure amplitude at P1 is 0.05 MPa. It indicates that much acoustic power is dissipated by the transfer line.¹¹ Taking all the factors into consideration, the connections to the reservoirs are set between the midpoints of the resonators and the CHXs where the pressure amplitudes are large enough and the impact of the acoustic components on the

performance of the thermoacoustic engines is relatively small, as shown in Fig. 3. By experimental optimization, the length of the connecting tube between the high mean pressure engine and check valve V2 is finally chosen to be 3 m. The pressure amplitude at P3 reaches 0.14 MPa corresponding to a pressure amplification factor of 1.29 when the mean pressure is 2.51 MPa.

Fig. 4 shows the variation of mean pressures and gas pumping rate from the low pressure reservoir to the low mean pressure engine with an initial mean pressure of 2.4 MPa in all components. The gas pumping rate is calculated according to the ideal gas equation of state. The heating powers are 1064 W and 632 W for the low and high mean pressure thermoacoustic engines, respectively. The mean pressures of the four parts of the thermoacoustic compressor depart from each other immediately after the ball valves are opened. The pressure difference between the two reservoirs increases and finally reaches 0.4 MPa. It takes about 122 seconds to achieve 90% of the final pressure difference, and the average pumping rate during this period is 0.51 Nm³/h. It is worth mentioning that the average pumping rate during the first 3 seconds reaches up to about 2.85 Nm³/h after the compression process starts. However, the experimentally achieved pumping rate is still far below the theoretical value. This may result from the following reasons. Firstly, the pressure waveforms of the upstream and downstream pressure waveforms of the check valves are seriously deformed during the gas pumping process. Secondly, the measured pumping rate of 2.85 Nm³/h is not the instantaneous value exactly at the moment when the pumping process starts because of the measurement method of flowrate. Thirdly, the performance of the check valves used in experiments are much worse than those ideal ones adopted in the simulation.

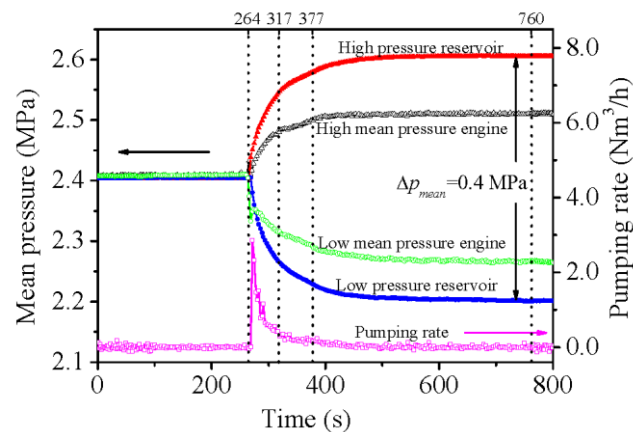


Fig. 4. Variations of mean pressure and instantaneous pumping rate during compression.

Fig. 5 shows the waveforms of pressure oscillations before and after check valves V2 and V3, and the timing are denoted accordingly in Fig. 4. As shown, as a result of the superposition effect of the alternating and the direct gas flows, pressure waveforms deviate from the sinusoidal shape drastically at the moment when the ball valves are just opened and the direct gas flow rate is the highest. The deformation of the pressure waveforms decreases the effective pressure difference across the check valve, which has a negative effect on the pumping rate. When the gas compression and pumping process halts at approximately 760 s, the pressure waveforms at P4 and P7 resume sinusoidal shape. Furthermore, it is shown that the gas transfer stops even though the instantaneous pressure of the low mean pressure engine side is higher than that of the high mean pressure engine side during some periods. This is due to the relatively large opening pressure difference and the limited response speed of the check valves. The response speed of a check valve is determined by the inertia and friction of the mechanical components as well as the pressure difference.^{13,14}

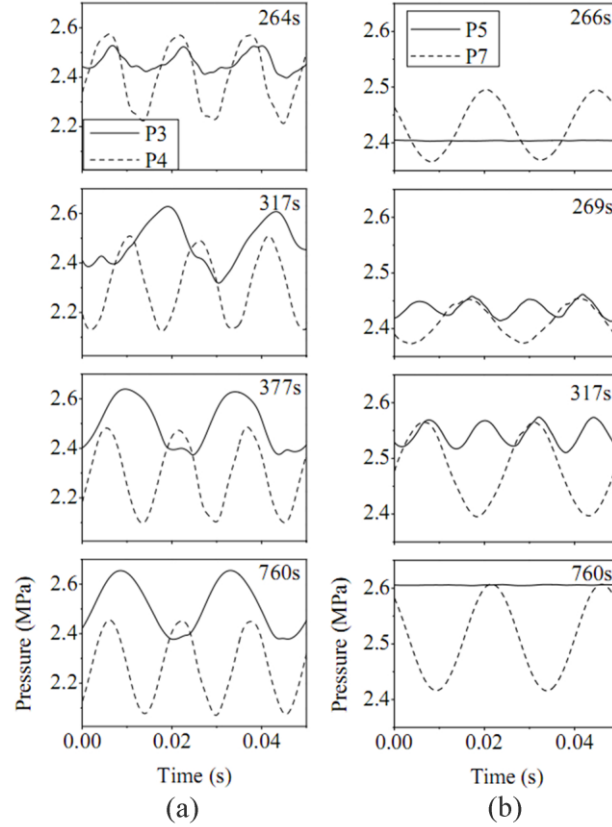


Fig. 5. (a) Pressure waves before and after check valve V2; (b) pressure waves before and after check valve V3. Because of the time consumption to open the ball valves, check valves V2 and V3 start at 264 s and 269 s, respectively.

IV. CONCLUSIONS

So far, a type of thermoacoustic compression effect has been proposed. It converts the alternating flow in thermoacoustic engines into direct flow and realizes the function of gas compression and transportation. A demonstrative thermoacoustic compressor composed of two standing-wave thermoacoustic engines was manufactured to verify its feasibility. With all components at an initial pressure of 2.4 MPa, a pressure difference of 0.4 MPa between the high and low pressure reservoirs has been achieved. The highest pumping rate reaches about $2.85 \text{ Nm}^3/\text{h}$ at the beginning of the gas pumping process.

It should be noted that this type of thermoacoustic compressor can be extended conveniently both by integrating more thermoacoustic engines to realize a much larger usable pressure difference and by increasing the output capacity of the thermoacoustic engines to achieve a much higher pumping rate. Furthermore, if an expander is combined with the compressors, a closed cycle solar power generation system and cooling system can be formed.

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