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Research Article

Performance Evaluation of 0.5 W at 80 K Stirling Cryocooler

Mandar M. Lele[†*]

[†]Department of Mechanical Engineering, MAEER's MITCOE, Pune -411038, Maharashtra, India

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Abstract

The analysis of Stirling Cryocooler is very important while designing a particular machine for specific application. The actual and ideal Stirling cycle differs substantially and the performance prediction of the actual cycle is also an important task for the scientists. The analysis of the Stirling Cryocooler is a complicated one as it involves both dynamic and thermodynamic aspects. The cyclic analysis is used for designing Stirling Cryocooler for desired capacity by considering dead volumes in the system, property changes of the matrix and working fluid, pressure and temperature changes in the system, speed and different losses like shuttle heat conduction, temperature swing, pumping loss, instantaneous pressure drop, conduction through various solid parts and the regenerator loss. The loss analysis also considers the flow rates at different intervals. Thus the cyclic analysis is more realistic in approach. The cyclic analysis of the Stirling cycle Cryocooler for performance evaluation of a 0.5 W at 80 K has been attempted.

Keywords: *Stirling cycle, Cryocooler, performance evaluation, cyclic analysis, regenerator, cooling capacity*

1. Introduction

The cyclic analysis is a methodology, which determines the design parameters of Cryocoolers by considering dead volume of the system, property changes of the matrix and working fluid, pressure and temperature changes in the system, speed and different losses occurring in the system including the regenerator loss. Also loss analysis, considers the flow rates at different intervals for the existing pressure and temperature. Further cyclic analysis helps to finalize different design parameters to suit design objective. The cyclic analysis is thus more realistic in approach.

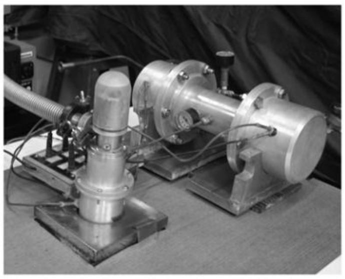

Fig. 1 Schematic diagram of split Stirling Cryocooler

2. Cyclic analysis of Stirling cycle Cryocooler

2.1 Analysis of Stirling cryocooler

The analysis of Stirling cryocooler is very important while designing a particular machine for specific application. The actual and ideal Stirling cycle differ substantially and the performance prediction of the actual cycle is also an important task for the scientists. In this paper, the cyclic analysis of the actual Stirling cycle cryocooler for 0.5 W at 80 K has been attempted.

*Corresponding author: **Mandar M. Lele**

2.2 Cyclic Analysis

The present analysis is based on the same approach as that of Lele [2005]. The Variable dead space as well as the dead space present in the regenerator and other components is considered for the analysis. The pressure drop is considered in regenerator and connecting tubing. This will affect the performance of the cooler in terms of power input. This ultimately depends on the choice of the matrix material, the mesh size of the matrix, diameter of connecting tube and length of connecting tube.

The important assumptions are listed below

1. The gas behaves as a perfect gas
2. Movements of the piston and displacer are sinusoidal
3. The pressure in all components of the system remains same at any instant
4. The compression process is adiabatic and expansion process is isothermal. The regenerator dead space temperature is logarithmic mean temperature of the two end temperatures of the regenerator
5. The regenerator mass flow rate is the average of the mass flow rates of the compression and expansion spaces

6. The temperature in the compression space is equal to the condenser temperature of the conventional system used for same application.

2.3 Pressure-Volume Variations

The volume variations in expansion and compression spaces vary sinusoidally because of piston and displacer movements. The volume variations for the expansion and compression spaces are given by the following expressions

$$V_E = V_{EC} + V_{EM} \times \frac{(1 - \cos\phi)}{2} \quad (1)$$

$$V_C = V_{CC} + V_{EM} \times \frac{1 + K + \cos\phi - K \times \cos(\phi - \alpha)}{2} \quad (2)$$

V_{EC} and V_{CC} is considered to be approximately 2% of V_{EM} and V_{CM} respectively.

The total mass of the working fluid is

$$M = M_C + M_{RD} + M_E \quad (3)$$

Using perfect gas laws,

$$P = \frac{M \times R}{\{(V_C / T_C) + (V_{RD} / T_{RD}) + (V_E / T_E)\}} \quad (4)$$

Assuming (M x R) to be unity, as exact value of M is not known. The complete cycle is considered to be split into 12 equal intervals of 30° crank angle. The pressure for the first interval is then obtained as,

$$P(1) = \frac{1}{\{V_C(1) / T_C(1) + V_{RD}(1) / T_{RD}(1) + V_E(1) / T_E(1)\}} \quad (5)$$

For adiabatic compression, assuming $(\gamma-1)/\gamma = E$

$$T_C(2) = T_C(1) \times \{P(2) / P(1)\} E \quad (6)$$

The value of P (2) could be obtained by using the Newton-Raphson method. The value of the P (2) obtained is then used further to calculate P (3) and so on. In the same way all the values of pressure and temperature are calculated for complete cycle, depending on the interval chosen

The Mean pressure, P is given by,

$$P_m = P_{TOTAL} / 12 \quad (7)$$

where,

$$P_{TOTAL} = \sum_{I=1}^{I=12} P(I)$$

The average existing pressure of the system, P_{avg} is known and the ratio of P_{avg} to P_m would give the correct value of (M x R). Firstly, (M x R) is assumed to be equal to unity.

So, with the new values of (M x R), all the pressure values calculated earlier are required to be corrected by multiplying P (1), P (2) ----- P (13) by a factor equal to (P_{avg} / P_m) so that Pm matches P_{avg}.

2.4 Calculation of Mass Flow Rates

The mass fraction into working spaces viz., compression space and expansion space is different because of the regenerator dead space and is calculated for each interval as follows-

$$F_E(I) = P(I)\, V_E(I) / (M\, R\, T_E) \quad (8)$$

$$F_C(I) = P(I)\, V_C(I) / (M\, R\, T_C(I)) \quad (9)$$

The mass flow rates are given as,

$$W_{ES}(I) = [F_E(I+1) - F_E(I)] \times M \times M_W \times N / (\phi / 360) \quad (10)$$

$$W_{CS}(I) = [F_C(I+1) - F_C(I)] \times M \times M_W \times N / (\phi / 360) \quad (11)$$

2.5 Calculation of Power Input and Refrigeration Effect

The ideal power input to the system, POW, and the refrigeration effect ideally available, RE, can be calculated as

$$POW(I) = \int P(I) \times dV\, T(I) \quad (12)$$

$$RE(I) = \int P(I) \times dV\, E(I) \quad (13)$$

$$V_T(I) = V_C(I) + V_D + V_E(I) \quad (14)$$

The integration can be done using the trapezoidal rule. The pressure taken for each interval is the average pressure of the interval. The algebraic sum of the product of pressure and volume difference for each interval gives the total POW and RE.

2.6 Loss Analysis

The loss analysis of the cooler is important and demonstrates its influence on the efficiency and performance of the cooler. There are various losses in the system causing the net power supply to increase and the net refrigeration available to decrease. The power requirement will increase due to pressure drop because of the flow friction in the regenerator and Mechanical loss, whereas refrigeration capacity decreases due to Regenerator ineffectiveness, Shuttle heat conduction, Temperature swing, Pumping loss, Instantaneous pressure drop and Conduction through various solid parts.

2.7 Loss due to Regenerator Ineffectiveness

The loss due to the regenerator ineffectiveness is one of the major losses. Due to regenerator ineffectiveness gas gets cooled from $T_C(I)$ upto ($T_E+\Delta T$) instead of T_E. A part of refrigerating effect is therefore lost.

An equation reported by Martini [1978] is

$$Q_R(I) = F_R \times W_{RS}(I) \cdot C_P \cdot (T_C(I) - T_E) - \left\{ \frac{V_{RD} \cdot C_P \cdot (P_{max} - P_{min})}{RM\ (F_R / N)} \right\} \times (2/NTU + 2) \quad (15)$$

2.8 Loss due to Temperature Swing

This loss accounts for the temperature changes in the matrix of the regenerator during the cycle. The expression for loss due to temperature swing, as given by Martini [1978] is,

$$D_{RMT} = \frac{W_{RS} \times C_V \times (T_C - T_E)}{N \times M_m \times C_m} \quad (16)$$

The temperature swing loss, therefore, is equal to,

$$Q_{TS} = (\text{Cycle Time}) \times W_{RS} \times C_V \times D_{RMT} / 2 \quad (17)$$

2.9 Pumping Loss

The pumping loss is due to fixed clearance volume that must exist between the displacer and the cryo-cylinder so that the displacer can move without rubbing. Martini [1978] has reported following relation for this loss.

$$Q_{PU} = \left(\frac{\pi DE}{k_g}\right)^{0.6} \times \frac{2 \times L_E \times (T_C - T_E)}{1.5} \times \left[\frac{(P_{max} - P_{min}) \times N \times C_P \times 2}{(T_C + T_E) \times R \times M}\right]^{1.6} \times C_C^{2.6} \quad (18)$$

2.10 Loss due to Shuttle Conduction

Shuttle heat transfer is a form of convective heat transfer, which occurs when a piston with an axial temperature gradient reciprocated inside the cylinder with a similar axial temperature gradient

The loss takes place when the displacer oscillates across a temperature gradient. The heat transferred per half cycle is given as,

$$Q = k_g A_S (T_C - T_E) t / CC \quad (19)$$

where,

A_S = Surface area of the displacer = $\pi \times D_E \times S_E$

Typical equation used for the present analysis is as given below

$$Q_{SH} = \frac{K_g \times \pi \times D_E \times S_E^2 \times (T_C - T_E)}{5.4 \times C_C \times L_E} \quad (20)$$

Here the factor "5.4" in the denominator is used for sinusoidal motion as mentioned by Zimmerman and Longsworth [1970].

2.10 Loss due to Pressure Drop

The present analysis considers the co-relation given by Chan and Griffin [1983], as it fits Kays and London data within ± 10 %. The pressure drop in connecting tube is also calculated by using the same co-relation. The additional power required because of the regenerator and connecting tube pressure drop is to be added to ideal power, which gives basic power requirement for the cryocooler. The empirical equation derived by Chan and Griffin [1983] is

$$f = 49.78 / Re_l + 0.318 \quad (3 < Re_l < 2000)$$

2.11 Conduction Loss

This loss is independent of the machine speed. It is simply the heat transferred through the working fluid and different components between the hot and cold ends of the machine. It involves loss due to conduction through, Displacer Material Regenerator Matrix Material, Cryocylinder

The losses are calculated from the basic equation of conduction heat transfer, in a cyclic manner,

$$Q_K = k_m A_k (Tc - Te) / L \quad (21)$$

Effective thermal conductivity of the regenerator matrix, kmx is given by Martini

$$k_{mx} = \frac{\dfrac{(1 + k_m / k_g)}{(1 - k_m / k_g)} - FF}{\dfrac{(1 + k_m / k_g)}{(1 - k_m / k_g)} + FF}$$

where,

FF = Fill factor = (1- Porosity), for the mesh.

The flow chart for the Cyclic Analysis is as shown in Fig. 2.

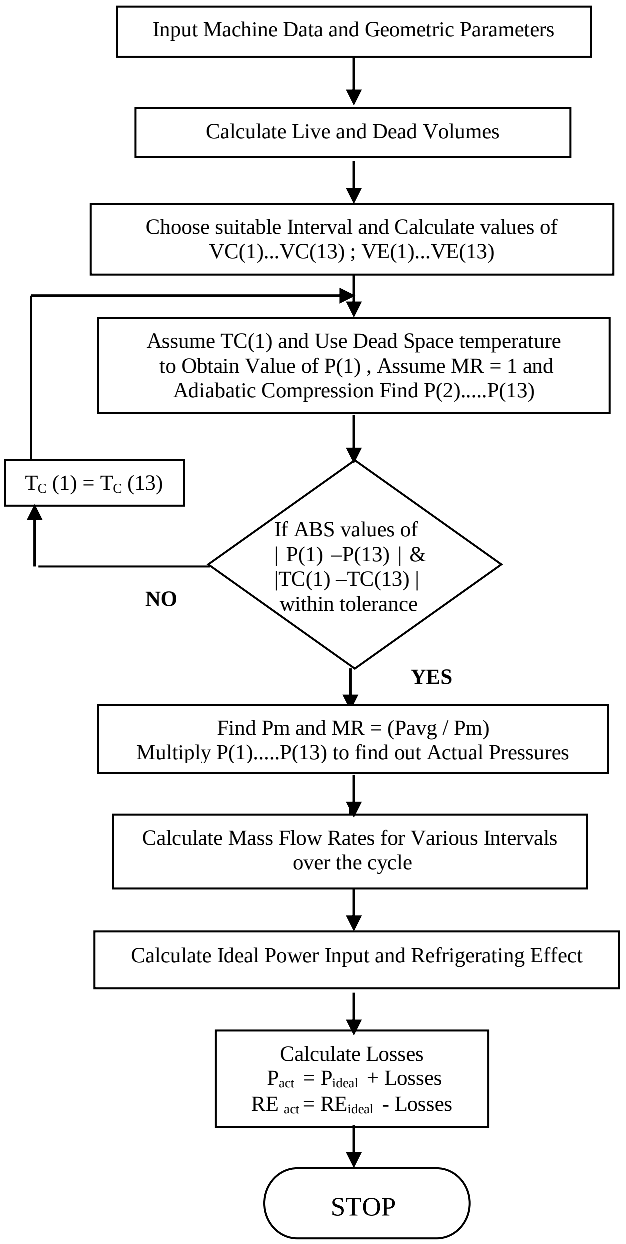

Fig. 2 Flow chart for Cyclic Analysis [Lele *et al.*, 2005]

Lele *et al*. [2005] developed a cyclic analysis for predicting theoretical performance of Stirling cryocooler to meet design objective of 3 W at 80 K application. Based on analysis, the Stirling cryocooler was designed and developed for 3 W at 80 K and it was observed that performance was in good agreement with theoretical predictions. In present study, the same methodology is used to design Stirling cryocooler to meet 0.5 W at 80 K.

Figure 3 shows the comparison of theoretical analysis and experimental investigations. It is observed that they are in good agreement at design condition, which confirms the design objective of 3 W at 80 K.

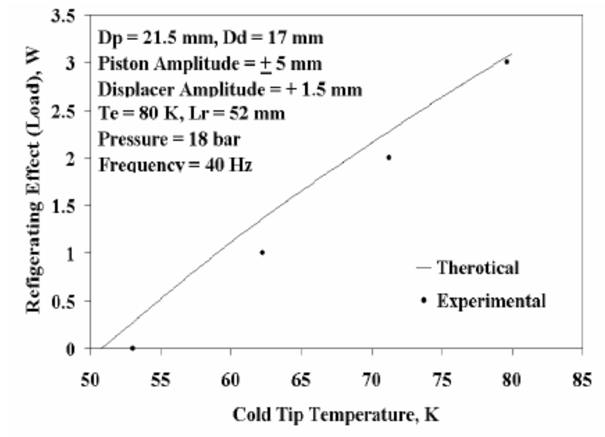

Fig. 3 Comparison of theoretical analysis and experimental investigations [Lele *et al*. 2006]

3. Parametric analysis of Cryocooler (0.5 W at 80 K)

The performance of Cryocooler depends upon various geometric and operating parameters viz. geometric parameter- diameter of piston, diameter of displacer, length of regenerator and the operating parameters - speed and operating pressure.

The choice of combination of all these parameters is critical and would decide the optimum operating condition. The cyclic analysis developed is used to carry out parametric analysis, to investigate the influence of operating and geometric parameters on the performance in terms of RE and COP.

Figure 4 shows the effect of displacer diameter on the performance of the Cryocooler in terms of the net refrigerating effect, actual power and actual COP. It is seen that the refrigerating effect of 0.5 at 80 K is obtained with 12 mm displacer diameter. For the same diameter the actual power is lowest with maximum COP value.

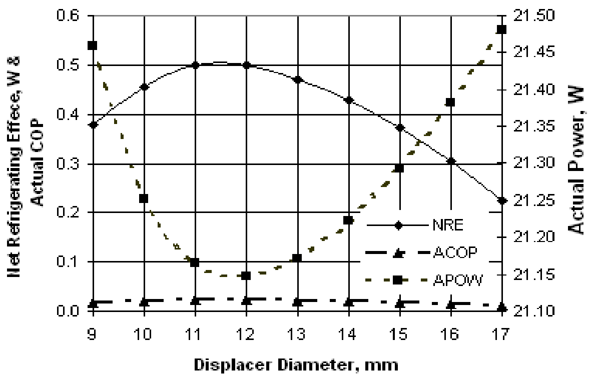

Fig. 4 Variation of Net refrigerating effect, actual power and actual COP with displacer diameter

Figure 5 shows the effect of piston diameter on the performance of the Cryocooler in terms of the net refrigerating effect, actual power and actual COP. It can be observed that the refrigerating capacity is matching with the piston diameter of 15mm with moderate COP at that condition. The actual power shows increasing nature with respect to the piston diameter. The choice of 15 mm piston diameter is good enough to match the requirement in terms of refrigerating effect and power.

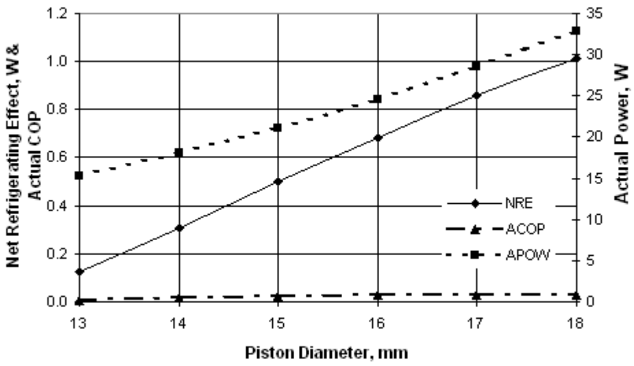

Fig. 5 Variation of Net refrigerating effect, actual power and actual COP with piston diameter

Figure 6 shows the effect of length of regenerator on the performance of the Cryocooler in terms of the net refrigerating effect, actual power and actual COP. It can be observed that the refrigerating capacity is increasing with increase in length of regenerator; with length of regenerator 60 mm the refrigeration capacity of 0.5 W is reached. At the same time actual power decreases with increase in length of regenerator due to decrease in pressure drop. Hence choice of 60 mm regenerator is best as 0.5 W is achieved with lowest power and maximum COP.

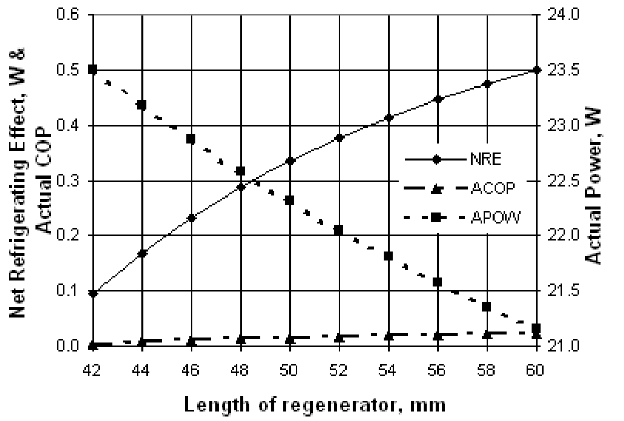

Fig. 6 Variation of Net refrigerating effect, actual power and actual COP with Length of regenerator

Figure 7 shows the effect of pressure on the performance of the Cryocooler in terms of the net refrigerating effect, actual power and actual COP. It can be observed that the refrigerating capacity, actual power and actual COP is increasing with pressure. At pressure of 12 bar required refrigerating capacity is reached with moderate power and actual COP

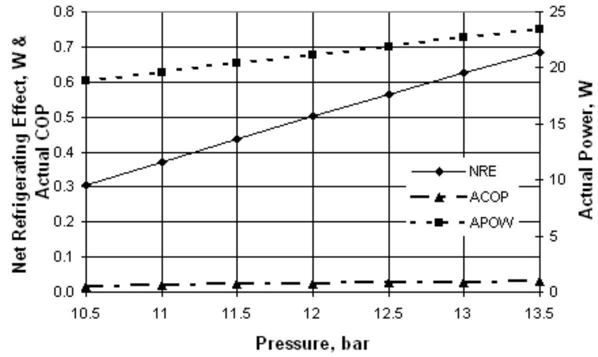

Fig. 7 Variation of Net refrigerating effect, actual power and actual COP with Pressure

Figure 8 shows the effect of Frequency on the performance of the Cryocooler in terms of the net refrigerating effect, actual power and actual COP. It can be observed that the refrigerating capacity, actual power and actual COP is increased with increase in frequency. At frequency of 43 Hz the required refrigerating capacity is reached with moderate power and actual COP

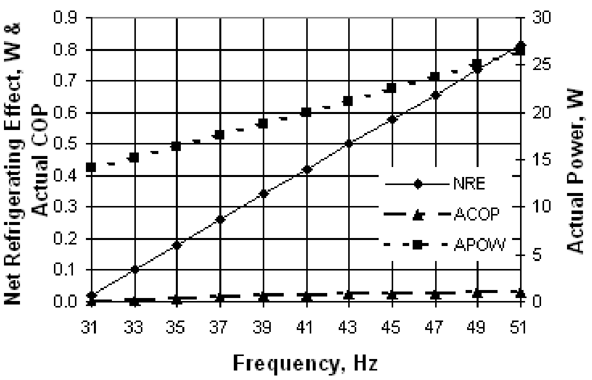

Fig. 8 Variation of Net refrigerating effect, actual power and actual COP with Frequency

4. Conclusions

1) The long life of Stirling Cryocooler is provided by careful selection of low outgassing materials and by non-contacting clearance seals that allow compression of the helium working fluid using a piston that never contacts the cylinder wall and requires no lubrication.
2) The displacer module consists of one or more stages of cooling and is connected to the compressor by a small tube. Stirling cycle cryocoolers provide cooling between 15 K and 150 K, depending on the application.

Nomenclature

A_k	Area of heat conduction	$T_C(1)$	Temperature of gas in compression space during first interval of cycle
C_C	Radial clearance between Displacer and cylinder (m)	$T_C(2)$	Temperature of gas in compression space during second interval of cycle
$F_E(I)$	Gas fraction in expansion space for the I st interval of cycle	V_C	Compression space volumes
$F_C(I)$	Gas fraction in compression space for the I st interval of cycle	V_E	Expansion space volumes
K_m	Thermal conductivity of the material	V_{CC}	Clearance volumes in compression spaces (m^3)
L	Distance between hot and cold ends	V_{EM}	Maximum expansion space volumes (m^3)
L_E	Distance between the hot and cold ends	V_{CM}	Maximum compression space volumes (m^3)
N	Cycle rate	$V_E(1)$	Volume variation in expansion space during first interval of cycle
P	Pressure (bar)	$V_C(1)$	Volume variation in compression space during first interval of cycle
P(1)	Pressure developed in working space during first interval of cycle	W_{ES} (I)	Mass flow rates of gas in the expansion spaces for first interval of the cycle.
P(2)	Pressure developed in working space during second interval of cycle	W_{CS} (I)	Mass flow rates of gas in the compression spaces for first interval of the cycle.
Q	Heat transfer between the displacer and the wall per cycle	ϕ	Crank angle
S_E	Stroke of displacer, m	α	Phase difference between displacer and piston
T_C	Temperature of gas in compression space	γ	Ratio of specific heat capacity
T_E	Temperature of gas in expansion space		

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