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Analysis of Stirling Cycle Cryocooler

Maheshbhai. J. Vanjara[1], Sanketkumar. D. Soni[2]

[1]Mechanical Engineering Department, GEC Modasa
[2]Mechanical Engineering Department, GEC Modasa

Abstract — *Walker reported charts for the Stirling Cycle machine, for cryocooler as well as for engines. The charts were derived from Schmidt's analysis and were based on the ideal working of the cycle. In practice, the operating cycle of the Stirling cycle machine is complex but with certain assumption it could be shown to be the function of six independent parameters, four forms a combination, which gives the design configuration of the machine. This paper aims at finding out this combination for different machines using a realistic way of the analysis of the Stirling cycle machines. Also, a detailed generalized design procedure of the cryocooler is included. The loss analysis also is the important aspect of the cyclic analysis. The different losses that are responsible for the increase in the power input to the system and decrease in the available refrigeration effect are calculated and the net values of power supply, Q_P and refrigeration effect, Q_A, are obtained. The losses in the refrigeration effect apart from the system working parameters are also dependent on the configuration and dimension of the component. This indicates very clearly that the cyclic analysis without consideration of configuration dependent losses is not enough.*

Keywords- *losses; Refrigeration capacity; Regenerator; COP-coefficient of performance; dead volume.*

I. INTRODUCTION

The loss analysis of the cryocooler is very important and demonstrates its influence on the efficiency and the performance of the cryocooler. There are various losses in the system because of which the net power supply increases and the net refrigeration effect decreases. A numbers of expressions are available in the literature to obtain pressure drop in the regenerator. Also there are various losses because of which the available refrigeration effect decreases. The relations which are chosen for the loss calculation are based on the criteria of application simplicity and machine type compatibility. Since the effort required or the benefit desired in any, practical situation can be expressed as a function of certain decision variables, the various design parameters form non-demensionalized number. The basic Analysis process therefore, starts from the identification of these design parameters and formulation of the objective function [1,2].

II. DESIGN PARAMETERS IN STIRLING CYCLE CRYOCOOLER

The major design parameters of the Stirling cycle machines, as named by Walker are:

1. Speed, N
2. Pressure of the working fluid,varies continuously from maximum to minimum, that is, from Pmax to Pmin.
3. The temperature variations in the compression space and the expansion space. The expansion space temperature, TE, is assumed to be constant while the temperature in the compression space varies.
4. The ratio of the swept volumes of the compression and the expansion spaces, K.
5. The phase angle, α, by which the volume variations in the expansion space lead those in the compression space.
6. The dead volume ratio, X. It is the ration of the total dead volume of the system and the swept volume in the expansion space.

III. DESIGN PROCEDURE

Design procedure of the cryocooler. The design procedure involves following major steps.

1. Calculation of QA,liq : The capacity of the cryocooler (W) for liquefaction of nitrogen is calculated by the following expression. QA,liq = Flow rate in kg/s (Enthalpy of gas at 300k -Enthalpy of gas at 77K + Latent heat of condensation at 77K).
2. Approximately initialization of VEM :Walker reported an approximate relation between qa, Pavg, N and the expansion space volume as, QA,liq = TE × 10-4 × VEM × Pavg × (N/60). Deciding the values of Pavg (bar) and TE (k), the approximately value of VEM (cm3) can be found out as an initial value. The diameter and the stroke of the displacer are calculated from this volume.

3. Allocation of dead volume: Assume a lower value of X, say 1 to start with, and calculate the dead volumes in regenerator, cooler and condenser. Take the percentage distribution of the dead volumes as per the assumption number 3. Calculate the dimension of different components subsequently.
4. Cyclic analysis: Initialize K and α. find out the value of OP by using present cyclic analysis. The QA,liq, otherwise the assumed value of VEM should be changed accordingly.
5. Optimum dead volume ratio: Increment the value of X and repeat the above procedure till the maximum value of OP is encountered. This value of X with corresponding K and α would form the optimum design combination.

For example, if the designed piston and displacer diameters are such that the number of slots cannot be accommodated on the water cooler the design needs to be changed and design has to be again verified from the heat exchange considerations. So, if these constraints are known, then they can be incorporated in the beginning or the data can be supplied after the first derivation process ends and the process of iterations have to be started again. The generalized procedure is applied for the design of a cryocooler of 8 l/h liquid nitrogen production capacity (PLN-108s). Also, the effects of dead space distribution on the design parameters are analyzed.

The objective functions those could be used for design parameters, in case of a cryocooler, can be of different types. The selection of these functions would be based on the particular application and objective.

The design parameters using COP as the criteria ensures maximum refrigeration effect per unit energy supply. The energy conservation necessitates the COP consideration for optimization. But COP consideration does not exercise control over the operating pressures and space distribution of the system configuration which are also the essentials of the design optimization. Optimization parameter based on the maximum pressure consideration, ensures a control over the maximum pressure occurring in the system. The concept of energy conservation has to be, therefore, compromised in this case. It does not consider the net power to the system but it considers the net power system but it considers the maximum work done on the compression space only. Also, as walker has pointed out, this parameter is on the same line of that of beale number used in Stirling engine optimization. Daley et al. used the same number for optimization of Stirling engines. The minimum pressure consideration for optimization is based on walker's consideration for the comparison purpose only.

3.1. Assumptions

Following assumption are made for the generalized design procedure:

1. The ratios of stroke and diameter for displacer and piston are assumed. The heights of cooler and condenser slots are assumed to be 62 and 60 mm respectively. The width of the slots is assumed to be 0.4 mm for both and the length of the slots is assumed to be 5.5 and 6 mm for cooler and condenser respectively. The height of regenerator is assumed to be 45 mm.
2. The water flow conditions are known. The water flow rate is kept as 0.75 m^3/h. At across the two ends is assumed to be 10°C, and Tm for water is 10°C.
3. The total dead space is assumed to be distributed in regenerator, cooler and condenser in the proportions of 80, 10 and 10 percent respectively. This distribution could be varied as per the design considerations.
4. The regenerator mesh material is assumed to be phosphor bronze. The regenerator matrix data can be varied if desired.

IV. REFRIGERATION CAPACITY

Reduction in refrigeration capacity is due to the following losses:

1. Regenerator ineffectiveness
2. Shuttle heat conduction
3. Temperature swing
4. Pumping action
5. Instantaneous pressure drop
6. Axial conduction through different parts

4.1. Loss due to Ineffectiveness of the regenerator

The loss due to the regenerator ineffectiveness (QR) is one of the major losses. Because of the ineffectiveness of the regenerator, the gas gets cooled from TC (I) up to (TE+ ΔT) instead of TE. A part of the refrigeration effect, therefore, is lost. To calculate this loss on the cyclic basis, for each flow rate in the regenerator, the effectiveness (∈) is calculated [1,3]. The present regenerator analysis is applied for the loss calculation. The loss $Q_R(I)$ for, the 1st interval, is given as

$$QR(I) = W_{RS}(I)\ CV\ (TC(I) - TE)(1-\in)(Interval\ time) \quad \text{...Eq.(1)}$$

The temperature difference (TC(I)-TE) varies for each mass flow rate, $W_{RS}(I)$ for I^{th} interval and therefore the effectiveness, $\in$. The loss in the refrigeration effect is calculated when the gas is moving up through the regenerator. This is taken care by the sign convertor of the mass flow rate, $W_{RS}(I)$. The cumulative value of QR(I) gives the total loss due to the ineffectiveness of the regenerator. The mesh structure used in present case is 200 mesh number and 47 wire gauge. Table 1 gives the loss for different interval of the cycle for PLN-106 machine.

Table 1. Loss due to regenerator ineffectiveness in each interval for PLN-106.

Interval (I)	$W_{RS}(I)$ (g/s)	QR(I) (W)
1	32.63	89
2	50.50	166
3	37.74	97
4	15.47	15
5	9.69	5

4.2. Loss due to shuttle heat conduction

The displacer absorbs heat at the hot end and gives off at the cold end during its stroke. Shuttle heat conduction depends on the area involved, the thickness of the gap between the displacer and out side wall, C_C, and the temperature gradient across the displacer. The transfer of heat takes place in the half cycle only as this is the time during which displacer comes in the centre of the stroke, goes to the top/bottom end of the stroke and again comes back to the centre[3,4]. During one half cycles, it takes up the heat from the hot end while during other half; it gives off heat to the cold end. The heat transferred per half cycle time, t, could be given as,

$$Q(I) = kg\ A\ (TC(I) - TE)\frac{t}{Cc} \quad \text{....Eq.(2)}$$

Where, A is the surface area of the displacer.
The rate at which heat is transferred from the hot end of the displacer to the cold end depends on,

1. Heat transfer between the displacer and the wall per interval, Q(I).
2. Distance through which average energy is transported during each cycle, Sd.
3. Cycle rate, N.
4. Distance between the hot and the cold ends, Ld.
5. Therefore, the loss is given as,

$$QSH(I) = Q(I)\ Sd\frac{N}{Ld\ (Interval\ time)} \quad \text{....Eq.(3)}$$

Depending on the interval, the half cycle period, during which the displacer passes through the cold end, has to be identified when the loss would take place. The loss calculation is done for each interval. In the present case, the intervals during which this loss would take place have to be marked.

If the displacer starts its movement from the top dead center, and if each interval is of 300, then the intervals in which this loss would take place would be 10,11,12,1,2 and 3 which comprise of the half cycle for this loss. Table 2 gives the calculated loss for each interval for PLN-106.

4.3. Loss due to Temperature Swing

This loss accounts for the temperature changes in the matrix of the regenerator during the cycle. It is the heat taken up by the matrix due to its finite heat capacity. The drop in the regenerator matrix temperature, all along the line, due to a single flow of gas in the expansion space, as given by Martini is,

$$DRMT = W_{RS}CV\frac{(TC - TE)}{N\ MMX\ CM} \quad \text{....Eq.(4)}$$

The temperature swing loss, therefore, is equal to,

$$Q\ TS = W_{RS}\ CV\ \frac{DMRT}{2} \quad \text{....Eq.(5)}$$

The loss is calculated when the gas moves up through the regenerator in the cycle. The loss has to be calculated for the average flow during this period. All other values of temperature are also taken as average values.

Table 2. Loss due to shuttle heat conduction in each interval for PLN-106.

Interval (I)	Q(I) (J/cycle)	QSH(I) (W)
10	0.10	1
11	0.44	1.5
12	0.99	5.8
1	1.66	9.81
2	1.14	6.72
3	0.53	3.15

4.4. Pumping Loss

As the cryocooler is pressurized and depressurized, the gas present in the gap between the displacer and the wall flows into and out of this gap. Therefore, at the cold end, some of the refrigeration effect available, is taken up by this gas, as given by Martiny,

$$QPU = \left(\pi Dd \times \frac{10000}{kg}\right)^{0.6} \left(\frac{2\,Ld\,(TC(I) - TE) \times 100}{1.5}\right) \left(\frac{(Pmax - Pmin)^{2} 10\,N\,Cp\,2\,Mw}{(TC + TE)R}\right)^{1.6} (Cc \times 100)^{2.6}$$

….Eq.(6)

This loss also has to be calculated for the average values in the cycle as it needs the maximum and minimum pressures developed in the cycle.

4.5. P-V Loss due to pressure drop

Due to pressure drop in the cooler, the regenerator, and the condenser, the expansion space pressure would always be less than that of the compression space. Table 3 gives the magnitude of pressure drop in the cooler, the regenerator, and the condenser of PLN-106 for different interval during the cycle. The pressure in the expansion space would therefore be,

$$PE(I) = P(I) - (Total\ pressure\ drop)$$ ….Eq.(7)

In the way, all the values of PE(I) in the cycle, are calculated. This would reduce the refrigeration effect available from the system. The refrigeration effect, therefore, would be,

$$QIP = \int PE(I)\, dVE(I)$$

….Eq.(8)

Again, the integration is carried out for same intervals of the cycle. The term (QI-QIP) would give the loss [5,6].

Table 3. Pressure drop in cooler, regenerator and condenser for PLN-106.

(I)	$\Delta P_{Coo.}(I)$ (bar)	$\Delta P_{Reg.}(I)$ (bar)	$\Delta P_{Cond}(I)$ (bar)	$\Delta P_{Total}(I)$ (bar)
1	0.102	0.210	0.0047	0.318
2	0.120	0.472	0.023	0.616
3	0.043	0.345	0.022	0.411
4	0.0008	0.095	0.009	0.105
5	0.020	0.008	0.0009	0.0302
6	0.051	0.093	0.001	0.145
7	0.067	0.183	0.005	0.256
8	0.066	0.237	0.009	0.312
9	0.051	0.252	0.012	0.317
10	0.028	0.225	0.014	0.268
11	0.003	0.135	0.011	0.150
12	0.021	0.005	0.002	0.029

4.6. Loss due to axial conduction through different parts

The loss due to conduction continues independent of the machine speed. It is the heat transferred through the different numbers between the hot and cold portions of the machine. It involves loss due to conduction through,
Displacer material: The loss is calculated by the following expression.

$$Qk1(I) = \frac{kg\ Ad\ (TC(I) - TE)}{Ld}$$ ….Eq.(9)

Where Qk1(I) is the loss due to conduction through the displacer area Ad for the Ith interval of the cycle.

$$Ad = \frac{\pi}{4(DID)^2}$$ ….Eq.(10)

$$DID = DIR - 2(Cc) - 2\ (WT1)$$ ….Eq.(11)

Where, DID and DIR are the internal diameter of displacer and regenerator, Cc is the gas filed gap around the displacer and WT1 is the wall thickness of the displacer.
Regenerator outer and inner holding rings: The loss is given by the following expression.

$$Qk2(I) = \frac{km\ AR\ (TC(I) - TE)}{Lr}$$ ….Eq.(12)

Where, AR is the cross section area of the ring and Lr is the length of regenerator. Qk2 and Qk3 give the loss due to conduction through inner and outer ring respectively.
Regenerator matrix material: The regenerator of Stirling machine is made up from many layers of fine screen that is lightly sintered together. The degree of sintering would have a big baring on the thermal conductivity of the screen stack since the controlling resistances is the contact between adjacent wires. Goring reported following formula for conduction through a square array of uniformly sized cylinders.

$$kmx = \left(\frac{\left(\frac{(1 + km/kg)}{(1 - km/kg)}\right) - FF}{\left(\frac{(1 + km/kg)}{(1 - km/kg)}\right) + FF}\right)$$ ….Eq.(13)

FF is filling factor (1-ϕs) for the mesh. The loss Qk4, is then calculated by the simple formula considering the cross sectional area of the regenerator matrix.The losses are calculated from the basic equations in a cyclic manner. The loss due to axial conduction through the sheepwool, packed in the displacer, is neglected. Table 4 summaries the loss calculated for different interval of the cycle for PLN-106 machine.

Table .4. Loss due to conduction through different parts in each interval for PLN-106.

(I)	QK1(I) (W)	QK2(I) (W)	QK3(I) (W)	QK4(I) (W)
1	1.26	0.18	0.26	0.36
2	1.2	0.17	0.25	0.34
3	1.11	0.16	0.23	0.3
4	1.02	0.15	0.21	0.27
5	0.95	0.14	0.20	0.24
6	0.91	0.13	0.19	0.23
7	0.90	0.13	0.19	0.22
8	0.91	0.13	0.19	0.23
9	0.96	0.14	0.20	0.24
10	1.04	0.15	0.22	0.27
11	1.14	0.16	0.24	0.31
12	1.23	0.18	0.26	0.35

V. RESULT AND DISCUSSION

From the above analysis the result of various heat loses are calculated and shown in the Table 5 of available refrigeration for PLN-106 machine.

Table .5. Available refrigeration for PLN-106

Basic refrigeration Capacity (W)	**1967.97**
Losses	
Regenerator loss **(W)**	372
Shuttle loss **(W)**	28.13
Pumping loss **(W)**	88.03
Temp. swing loss **(W)**	503
P-V loss due to pressure drop **(W)**	137
Conduction loss **(W)**	20.68
Total Losses (W)	**1148.84**
Net capacity (W)	**819.13**

VI. CONCLUSION

It is to be clear from the result table that the available refrigeration is reduced due to the various losses but the major refrigeration loss is due to the temperature swing loss and regenerator loss.

REFERENCES

[1] Atery M.D. Analysis & Performance investigation of Stirling cycle cryocooler.
[2] Bretheron A.,Cranville W.H.,Harness J.B. Performance of Regenerator at low temperature.
[3] Walker G. design guideline for large stirling cryocoolers, Cryogenics, 23pp-113-114 feb.(1983).
[4] Rios, P.A. An approximate solution to the shuttle heat transfer losses in a reciprocating machine Journ. Of engg. For Power pp-177-182,April(1971).
[5] Rios, P.A. An analytical and experimental investigation on stirling cycle, Ph.D. thesis, M.L.T.(1969).
[6] Rios, P.A. and J.L.Smith Jr. An analytical and Experimental: Evaluation of the pressure drop losses in the stirling cycle.ASME Paper No. 69-New York (1969).

