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Schmidt cycle analysis in the quest of designing stirling cryocooler

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Abstract

Design of Reverse Stirling Cycle based refrigerator can be predicted by Schmidt theory as a useful tool and by experiment it is found that for practical purposes the power and efficiency predicted by this analysis are about 35% of the actual values. Therefore, appropriate provision is to be made for getting the realistic result with the minimum deviation. The present paper first investigates the suitability of application of Schmidt design analysis for standard ZIF-1002 and PLN-106 Single cylinder Cryogenerator model. As the result is found to be optimistic, the same design procedure is applied for the design of a separate Cryogenerator for generating a cooling effect which is sufficient to produce 7 kg per hour liquid nitrogen using an indigenous condenser of 80% effectiveness. The paper describes all the details of the design methodologies and relevant results are found to be satisfactory.

Keywords: reverse stirling cycle, liquefaction, stirling cryocooler; schmidt analysis; regenerator

1. INTRODUCTION

The gas to be liquefied is contacted to the refrigerated wall of the condenser. The gas is liquefied by the refrigerating effect produced in the inner side of the condenser by the repeated compression and expansion of the refrigerant gas such as hydrogen or helium and after several cycle of compression and expansion, a low temperature in the range of 50K to 60K is achieved. The refrigerating effect is imparted to the incoming purified Nitrogen gas which in turn gets liquefied at the outside wall of the condenser at 77K and thus further decrease in temperature leading to solidification of liquid nitrogen is not possible. The nitrogen gas and the refrigeration gas never come in contact with each other as they flows in two different circuits. There is a temperature gradient of around 17 K between the inner side and outside of the condenser. The liquefied gas thus flows down the piping of the liquefier as shown in Fig. 1. The main units of the Cryorefrigerator are compressor, cooler; regenerator, displacer / expander and condenser.

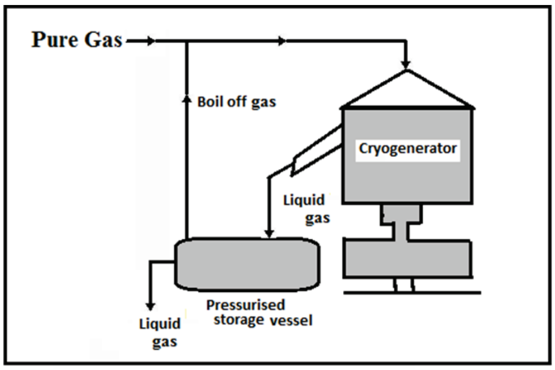

Fig. 1. Cryogenic system for gas liquefaction process.

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These components are hermetically sealed as a single unit. Expansion zone is situated above the expander (displacer) and the compression zone is under the expander and above the piston head.

The squirrel cage motor rotates the crank shaft which is converted into the reciprocating motions of the piston and of the expander. Both the piston and expander are coupled in the same shaft. The expander (displacer) unit is in phase advance of 70-80 degree of the piston.

Several analytical and computerized models for Reverse Stirling Cycle thermodynamic analysis are available in the literature [1-6]. Schmidt analysis is adopted as it is widely used for initial sizing of engine and its produces closed form solution for the engine performance which can be easily manipulated by the designer though its efficiency is temperature dependent.

The present paper describes the methodology of design of an indigenous Cryocooler based on Reverse Stirling Cycle applying Schmidt analysis.

2. CRYOCOOLER DESIGN

The important assumptions on which cryocooler design is based on Schmidt are as follow:

a) The working fluid behaves as ideal gas.
b) Regenerator effectiveness is 100%.
c) Uniform instantaneous pressure throughout the system is assumed.
d) Constant mass of working gas is assumed thus ensuring no leakage of gases.
e) Sinusoidal volume variation in the working space is to be ensured.

f) Temperature gradient in the heat exchanger is zero.
g) Constant temperature of cylinder wall and that of piston are assumed and cylinder content are well mixed.
h) Temperature of the working fluid in the ancillary space is constant.
i) Machine speed is constant and steady state is reached.

Cryocooler performance is quite sensitive to regenerator effectiveness [7]. If the regenerator does not have an effectiveness of 100%, the temperature of the gas leaving the regenerator during cool down will be higher than the desired refrigeration temperature. The net results of this decrease in effectiveness are that less energy is absorbed in the refrigerator because more energy is required to cool the refrigerant to the desired refrigerant temperature. This loss in refrigeration can be expressed in terms of actual energy, Q_{actual}, absorbed in the regenerator during the constant volume cool down process of the refrigerant depicted by step 2-3 of Fig. 2.

$$Q_{actual} = Q_{ideal} - \Delta Q \tag{1}$$

where Q_{ideal} is the heat absorbed in a regenerator of 100% effectiveness and ΔQ represents the energy which is not absorbed in the regenerator due to deviation of its effectiveness from 100%.

The effectiveness of the regenerator is given by

$$E = \frac{Q_{actual}}{Q_{ideal}} = \frac{(Q_{ideal} - \Delta Q)}{Q_{ideal}} \tag{2}$$

Q ideal is given by

$$Q_{ideal} = m_g C_v (T_2 - T_3) \tag{3}$$

In equation (3), m_g and C_V denote mass of the refrigerant (working gas) flowing through the regenerator and specific heat at constant volume respectively.

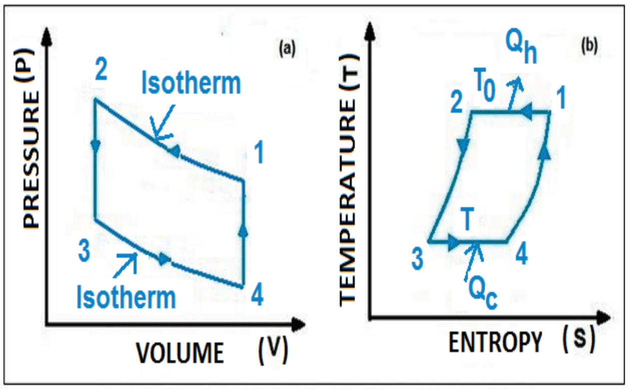

Fig. 2. Stirling Refrigeration Cycle (a) P-V diagram, (b) T-S diagram.

Equation (2) and equation (3) result into formation of equation (4),

$$\Delta Q = (1 - E) m_g C_v (T_2 - T_3) \tag{4}$$

Assuming ideal gas behaviour of the refrigerant gas, the equation becomes,

$$Q_{ideal} = m_g T_3 (S_4 - S_3) \tag{5}$$

S_4 and S_3 are the entropies at point 4 and 3 respectively as shown in Fig. 2.

For an isothermal process, the change in entropy can also be expressed as the ratio of the specific volume as

$$\Delta S = m_g R T_3 \ln(\frac{V_4}{V_3}) \tag{6}$$

For helium $C_v = (\frac{3}{2})R$ and for a well-defined machine

$$(\frac{V_4}{V_3}) = 1.24$$

Therefore, the fraction of the cold lost due to inefficiency of the regenerator is

$$\frac{\Delta Q}{Q_{ideal}} = [\frac{(1-E) C_v (T_2 - T_3)}{R T_3 \ln(\frac{V_4}{V_3})}] \tag{7}$$

Refrigeration effect loss due to 1% decrease in effectiveness for a Cryocooler operating between 60K and 300K with helium as the working fluid, can be given with the help of equation (7) as follows

$$\frac{\Delta Q}{Q_{ideal}} = [\frac{(1-0.99)\frac{3}{2}(300-60)}{60 \ln 1.24}] = 0.279 \tag{8}$$

Therefore, it reveals that there will be a loss of 27.9% in refrigerating effect due to 1% decrease in effectiveness. Following the same calculation, the effectiveness of the regenerator at which the refrigerating effect altogether vanishes can be found out.

Thus

$$\frac{\Delta Q}{Q_{ideal}} = [\frac{(1-E)\frac{3}{2}(300-60)}{60 \ln 1.24}]$$

or, $$1 = [\frac{(1-E)\frac{3}{2}(240)}{60 \ln 1.24}]$$

for which E = 0.964.

Therefore, regenerator of effectiveness 96% would result in zero refrigerating effect which clearly explains the crucial role played by regenerator in the development of Reverse Stirling cycle based cryocooler.

For the present analysis, the regenerator having effectiveness of 100% is used. However, it is established that the prediction for the overall design parameters from Schmidt analysis is 35% efficient considering even perfect regeneration and isothermal compression and expansion [8].

2.1. Effect of principal design parameters on cold production

By Schmidt Cycle analysis,

$$Q_E = P_{\max} V_T \left(\frac{1}{k+1}\right)\left(\frac{A-B}{A+B}\right)^{1/2} \frac{K\sin\alpha}{A+(A^2-B^2)^{1/2}} \tag{9}$$

$$A = T + K + 4XT/(T+1) \tag{10}$$

$$B = (T^2 + 2TKCos\alpha + K^2)^{1/2} \tag{11}$$

$$P_{\max} = P_{\min}(A+B)/(A-B) \tag{12}$$

$$V_T = V_C + V_E = (1+K)V_E \tag{13}$$

$$W = (T-1)Q_E \tag{14}$$

From the above equations, it is noted that Q_E (cold produced in the expansion space) depends on the following independent design parameters i.e. Temperature ratio of the compression space to that of the expansion space (T), swept volume ratio (K), Dead volume ratio of the total internal volume of heat exchanger, ducts, ports to the expansion volume (X), phase angle (α), maximum and minimum pressure of the refrigerant ($P_{\max}$ and $P_{\min}$), speed of the engine (N) and expansion space volume (V_E).

The effect of the four principal design parameters T, K, X and α to produce Q_E is not known initially but these important design parameters must be determined optimally in advance with optimum charts [9, 10]. Chart as in Fig. 3(a-c) along with other design guidelines [11, 12] are used in the subsequent design calculation for Stirling refrigerator.

3. SCHMIDT THEORY IN THE DESIGN OF CRYOGENERATOR UNITS

Based on the Schmidt analysis, computations are done on different standard units such as ZIF-1002 and PLN-106.

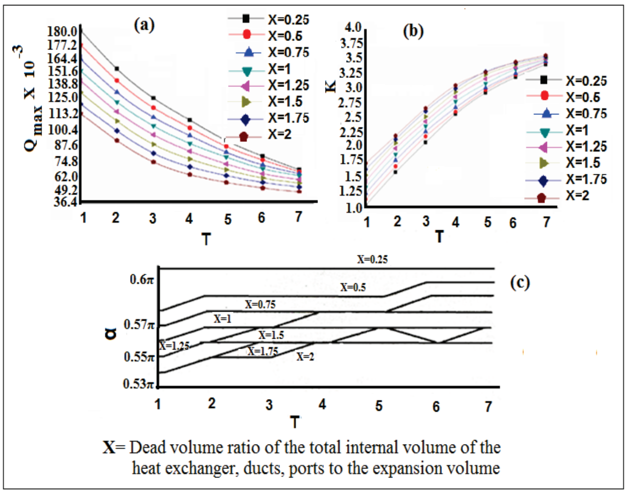

Fig. 3. Design calculation based on optimum design charts: (a) Qmax x 10^{-3} vs. T (b) (Swept ratio) K vs. (Temperature ratio) T (c) (Phase angle) α vs. (Temperature ratio) T.

TABLE1
COMPARISON OF DIFFERENT PARAMETERS GENERATED FROM SCHMIDT ANALYSIS FOR ZIF-1002 PLN-106 CRYOGENERATOR STANDARD MODELS.

	T_E	45K	50K	55K	60K	65K
ZIF 1002	T	6.7	6	5.4	5	4.6
	Q_{max}	$72.5*10^{-3}$	$72.8*10^{-3}$	$75*10^{-3}$	$74.8*10^{-3}$	$75.4*10^{-3}$
	K	3.6	3.5	3.2	3	2.85
	α	108°	107.2°	106.2°	105.8°	104.4°
	A	12	11.21	10.3	9	9.09
	B	6.55	5.98	5.45	5.77	4.77
	P_{max}	$3.45*10^5$	$3.33*10^5$	$3.29*10^5$	$4.63*10^5$	$3.25*10^5$
	V_E	$134.22*10^{-6}$	$136.36*10^{-6}$	$142.099*10^{-6}$	$149.6*10^{-6}$	$155.32*10^{-6}$
	V_T	$617.41*10^{-6}$	$614.83*10^{-6}$	$596.81*10^{-6}$	$598.4*10^{-6}$	$598.41*10^{-6}$
	V_C	$483.19*10^{-6}$	$478.20*10^{-6}$	$454.71*10^{-6}$	$448.8*10^{-6}$	$442.97*10^{-6}$
	W	18.99kW	16.66kW	14.66kW	13.33kW	11.99kW
PLN 106	T	6.7	6	5.4	5	4.6
	Q_{max}	$72.5*10^{-3}$	$72.8*10^{-3}$	$75*10^{-3}$	$74.8*10^{-3}$	$75.4*10^{-3}$
	K	3.6	3.5	3.2	3	2.85
	α	108°	107.2°	106.2°	105.8°	104.4°
	A	12	11.21	10.3	9	9.09
	B	6.55	5.98	5.45	5.77	4.77
	P_{max}	$3.45*10^5$	$3.33*10^5$	$3.29*10^5$	$4.63*10^5$	$3.25*10^5$
	V_E	$121.49*10^{-6}$	$91.186*10^{-6}$	$94.44*10^{-6}$	$397.72*10^{-6}$	$394.55*10^{-6}$
	V_T	$558.84*10^{-6}$	$410.337*10^{-6}$	$396.66*10^{-6}$	$397.72*10^{-6}$	$394.55*10^{-6}$
	V_C	$437.35*10^{-6}$	$319.151*10^{-6}$	$302.22*10^{-6}$	$298.29*10^{-6}$	$292.07*10^{-6}$
	W	12.54kW	10.99kW	9.68kW	8.8kW	7.91kW

The results are then verified with existing dimensions available in the manual of those machines [13-15].

3.1. Refrigerating effect evaluation of Cryogenerator units

3.1.1. Cooling capacity of Model ZIF-1002:

Refrigerating capacity required per cycle at 77K for ZIF-1002 Cryogenerator model is 47.94 J, for operating speed of 1460 rpm and refrigerating ability of 4200 kJ/hr [14]. Now, Q_E = heat lifted in the expansion space (Refrigeration produced) = 47.94/0.35= 136.97 J/cycle.

3.1.2. Cooling capacity of PLN-106:

Cooling load required per cycle at 77K for PLN-106 Cryogenerator model is 31.86 J, for operating speed of 1450 rpm and refrigerating ability of 2772 kJ/hr [15]. Q_E = heat lifted in the expansion space (Refrigeration produced) = 31.86/0.35= 91.034 J/cycle.

For both the units, T_c = temperature of the working fluid in the compression space = 300K, X= Dead volume ratio= 0.5 are assumed. By varying T_E= temperature of the working fluid in the expansion space, the different parameters generated for ZIF-1002 and PLN-106 are presented in Table 1.

3.2. Design of the expansion zone and compression zone:

Guidelines for the design calculation are available in the literature [16].

3.2.1. Cryogenerator model ZIF-1002:

$$V_E = \frac{\Pi}{4} D_E^2 L_E = \frac{\Pi}{4} (0.07)^2 * 0.03 = 115.45*10^{-6} \text{ m}^3$$

$$V_C = \frac{\Pi}{4} D_C^2 L_C = \frac{\Pi}{4} (0.1016)^2 * 0.052 = 421.58*10^{-6} \text{ m}^3$$

3.2.2. Cryogenerator model PLN-106:

$$V_E = \frac{\Pi}{4} D_E^2 L_E = \frac{\Pi}{4} (0.07)^2 * 0.03 = 115.45*10^{-6} \text{ m}^3$$

$$V_C = \frac{\Pi}{4} D_C^2 L_C = \frac{\Pi}{4} (0.08)^2 * 0.052 = 261.38*10^{-6} \text{ m}^3$$

TABLE 2
COMPARATIVE STUDY OF THEORETICAL AND ACTUAL VALUES OF COMPRESSION AND EXPANSION SPACE.

Cryogenerator model	**ZIF-1002**	**PLN-106**
Actual swept volume of expansion space calculated from actual displacing unit stroke length (m^3)	$115.45*10^{-6}$	$115.45*10^{-6}$
Swept volume of expansion space predicted by Schmidt analysis (m^3)	$149.6*10^{-6}$	$99.43*10^{-6}$
Actual swept volume of compression space calculated from actual compressor cylinder diameter and piston stroke (m^3)	$421.58*10^{-6}$	$261.38*10^{-6}$
Swept volume of compression space predicted by Schmidt analysis (m^3)	$448.8*10^{-6}$	$298.29*10^{-6}$

TABLE 3
COMPARATIVE STUDY OF THEORETICAL AND ACTUAL VALUES OF CRANK ANGLES AND POWER.

Cryogenerator model	**ZIF-1002**	**PLN-106**
Actual crank displacement angle (degree)	70	70
Theoretical crank displacement angle (degree)	104.4	104.4
Actual power of the motor in the installed machines (kW)	17	11
Theoretical power of the motor (kW)	13.3 (For T_E= 60K) 16.67 (For T_E= 50K)	8.8 (For T_E= 60K) 10.5 (For T_E= 50K)

Comparison of theoretical with calculated value of swept volume of expansion space, compression space, crank angle, power requirement with actual value for ZIF-1002 and PLN-106 are reported in Table 2 and Table 3. As the results are very optimistic, the same procedure can be adopted for the design of any new Cryogenerator.

4. INDIGENOUS DEVELOPMENT OF THE CRYOGENERATOR

It is intended to develop a Cryogenerator having a sufficient cooling capacity to liquefy pure nitrogen gas at the rate of 7kg/hr. The nitrogen is to be fed to the cryocooler at 300K. The actual refrigerating effect required is calculated by energy balance. Schmidt theory is then applied to evaluate the design load and for fixing different dimension of the machine.

5. DESIGN OF THE CRYOGENERATOR

Refrigeration load of the indigenous Cryogenerator model for liquefaction of 7 kg/hr. liquid nitrogen is computed to be 3801.43 kJ/hr. (1055.67 watt) using a condenser having effectiveness of 80%. Thus, refrigerating capacity required at 77K is 43.39 J per cycle. The speed of motor is noted to be 1460 rpm.
Refrigeration to be produced in the expansion space (Q_E) = 43.39/0.35 = 123.98 J/cycle. The temperature of the working fluid in the compression space (T_c) and dead volume ratio (X) are as 300 K, 0.5 respectively. Different parameters generated for varying temperature of the working fluid in the expansion space (T_E) are reported in Table 4.

The result shows more or less similar trends as predicted by ZIF-1002 and PLN-106. Compressor cylinder diameter, piston stroke, displacing unit diameter, displacing unit stroke, crank displacement angle parameters are fixed as 102 mm, 52 mm, 70 mm, 30 mm respectively based on the prediction from Schmidt analysis and considering the required factor of safety for the indigenous built Cryogenerator for producing 7 kg/hr. liquid nitrogen.

TABLE 4
SCHMIDT DESIGN CALCULATION OF INDIGENOUSLY BUILT CRYOGENERATOR.

T_E		45K	50K	55K	60K	65K
CRYOGENERATOR	T	6.7	6	5.4	5	4.6
	Q_{max}	72.5 $*10^{-3}$	72.8 $*10^{-3}$	75 $*10^{-3}$	74.8 $*10^{-3}$	75.4 $*10^{-3}$
	K	3.6	3.5	3.2	3	2.85
	α	108°	107.2°	106.2°	105.8°	104.4°
	A	12	11.21	10.3	9	9.09
	B	6.55	5.98	5.45	5.77	4.77
	P_{max}	3.45 $* 10^5$	3.33 $*10^5$	3.29 $*10^5$	4.63 $*10^5$	3.25 $*10^5$
	V_E	121.49 $*10^{-6}$	123.68 $*10^{-6}$	128.62 $*10^{-6}$	135.41 $*10^{-6}$	139.57 $*10^{-6}$
	V_T	558.84 $*10^{-6}$	556.54 $*10^{-6}$	540.21 $*10^{-6}$	541.66 $*10^{-6}$	537.35 $*10^{-6}$
	V_C	437.35 $*10^{-6}$	432.86 $*10^{-6}$	411.59 $*10^{-6}$	406.25 $*10^{-6}$	397.78 $*10^{-6}$
	W	17.20 kW	15.08 kW	13.27 kW	12.06 kW	10.86 kW

TABLE 5
COMPARATIVE STUDY OF THEORETICAL AND ACTUAL VALUES OF COMPRESSION AND EXPANSION SPACE.

Indigenously built Cryogenerator	
Actual swept volume of expansion space calculated from actual displacing unit stroke length (m^3)	$115.45*10^{-6}$
Swept volume of expansion space predicted by Schmidt analysis (m^3)	$133.78*10^{-6}$
Actual swept volume of compression space calculated from actual compressor cylinder diameter and piston stroke (m^3)	$424.90*10^{-6}$
Swept volume of compression space predicted by Schmidt analysis (m^3)	$401.32*10^{-6}$

TABLE 6
COMPARATIVE STUDY OF THEORETICAL AND ACTUAL VALUES OF CRANK ANGLES AND POWER.

Indigenously built Cryogenerator	
Actual crank displacement angle (degree)	70
Theoretical crank displacement angle (degree)	104.4
Actual power of the motor in the installed machines (kW)	17
Theoretical power of the motor (kW)	12.06 (For T_E= 60K) 15.08 (For T_E= 50K)

$$V_E = \frac{\Pi}{4} D_E^2 L_E = \frac{\Pi}{4} (0.07)^2 * 0.03 = 115.45*10^{-6}\ m^3$$

$$V_C = \frac{\Pi}{4} D_C^2 L_C = \frac{\Pi}{4} (0.102)^2 * 0.052 = 424.90*10^{-6}\ m^3$$

Comparison of actual swept volume of expansion space, compression space, crank angle, power used with the theoretical result obtained from Schmidt analysis are reported in Table 5 and Table 6 for the indigenous cryocooler.

6. RESULT AND DISCUSSION

From the Table 5 and Table 6, it can be inferred that design methodology adopted is suitable for Cryogenerator development based on Reverse Stirling Cycle. There are minimum deviations of the actual dimension of the machine from the theoretically predicted dimension by Schmidt analysis. The power requirement depends on the lowest cryogenic temperature to be attained in the expansion space of the refrigerator and it decreases with increase in temperature of the expansion space. The power is also accurately calculated by the analysis. Refrigerating capacity at 77K for liquefaction of nitrogen gas is computed as 43.39J for the indigenous Cryogenerator.

7. CONCLUSION

The result found is very encouraging for application of Schmidt analysis in the design of Cryocooler. The investigating team has successfully attempted an indigenous development of a Reverse Stirling cycle based Cryogenerator which is the main component of the Stirling Cycle based liquefaction system for cryogenic gases. Though the liquefaction of nitrogen gas is presently studied, the same may be extended for liquefaction of methane or other cryogenic gases excepting hydrogen and helium. The same cryocooler may also be used for studying superconductivity of materials and cooling superconducting cables.

NOMENCLATURE

$Q_C = Q_E$ = Heat lifted in the expansion space or refrigeration produced per cycle.

A and *B* are two factors.

α = Angle by which volume variation in the expansion space lead to those in the compression space.

P_{max} = Maximum cycle pressure

P_{min} = Minimum cycle pressure

V_T = Combined swept volume

V_C = Swept volume of the compression space

V_E = Swept volume of the expansion space

V_D = Total internal Dead volume

$X = \frac{V_D}{V_E}$ = Dead volume ratio

$K = \frac{V_C}{V_E}$ = Swept volume ratio

$T = \frac{T_C}{T_E}$ = Temperature ratio

T_C = Temperature of the compression space

T_E = Temperature of the expansion space

W = Power requirement of the motor

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