Proceedings of the International Conference on Applied Mechanical Engineering Research (IC-AMER2019)
NIT Warangal, India – May 02-05, 2019
Paper No. XXX (The number assigned when the papaer is accepted)

Theoretical Analysis of Counter Flow Pulse Tube Refrigerator

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Pulse tube refrigerators with minimum vibration, large life span and more cooling capacity must be developed for applications like military and space. Latest designed pulse tube refrigerators have the capacity to achieve these demands. Pulse tube refrigerators perform better than other types of known refrigerators. In the last few years, interest in the pulse tube refrigerator is increased due to mechanically simple, highly reliable design. There are no moving parts at the cold end of pulse tube, which reduces vibrations in the pulse tube refrigerator system. Present paper deals with modification of Gawali model in which the reservoir is eliminated and replaced with another pulse tube refrigerator. These two pulse tube refrigerators are operated at 180 degree out of phase. Because of this, pulse tubes refrigerators are called as counter flow pulse tube refrigerator. Theoretical model is developed for the analysis of CFPTR i.e. counter flow pulse tube refrigerator. Cyclic analysis approach is considered for analysis. Modified approach of analysis is presented in the paper.

***Keywords*:** Counter flow pulse tube refrigerator, Theoretical analysis, Cyclic approach

1. Introduction

Gifford and Longsworth [1] discovered first pulse tube refrigerator (PTR), which have the potential to be widely applied for producing low temperatures. The main benefits of this system are the no presence of moving mechanical parts at the regions of low temperature. This lowers the induced mechanical vibrations significantly and strongly improves the reliability of the subsystems. From the invention of basic pulse tube refrigerator, researchers get attracted towards it to design new pulse tube refrigerator system which have capacity to achieve lower refrigeration temperatures. Based on shifting of phase between mass and pressure pulse, the new design variations came into focus, which includes orifice pulse tube refrigerator (OPTR) [2], double inlet orifice pulse tube refrigerator (DIOPTR) [3], and most recently the inertance tube pulse tube refrigerator (IPTR) [4].

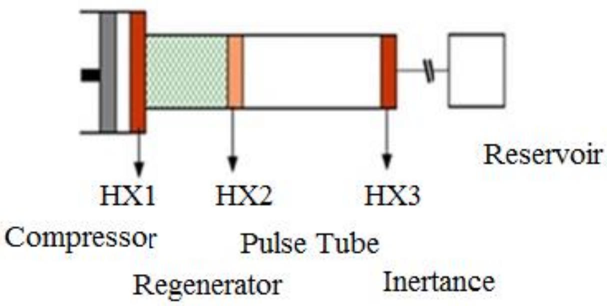

Fig. 1. Pulse tube refrigerator system

According to speed and valve arrangement PTR are classified as Stirling PTR and G-M PTR. These are mainly consisting of compression device (compressor), cyclic flow heat exchanger, regenerator (heat exchanger), cold end heat exchanger, pulse tube, hot end heat exchanger and phase shifting mechanism. The G-M pulse tube is consist of rotary valve and having low frequency flow; while Stirling type pulse tube has no valve mechanism and operating with high frequency.

Pulse tube refrigeration is defined as a continuous pressurization and depressurization of any closed volume with air or gas, sets up temperature gradient in the volume from hot end to pulse tube end where cooling obtained. The compressor head is modified to replace the valve assembly to get the direct pressure pulse as given in Fig.1.

It enclosed the two slotted heat exchangers made up of copper and fitted into water coolers. The regenerator consisting of a small thickness stainless tube which is meshed by stainless steel wire mesh. The pulse tube is also made of thin walled stainless steel tube. The recuperative heat exchangers are of copper. Thermodynamic cycle of Striling refrigerator and Stirling type PTR are more or less similar to each other and can be seen as a replacement of displacer by a gas piston i.e. the active phase shifter of the Stirling refrigerator is replaced by a passive phase shifter. Refrigerating effect is obtained by maintaining the appropriate phase shift between the pressure pulse and mass flow rate.

2. Counter flow pulse tube refrigerator system

The performance of pulse tube refrigerator depends upon the geometrical and operating parameters; also the performance of the systems decreases due to irreversibility in hot and cold end heat exchangers and regenerators [5]. Proposed system comprises two Stirling type pulse tubes. Fig. 2 represents the Stirling type Counter Flow Pulse Tube Refrigerator (CFPTR) system. The two identical pulse tubes are connected by an inertance tube at the hot end heat exchanger, eliminating the reservoir of OPTR. The two identical compressors operating in 180 degree phase shift develops the pressure waves. As the gas flows at 180 degree out of phase in the system it is called as CFPTR. A special feature of proposed system is that the pressure wave is generated by two identical compressor operating in 180 degree out of phase. Also, the orifice and reservoir in G-M type pulse tube are replaced by an inertance tube, which reduces the total volume of the system. The flows in the two systems are opposite at all times. The one tube is high pressure side and another tube is low pressure side. After every 180 degree of crank angle the flow gets reverse in direction, thus we have oscillating flow in the CFHX.

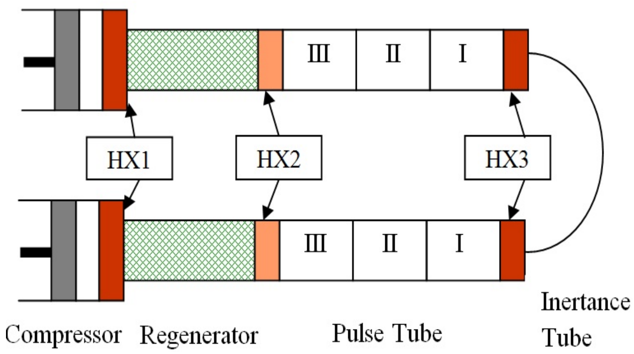

Fig. 2. Counter flow pulse tube refrigerator

3. Theoretical model

Proposed model is modification of Gawali et.al [6] model in which the reservoir is eliminated and replaced with another pulse tube refrigerator, as shown in Fig. 2. As the gas flows 180 degree out of phase in pulse tubes it is called as counter flow pulse tube refrigerator. Advantage of this arrangement is to eliminate the reservoir volume required in the orifice pulse tube refrigerator.

To carry out the cyclic analysis, a pressure volume cycle of 360^0 is divided in 12 small intervals. For each interval, the ideal gas law is applied and the equation is rearranged to give the pressure for the corresponding point of the interval. Calculating the flow rate through the orifice, mass balance is carried out for section I, II and III. Based on the volume variations in section I, II and III mass flow rates, the ideal refrigerating effect and ideal power input are calculated.

3.1 Pressure volume variation:

Here, two identical pulse tube refrigerator are connected through an inertance tube at the hot end exchanger, operating at 180 degree out of phase shift.

Pulse Tube set I:

The volume variation in the compression space of pulse tube set I varies sinusoidally and is given as

$$V_{c1} = V_{cd1} + V_{cm1}(1 + cos\Theta)/2 \tag{1}$$

Where θ is the crank angle, V_{cd1} is the clearance volume.

Pulse Tube set II:

The volume variation in the compression space of pulse tube set II varies sinusoidally and is given as

$$V_{c2} = V_{cd2} + V_{cm2}(1 + cos\alpha)/2 \tag{2}$$

Where $\alpha = (180 - \theta)$, V_{cd2} is the clearance volume.
The total mass of working fluid is

$$M = M_c + M_{dc} + M_r + M_{dc} + M_e \tag{3}$$

Using the perfect gas laws

$$P_1(1) = \frac{MR}{\frac{V_{c1}(1)}{T_{c1}(1)} + \frac{V_{dc1}}{T_{dc1}(1)} + \frac{V_{r1}}{T_{r1}(1)} + \frac{V_{dcd1}}{T_{dcd1}} + \frac{V_{e1}(1)}{T_{e1}}} \tag{4}$$

The magnitude of pressure, P_1 (1), and volume V_{c1} (1) and V_{e1} and temperature T_{c1} vary with crank angle θ. The temperature of expansion space Te1 is assumed as constant. At the starting, the temperature of cooler dead space and compression space is assumed as same. Temperature of regenerator is found to be equal to the log mean temperature difference of the two end temperatures. Assume the product of the universal gas constant, R and total mass in terms of moles of the gas, M i.e. (MR) to be one, as the value of MR is not known

$$P_1(1) = \frac{1}{\frac{V_{c1}(1)}{T_{c1}(1)} + \frac{V_{dc1}}{T_{dc1}(1)} + \frac{V_{r1}}{T_{r1}(1)} + \frac{V_{dcd1}}{T_{dcd1}} + \frac{V_{e1}(1)}{T_{e1}}} \tag{5}$$

Assuming, $\frac{\gamma - 1}{\gamma} = y$

$$T_{c1}(2) = T_{c1}(1)\left[\frac{P_1(2)}{P_1(1)}\right]^y \tag{6}$$

Now, substitute the values of T_{c1} (2), V_{c1} (2) and V_{e1}(2) in equation (4), in that equation only P_1(2) is unknown variable. Then value of the P_1(2) is used to find out P_1(3) and so on. Similarly, the pressures and temperatures values are calculated for the one complete cycle for the all intervals and one interval is θ. If value of θ is 30^0, then pressures and temperatures values of 1st are same as that of 13th interval. The pressures and temperatures at 1st and 13th point should be within the acceptance limit to show that the repetition of cyclic process; otherwise the process of calculation restarts form the first point.

Mean pressure, P_m would therefore be equal to

$$P_m = P_{total}/12 \tag{7}$$

The P_{avg} is known, and the ratio P_{avg} and P_m would give exact value or correct value of (MR). Previously, assumed value of MR is one. So, with this correct value of (MR), all the pressure values should be corrected so that P_m matches $P_{avg.}$

3.2 Calculation of mass flow rate through the inertance:

The mass flow rate through the inertance will depend on pressure in the PTR1 and PTR2.

For ptr1>ptr2

$$m_i = 0.25\times\pi\times itd^2\times\sqrt{\left(ptr1-ptr2\right)\times itd\times 2 \Big/ itl\times f\times V_{vhIII(1)}} \tag{7}$$

For ptr1<ptr2

$$m_i = -0.25\times\pi\times itd^2\times\sqrt{\left(ptr1-ptr2\right)\times itd\times 2 \Big/ itl\times f\times V_{vhIII(2)}} \tag{8}$$

Where f is friction factor and V_{vhIII} is specific volume of hot end heat exchanger

3.3 Mass flow rate at section I:

Mass flow from the section one depends on the mass flow through inertance and the pressure variation in the volume I.

$$M_I = M_i + \frac{V_{hIII}}{R\times T_c}\times\left(P\left(I+1\right)-P(I-1)\times 0.5\right)\times(\text{Time Interval}) \tag{9}$$

3.4 Calculation of volumes V_I, VII, VIII:

Hot end isothermal volume V_I is calculated as

$$V_I = \frac{M\times R\times T_c}{P} \tag{10}$$

Adiabatic part of the pulse tube V_{II}

$$V_{II} = C\times P^{\frac{1}{\gamma}} \tag{11}$$

The corrected value of C is obtained with V_{III} (min)=0.0

The cold end heat exchanger volume VIII

$$V_{III} = V_{PT} - (V_I + V_{II}) \tag{12}$$

3.5 Calculation of power input and refrigeration effect:

$$Q_{IN}(I) = \int P(I)dV_T\left(I\right)$$

$$Q_I(I) = \int P(I)dV_e\left(I\right) \tag{13}$$

$$V_T\left(I\right) = V_C(I) + V_D + V_e(1)$$

The system have ideal power input which is indicated by Q_{IN} and the cooling effect by Q_I, ideally available are calculated once the pressure variation for the complete cycle is known.

The above equation is solved by numerical method like trapezoidal rule. The pressure taken for each interval is the average pressure of the interval. The algebraic sum of the product of pressure and change in volume for every interval would give the total power and ideal cooling effect. The values of pressure and temperature for each interval is calculated and plotted graphs for the same and shown in Fig. 3 and Fig. 4.

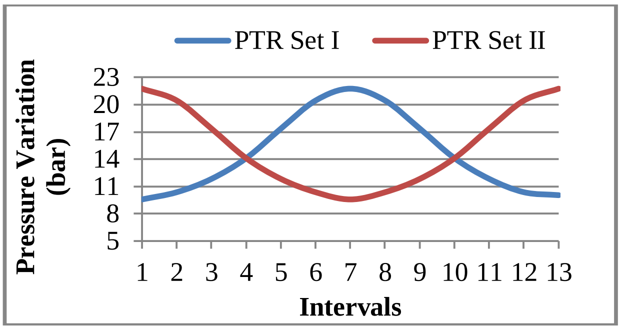

Fig. 3. Pressure variation in compression space

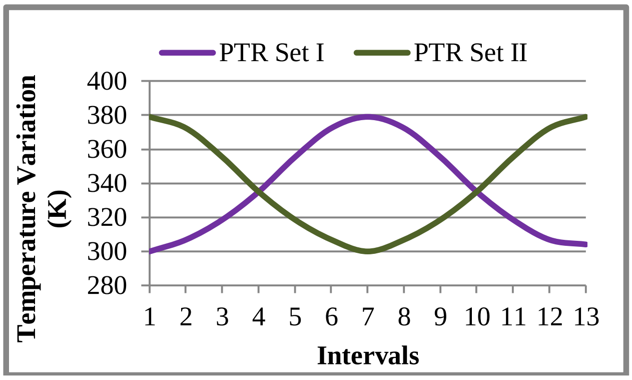

Fig. 4. Temperature variation in compression space

4. Conclusion

Theoretical model is developed to simulate the performance of the counter flow inertance Pulse Tube Refrigerator. A new concept of phase shifter is introduced with elimination of reservoir. Theoretical model is applied to predict the performance of the refrigerator. The required computer code is under development.

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