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Performance of V-type Stirling-cycle refrigerator for different working fluids

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ABSTRACT

The thermodynamic analysis of a V-type Stirling-cycle Refrigerator (VSR) is performed for air, hydrogen and helium as the working fluid and the performance of the VSR is investigated. The V-type Stirling-cycle refrigerator consists of expansion and compression spaces, cooler, heater and regenerator, and it is assumed that the control volumes are subjected to a periodic mass flow. The basic equations of the VSR are derived for per unit crank angle, so time does not appear in the equations. A computer program is prepared in FORTRAN, and the basic equations are solved iteratively. The mass, temperature and density of working fluid in each control volume are calculated for different charge pressures, engine speeds, and for fixed heater and cooler surface temperatures. The work, instantaneous pressure and the COP of the VSR are calculated. The results are obtained for different working fluids, and given by diagrams.

Diminution de la consommation d'électricité dans les systémes frigorifiques á plusieurs compresseurs

Mots clés : Systéme frigorifique ; Systéme á compression ; Compresseur á vis ; Modélisation ; Optimisation ; Régulation ; Vitesse variable ; Économie d'énergie

1. Introduction

The first Stirling-cycle machine was built by R. Stirling in 1816 as an alternative to the steam engine. The classical analysis of Stirling-cycle machine is given by Schmidt, and in the literature considerable efforts have been made to develop and improve this analysis using different working fluids (Walker, 1973).

Tew et al. (1978) analyzed the thermodynamic characteristics of the Stirling-cycle engine, and simulated by software. The model represents the working space by a series of

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subdivisions, which is called the nodal model. Urieli and Kushnir (1982) shown that their analysis can be utilized in order to evaluate the various practical effects of non-ideal regenerators, heat exchangers, including heat transfer and pressure losses.

Discussing the irreversibilities Berchowitz (1992) optimized a free-piston Stirling-cycle refrigerator with reference to designing machines for operation at intermediate temperatures. Walker et al. (1992) gave the brief review of the previous work for the Stirling-cycle refrigerators.

Angelino and Invernizzi (1996) showed that the heat pumps based on Stirling cycle are positively influenced by real gas effects. Kaushik and Kumar (2000) presented an investigation of a finite-time thermodynamic analysis of an endoreversible Stirling engine to maximize the power output and the corresponding thermal efficiency of an endoreversible Stirling heat engine with heat loss in the regenerator.

Otaka et al. (2002) developed a prototype of β-type Stirling-cycle refrigerator using helium, hydrogen and nitrogen as a working fluid. A Stirling-cycle machine of 100-W capacity was designed and tested. The effect of the dead space and phase angle on the refrigeration capacity was investigated.

Heidrich et al. (2005) developed a mathematical model for numerical simulations of free-piston Stirling coolers. As an alterative for conventional vapor compression cycles, a Stirling cooler for domestic refrigerators is developed. The model explores the working gas thermodynamic behavior and evaluates the performance of the Stirling cooler components.

There are a few studies about V-type Stirling-cycle Refrigerator in the literature. The V-type Stirling-cycle Refrigerator (VSR) is analyzed by Ataer and Karabulut (2005). In this study the Stirling-cycle refrigerator was divided into 14 fixed control volumes which were subjected to a periodic mass flow. The work, instantaneous pressure and COP of the Stirling-cycle refrigerator are calculated and the results are given by diagrams. Le'an et al. (2008) designed and tested the V-type Stirling refrigerator. The power consumption and the coefficient of performance (COP) are investigated under various rotating speeds and charged pressures.

In this study it is aimed to design a new V-type Stirling Refrigerator which has higher cooling capacity and the COP as a new alternative for the vapor compression-type refrigerator.

In this study the control volume analysis of the VSR is performed to determine the COP and cooling load of the VSR for the different working parameter. The refrigerator is divided into the control volumes. Heat is transferred to the refrigerator in the expansion space and heater, and transferred to the environment in the compression space and cooler. The VSR consists of a cylinder with a piston on each side of the regenerator as shown on Fig. 1 and the heater and cooler consist of multiple passes. For the storage of heat cupper wire mesh is used in the regenerator. As shown in Fig. 1, especially geometry of heater and cooler is designed to increase to the heat transfer.

Each control volume of the VSR is exposed to a periodic mass flow. The control volumes of compression and expansion spaces are variable as regards to the motion of the pistons and the other volumes are fixed. The conservation of mass, momentum and energy equations are written for each control volume of the VSR, a computer program is written in FORTRAN, and the equations are solved iteratively.

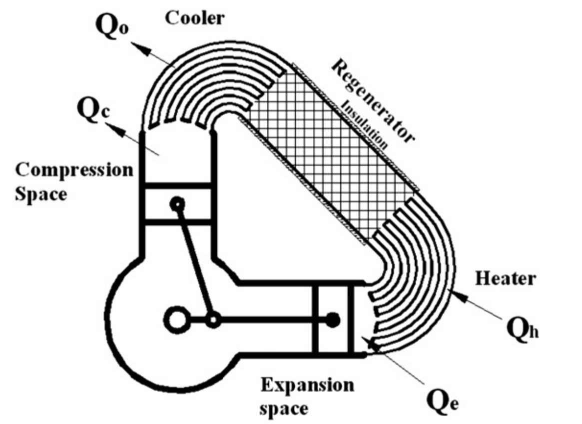

Fig. 1 – Schematic diagram of the VSR.

2. Theory

In order to simplify the analysis the following assumptions are made:

(a) There is no leakage and the total mass of the working fluid is constant within the VSR.
(b) The working gas is treated as an ideal gas within the working conditions.
(c) The angular speed of the VSR is constant.
(d) The cyclic conditions are established in the VSR
(e) The gas flow in the VSR is one dimensional.

The variable volumes of the expansion and compression spaces of the VSR are expressed respectively

$$V_e = V_{e,d} + \frac{1}{2}V_{e,sr}(1+\cos\theta) \tag{1}$$

and

$$V_c = V_{c,d} + \frac{1}{2}V_{c,sr}[1+\cos(\theta-\phi)] \tag{2}$$

where $V_{e,d}$ and $V_{c,d}$ are the dead volumes of the expansion and compression volumes respectively, φ is the phase angle of the compression space relative to the expansion space. The mass balance for a control volume is written as

$$\frac{dM}{d\theta} = m^{\theta}_{in,i} - m^{\theta}_{out,i} \tag{3}$$

where $m^{\theta}_{in,i}$ and $m^{\theta}_{out,i}$ are the mass flow per unit crank angle of control volume "i". The total mass of the VSR is constant and equal to

$$M_t = \sum_{i=1}^{n} M^{\theta}_i \tag{4}$$

where n is the number of the nodes. The flow in the control volumes of the VSR is compressible flow, and the simplified momentum equation is written as