Free piston pulse tube engine: a numerical and experimental power generation estimation

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Abstract. The results of theoretical and experimental study of the atmospheric pulse tube engine (PTE), which is a kind of thermal inertial Stirling engine, are presented. The main advantage of the PTE is the absence of the moving design elements in heated part make it a perspective technology for waste heat recovery. Firstly, numerical simulation was conducted to design the system and optimum structure parameters of the system were obtained as well as the values of the free-piston PTE resonant frequency, the levels of the input and rejected heat, and the net power. Comparison of the net power and the thermal efficiency of the pulse tube engine with the crank mechanism and the free piston is provided. A comparison of operating conditions and engine start conditions for different free piston masses and temperatures in the hot heat exchanger is performed. The results are compared to prior numerical studies. The maximum value of the net power of the PTE was achieved for the frequency of 11.8 Hz, the input thermal power of 160 W and the temperature of 640 K.

1. Introduction

Unlike traditional Stirling engine schemes, the pulse tube engine (PTE) does not have a piston in the hot part, this simplification of the design provides long life and reliability. It has a high potential to be used as a prime mover for the conversion of low grade waste heat into mechanical or electrical energy. Despite numerous attempts to simulate the working process of PTE [1, 2, 3] and the availability of prototypes of various design schemes, the work process theory of the engine is not fully disclosed, and experimental studies are limited [4, 5]. In the strict sense, it is not determined to which category of the process models implemented in the PTE should be attributed: thermoacoustic [6] or close to thermodynamic similar to Stirling [5].

The working process of the engine based on the thermodynamics of the variable mass of gas is presented in [2; 5]. The working process model is presented as the interaction between the hot and cold volumes of the mass and the enthalpy of the flow. There are also hybrid models where there is no flow rate of the working gas [1] in the PTE working process, the gas moves without mixing, the thermodynamic process and generation of useful work in the engine are carried out due to the surface heat and mass transfer near the walls of the pulse tube, which was estimated using the temperature profiles of the walls and gas. The approach based on the principle that the axial temperature gradient along the channel can amplify acoustic oscillations, wherein the thermal power is converted into a cycle operation, if implementation of the compression and expansion processes occurs at such a time.
scale that presented in [7]. The internal irreversible thermodynamic cycle in PTE is similar to the standing-wave thermoacoustic engines, which is proved in [6] and can be described by the equations of linear thermoacoustics. Thus, in the thermoacoustic aspect, the piston motion occurs due to the generation of pressure oscillations occurring during the heat transfer between the hot and cold ends of the regenerator and the pulsating tube with a phase delay determined by the viscosity forces in the boundary layer corresponding to the propagation of a plane standing wave.

Whilst both thermodynamic and hybrid models provide a qualitative explanation for why the pulse tube engine works, unfortunately presented experiments are not sufficient to prove thermodynamic concept and numerical simulation does not correlate with experimental results. Therefore, the understanding of the thermodynamic phenomena that allows the PTE to produce work remains uncertain, and will only be disclosed through a numerical and experimental investigation as Organ suggests [1]. In some case studies, there are no significant matches between the results of numerical simulation using this approach and experimental data. Experimental data are presented as the comparison of the operating parameters for the modes of operation of the PTE with the crank mechanism and with the free piston.

The aim of this study was to compare the results of the theoretical numerical simulation of free piston PTE based on linear thermoacoustics with experimental data.

2. System configuration

The main design scheme of the pulse tube engine (PTE) is presented in figure 1.

![System configuration diagram](image)

**Figure 1.** The pulse tube engine (PTE) scheme with Delta EC simulation model's elements.

Heat is supplied through a hot heat exchanger (HHX). At the ends of the thermal buffer tube (pulse tube) are hot and cold heat exchangers (CHX1). The existence of a temperature gradient on the regenerator is provided by hot and cold (CHX2) heat exchangers located at its ends. The piston performs a dual role: net power generation and maintaining the oscillating process.

The conversion of the compression/expansion of the working gas into mechanical power and vice versa is carried out by the influence of variable pressure. The generator for the conversion of mechanical energy into electrical energy can be connected to the piston via a crank gear mechanism, or in the form of a linear electric generator. At the same time, there are no mechanisms in the PTE responsible for regulating the processes of compression-expansion of gas similar to Stirling. Due to the features of the processes of gas dynamics and heat transfer in the PTE, there is a self-regulation of pressure fluctuations and volume velocity with phase delay, which allows obtaining mechanical work.

3 Simulation details

The simulation was conducted with DeltaEC, which is based on the classical thermoacoustic theory [8]. Software provides a series of modules, such as, cylindrical or conical ducts, slot type of heat exchanger, regenerators, compliances, thermal buffer tube and electromechanical transducers as concentrated impedance for free piston simulation et al, to simulate different thermoacoustic components and systems. Initially some parameters such as gas temperature, pressure amplitude and phase are set and initialized by the user. A shooting method is introduced in the software to satisfy the
boundary conditions. In Delta EC it is realised by varying and fixed parameters called guesses and targets; all values of them are set by the users. Piston mass, heat exchanger temperatures pressure amplitude was set as guesses. The open end of the cylinder with the piston was set as a SOFTEND.

The PTE working process calculation was carried out via integration of the linear theory of thermoacoustic equations. The aim is to determine two complex variables for an engine segment: pressure amplitude $p_1(x)$ and volume velocity $u_1(x)$, and two real variables: gas temperature $T_m(x)$ and enthalpy $H_{tot}(x)$. Performing the integration from one segment to another, the condition of equality of parameters on the element bound is imposed. In the simulation, the working fluid is air and mean pressure is set to be 0.1 MPa. In the main and secondary ambient heat exchangers, the solid temperature is set to be 300 K. The solid temperature of the high temperature heat exchanger is set to be 650 K. Heat transfers with the environment are set to be 0 for most components except for the heat exchangers. With the above initial and boundary conditions, simulations can be carried out.

The value of the heat input and output in the heat exchangers, the working frequency, pressure amplitude, and free piston mass were initially estimated and then varied during the calculation as assumptions. The following parameters were taken as target values in the "shooting" method used in Delta EC: the wall temperatures values in heat exchangers and zero boundary conditions of real $Re(Z)$ and imaginary $Im(Z)$ parts of the impedance, the total flow enthalpy $H_{tot}(x)$. The relative Carnot efficiency, defined in equation (1), was taken as the optimization target:

$$h = \frac{W_{ac}}{Q_h} \frac{T_h-T_c}{T_h}$$

where $W_{ac}$ is produced acoustic power, $Q_h$ is the heat input.

4. Simulation results and discussions

4.1. Geometry parameters

The main optimized dimensions of a free piston PTE are as follows. In the PTE the secondary ambient heat exchanger, regenerator, high temperature heat exchanger have a common diameter of 45 mm and the lengths of 20 mm, 105 mm, 2,5 mm, respectively. The regenerator is a stack of stainless steel wire mesh with a porosity of a 0.67 and hydraulic radius of 100 $\mu$m. The pulse tube which connects high temperature heat exchanger and the cylinder with power piston is 16 mm in diameter and 450 mm in length. The piston diameter obtained in simulation is 40 mm and the piston mass for which the maximum Carnot effectiveness of the PTE was produced is 55 g. The value of the working frequency, obtained in the calculation is 11.8 Hz.

The distribution of acoustic power along the length of the PTE obtained by the simulation in Delta EC (figure 2) shows that the pulse tube is an element in which there is an increase in acoustic power, equivalent to the net work of the cycle.

![Figure 2. Axial distribution of total gas enthalpy and acoustic work in PTE.](image-url)
4.2 Experimental verification and discussion

The simulation model is used to design an experimental test engine. The aim of the experimental set-up is to validate the simulation model. Furthermore, the experimental test engine is also used to study the pulse tube engine performance for different designs and operation conditions (figure 3), providing measurement of gas temperature in hot and cold heat exchangers, indicating the pressure in the compression space in cylinder; current position and piston stroke measurement.

Figure 3. Experimental pulse tube engine setup.

Pressure recording (indication of the compression space) was carried out by the sensor U11323 (3B Scientific) with relative measurement error of 0.6%. The piston current position was recorded using a laser meter Balluff BOD 26K, 0.2% max. relative error, synchronously with the pressure recording digital oscilloscope AKIP. The response time of the oscilloscope was allowed getting up to 2000 points per cycle (one work cycle of engine) that allows obtaining a solid line of pV-work diagram. Net work was calculated using experimental data distributions in MATLAB. The results of measuring the indicator work of the cycle are 11.8 ± 0.044 joule. This extended uncertainty is the product of the standard uncertainty and the coverage factor based on the assumed normal distribution and is defined an interval corresponding to a coverage probability of 95% approximately. The engine was run at $T_h = 650 \, ^\circ C$, $T_c = 25\, ^\circ C$ and atmospheric pressure. The power input was measured from the electrical power consumption of the heater placed inside the hot heat exchanger. The measured thermal power input was between 40% (0.1 MPa, 16 Hz) and 95% (0.1 MPa, 10 Hz) higher than the calculated thermal power input. The tasks of the tests were to identify the mode-parametric conditions of start-up and stable operation; to obtain the distribution of the net power vs frequency (figure 4).

The indicator efficiency was determined with and without taking into account thermal leaks ("net" and "gross"): 

$$ h_{th, gross} = \frac{W_{net}}{W_{el}}, \quad W_{net} $$(2)

where the pV power of the cycle, $W_{el}$ the power on the heater,

$$ h_{th, net} = \frac{W_{net}}{Q_{hs}} $$ (3)

as the ratio of the pV power of the cycle to the supplied to the working fluid power

$$ Q_{hs} = W_{el} - Q_{ext} $$ (4)

where the $Q_{ext}$ external heat flux, loosed to the environment (figure 4). Mode-parametric conditions of start-up and stable operation correspond to the values:

1. the temperature ratio $\tau = T_h/T_c = 1.3...2$. Further increasing hot heat exchanger gas temperature $T_h$ results in net power decreasing. The increase slows down strongly for higher temperatures, such that the temperature has no significant influence on the efficiency anymore. This may be due to a limited amount of enthalpy transferred from the hot to the cold end of the pulse tube.
2. Another parameter characterizing the design of the pulse tube engine is the regenerator and pulse volume ratio \( r_d = \frac{V_{\text{reg}}}{V_{\text{pt}}} \). The developed PTE is characterized by a lower value of the distribution coefficient \( r_d = 2.44 \) in comparison with pulse tube engine in [5] \( (r_d = 3.4) \). The relative Carnot thermal effectiveness of the engine was obtained numerically using thermoacoustic model for the distribution coefficient range \( r_d = 3.2 \ldots 4.2 \). The simulation results shows that thermal effectiveness \( h_{th} \) decreases by reducing the amount of gas enthalpy transferred in the pulse tube. Figure 4 shows that the pulse tube engine considered in this paper is characterized by a 30-50% increase in the level of produced net power in comparison with the engine in [5]. At the same time, the filling pressure of the analogous engine is 0.4-1 MPa.

3. The maximum of the net power of the PTE corresponds to the resonance frequency of the free piston engine design depending on the amount of heat input \( Q_h \) and the mass of the free piston. According to the experimental results, it can be concluded that the values of the obtained net power of the developed engine with crank gear mechanism correspond to the power level of the prototype (figure 5) at a filling pressure below 4...10 times than in [5].
For the modes with free piston masses $m=57 \text{ g}$ and $m=52 \text{ g}$ net power of the developed engine exceeds the power level of the prototype by 30...40%. Based on the classical thermoacoustic theory, numerical simulation was performed to design and obtain optimum structure parameters of the system. The regime-parametric start-up conditions of the developed free piston PTE design were identified for the first time. Experimental and numerical results show the influence of the piston mass on the net power level of the PTE. For the developed experimental design of the PTE the highest power value is 17 W with a piston mass of 57 g and a frequency of 11.8 Hz. The calculated and experimental values have a correlation coefficient of 0.92...0.95.

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