



# Operating characteristics of thermoacoustic compression based on alternating to direct gas flow conversion



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## ABSTRACT

A thermoacoustic compressor is capable of converting an alternating gas flow to a direct one with a large pumping rate on the basis of the pressure oscillation nature of thermoacoustic engines and the flow rectification effect of check valves. Theoretical calculations are first carried out to study the factors that affect the performance of the closed and open thermoacoustic compression systems. It is shown that the frequencies of directly connected thermoacoustic engines should avoid small integer multiple relationships to operate efficiently. Increasing the pressure amplitudes is beneficial for the pressure lift in a closed system as well as the pumping rate in an open system. A demonstrative closed thermoacoustic compressor was then experimentally studied. A maximum average gas pumping rate of 4.55 Nm<sup>3</sup>/h during the first 2 s of the compression process was achieved when all components were at the same initial mean pressure of 2.13 MPa. The maximum pressure lift reached 0.4 MPa when the initial mean pressure was 2.4 MPa. It was found that the pressure lifts were roughly proportional to the pressure amplitudes. Due to the superposition of alternating and direct gas flows, deformation of pressure waveforms which has a negative effect on the performance was observed.

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## 1. Introduction

Both alternating and direct flows are widely used in industries, such as in reciprocating and centrifugal compressors. In some applications, the two flow types even coexist. Taking G-M cryocooler [1] as an example, the gas flow supplied by the scroll helium compressor is direct flow while what required by the cold head is an alternating one. The rotary valve between the helium compressor and the cold head converts the direct gas flow into an alternating one by periodically switching between the high and low pressure ports of the compressor.

In recent years, the technology of flow type conversion has given birth to some novel promising applications. Recently, a mean flow acoustic engine composed of a main pipe and two single end closed branch tubes was proposed to induce acoustic oscillation from the mean flow (direct flow) in the main pipe [2–6]. Based on this conversion technique, aerodynamically driven thermoacoustic refrigerators and piezoelectric generators can be developed for mean flow energy (e.g. wind energy, fluid energy in pipelines) exploitation. With the development of smart materials, piezoelectrically

driven hydraulic micropumps have been proposed for applications such as chemical process control, drug delivery system, and cooling of tiny electric element [7–9]. The hydraulic pump utilizes reed valves, diffuser nozzles and active piezoelectric discs to convert the reciprocating motion of a piezoelectric actuator to a pumping action. In 2004, a resonant self-pumped circulating thermoacoustic heat exchanger was presented by Swift and Backhaus [10] to exchange heat with a remote heat source. Hydrodynamic asymmetrical constrictions were adopted as an imperfect gas diode to generate substantial mean flow out of a thermoacoustic engine. Later, check valves were adopted by Gao et al. [11] to realize the conversion from sinusoidally alternating to direct gas flow in a non-resonant self-circulating thermoacoustic heat exchanger.

Recently, we proposed a remarkable thermoacoustically driven compression effect based on the conversion of gas flow from an alternating state to a direct one [12–14]. Due to the unique features of thermoacoustic engines [15–18], thermoacoustic compression systems have the advantages of simple structure, thermally driven nature, oil-free operation and large operating range, and may find applications in gas compression, power generation, and refrigeration systems. If an expander is introduced further, a closed cycle solar power generation system or cooling system can be formed [14]. Besides, because of the already-demonstrated feasibility of micro-miniaturized thermoacoustic engines [19] and check valves

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[20–22], micro thermoacoustic compressors or pumps may find potential applications in MEMS field in the future.

In the previous paper [12], the thermoacoustic compression process is first proposed and a demonstrative thermoacoustic compressor verifies the feasibility of the concept. In order to reveal the operating principles and show the basic guidelines for the design of a thermoacoustic compression system, a detailed study of the working characteristics of closed and open thermoacoustic compression systems is carried out. The effects of operating frequency, pressure amplitude, phase difference, and mean pressure on the performance of such systems are presented in detail.

## 2. Computation and analysis

Fig. 1(a) shows the schematics of two types of thermoacoustic compression system, i.e., a closed and an open thermoacoustic compression systems. The CTCS (closed thermoacoustic compression system) consists of two or more thermoacoustic engines and in-between check valves, as denoted in the dashed box. The number of thermoacoustic engines is always one more than that of check valves in CTCS. The OTCS (open thermoacoustic compression system) consists of an arbitrary number of thermoacoustic engines and one more check valves. Unlike the CTCS inside which gas is always restricted, the OTCS continuously pumps gas from a low

are calculated. For the OTCS, as the gas is continuously supplied to the LMPE and discharged from the HMPE, the mean pressures stay constant. Thus, the pumping rate through the check valve is what applications concern. Therefore, a basic OTCS unit consisting of a check valve and two thermoacoustic engines at constant mean pressures is studied, and the average pumping rate through the check valve is calculated. The average pumping rates in both CTCS and OTCS in the computations are calculated via,

$$U = \frac{\int_0^\tau \dot{U} dt}{\tau} = \frac{\int_0^\tau \frac{\dot{m} R_g T_0}{p_0} dt}{\tau} = \frac{\dot{m} R_g T_0}{p_0 \cdot \tau}, \quad (1)$$

where  $R_g$ ,  $T_0$ , and  $p_0$  are the gas constant, temperature, and pressure under standard conditions, respectively.  $\dot{m}$  and  $\tau$  denote the total gas mass transported from the LMPE to the HMPE and the time needed to complete the compression process in CTCS. While in OTCS,  $\dot{m}$  denotes the total gas mass transported in the calculation time  $\tau$ , which is 10 s in the calculation.  $\dot{m}$  and  $\dot{U}$  denote the instantaneous mass flow rate and volumetric pumping rate through the check valve respectively.

The instantaneous mass flow rate  $\dot{m}$  is determined by the pressure difference across the check valve  $\Delta p$ , and is calculated by Refs. [12,23]:

$$\dot{m} = \begin{cases} \sqrt{\Delta p \frac{2\rho A^2 R^2}{KF_{\text{mult}}}} & \text{if } 0 < \Delta p < \Delta p_c, \text{ fully closed} \\ \sqrt{(\Delta p - \Delta p_c) \frac{2\rho A^2}{KF_{\text{mult}}}} & \text{if } \Delta p > (4R^2 + 1)\Delta p_c, \text{ fully open,} \\ \left(1 + \frac{1}{4R^2} \left(\frac{\Delta p}{\Delta p_c} - 1\right)\right) \sqrt{\Delta p_c \frac{2\rho A^2 R^2}{KF_{\text{mult}}}} & \text{otherwise} \end{cases} \quad (2)$$

pressure gas source to a high pressure one. Check valves are set at the downstream and the upstream sides of the low and high pressure gas sources, respectively. CTCS can easily demonstrate the working principle of the thermoacoustic compression, while OTCS is more likely to be used in industry where continuous gas compression and pumping are required. The arrows in the figure show the directions of the gas flow. Due to the thermoacoustic conversion effect, the pressures  $p_H$  and  $p_L$  in the HMPE (high mean pressure engine) and the LMPE (low mean pressure engine) oscillate periodically with mean pressures  $p_{H,m}$  and  $p_{L,m}$ , respectively, as shown in Fig. 1(b). As denoted by the shadows, the pressure  $p_L$  in the LMPE can be higher than the pressure  $p_H$  in the HMPE at some moments due to pressure oscillation. The check valve enforces a one direction gas flow from the LMPE to the HMPE when the pressure difference  $\Delta p = p_L - p_H$  exceeds the opening pressure difference  $\Delta p_c$  of it. This is the main operating principle of a thermoacoustic compressor.

To simplify calculation, both the basic CTCS and OTCS units with nitrogen as working gas are studied. The basic CTCS unit consists of two thermoacoustic engines and a check valve. The volumes of the LMPE and the HMPE are 6.3 L and 5.3 L, respectively. The initial mean pressures of the LMPE and the HMPE are both set at 2.4 MPa unless otherwise stated. Due to the thermoacoustic compression effect, the gas in the LMPE will be transported to the HMPE until the transient pressure difference  $\Delta p$  across the check valve can not exceed  $\Delta p_c$  anymore. The final pressure lift  $\Delta p_m$  and pumping time

where  $\rho$ ,  $A$ ,  $K$  and  $F_{\text{mult}}$  denote gas density, flow area of the check valve, local loss coefficient and empirical multiplier for viscous pressure drop, respectively.  $R$ , generally a positive number  $\ll 1.0$ , is the ratio of fully-closed to fully-opened flow area of the check valve. In the present calculations,  $K$ ,  $F_{\text{mult}}$ , and  $R$  are set as 2.8, 1, and 0.01, respectively. The opening pressure difference  $\Delta p_c$  and the maximal inner diameter of the check valve are set at 0.01 MPa and 6 mm, respectively. To simplify the analysis, it is assumed that the pressure waveforms are sinusoidal and the response time of check valve is short enough to be ignored. The pressure waveform of the HMPE leads that of the LMPE by a phase  $\theta$ . The time step is set at  $1 \times 10^{-6}$  s.

The direct driving sources of the gas compression and pumping in thermoacoustic compression systems are the pressure oscillations in thermoacoustic engines excited by the input thermal energy. With the thermoacoustic energy conversion effect, the input thermal energy is converted into acoustic power, which is then consumed mainly in the following ways: gas pumping through check valves, viscous and thermal-relaxation effects in thermoacoustic engines. Thus, the efficiency of the whole system is the product of thermal-to-acoustic efficiencies of thermoacoustic engines and the compression efficiency of the compression process. The thermal-to-acoustic efficiencies are strongly influenced by the design and the operation of the thermoacoustic engines. Typically, the thermal-to-acoustic efficiency of a well-designed standing-wave thermoacoustic engine can reach about 0.15–0.20 [24]. As this study mainly focuses on the characteristics of gas compression

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