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Generation of Heat Transfer Coefficient Data in Regenerator for Stirling Cycle Refrigeration System

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Abstract*: This paper attempts to generate the data of heat transfer coefficient for regenerator in stirling cycle refrigeration system on the basis of available experimental data. The available data is based on assumption that the mode of heat transfer in regenerator is conduction. This data does not produce optimal design of regenerator. Heat transfer coefficient must be modified to account for heat transfer in a regenerator by all three modes i.e conduction, convection & radiation. The data for heat transfer coefficient is generated on the basis of experimental results available in literature of performance of various stirling cycle refrigeration systems with different designs of regenerator. These models can be used to predict the performance of stirling cycle refrigerator system on the basis of dimensions of regenerator. The models are validated and optimized.

This paper also presents the effect of variations in regenerator dimensions i.e. regenerator length, regenerator diameter, wiremeshsize, wire mesh arrangement, and wire mesh materialonHeat Transfer coefficient of regenerator.

This data for heat transfer coefficient for regenerator of Stirling cycle refrigeration system can be used for optimizing design of regenerator of stirling cycle refrigeration system or predicting performance of stirling cycle system accurately. ,

Keywords: Regenerator, Performance, Regenerator length, Regenerator diameter; optimal value; Regenerator; Heat Transfer Coefficient.

I. Introduction

Presently vapour compression refrigeration systems are the most commonly used among all refrigeration systems. Cloro-Fluoro-Carbon (CFC) is used as a refrigerant in this system, this refrigerant CFC are most destructive to environment as they caused depletion of stratospheric ozone layer and contribute to the green house global warming. Our country is a party to Montreal protocol and Kyoto protocol. as per Montreal protocol The CFC group of refrigerant which cause ozone layer depletion should be banned by year 2010 and HCFC refrigerant do not cause ozone layer depletion but lead to global warming.kyotoprotocol is signed in 1997 and banned used of HCFC as a refrigerant to prevent global warming by year 2030.An alternate refrigeration system is working on non CFC, non HCFC and environment friendly refrigerant should be used.

1.1 DRAWBACKS OF PRESENT VAPOUR COMPRESSION REFRIGERATION SYSTEM

1. The heat transfer through refrigerants in VCRS is in the form of latent heat; so the refrigerants used in this system should have the property to change their phase at the desired pressure and temperature conditions. The refrigerant having the above properties is Cloro-Fluoro-Carbon (CFC). The CFC refrigerants have very high ozone depletion potential and also cause global warming.
2. High initial cost.
3. CFC refrigerant are hazardous for environment.

1.2 NECESSITY OF NEW REFRIGERATION SYSTEM

Chlorofluorocarbons (CFCs) have been used extensively in last five or six decades as refrigerants in the vapor compression cycle to produce refrigerating and air-conditioning effects. In recent years it has been found that CFCs are most destructive to the environment. It has been proved that CFCs are a major cause of depletion of the earth's stratospheric ozone layer and contribute to the greenhouse effect (global warming).

Presently large quantities of CFCs are being used as refrigerants in a number of refrigerating and air-conditioning systems. Though the refrigerant moves in a closed cycle, there are lots of leakages that escape to the atmosphere and cause destruction of the ozone layer. The most shocking fact about CFCs is that they have exceptionally long atmospheric life which, in certain cases, even extends to 100 years. This means that if CFC refrigerants are leaked today in the atmosphere, they will keep depleting ozone layer for the next 100 years.

When the CFC refrigerants are leaked from refrigeration or air-conditioning systems, they drift around the lower layers of the atmosphere. Slowly they start infiltrating into the upper layers

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of the atmosphere and soon reach the ozone rich stratosphere, where they undergo major chemical changes.

In the ozone layer, sunlight enters in its pristine pure form; it is called ultraviolet radiation, which is highly intense and dangerous to plant and human life. The ozone layer filters this highly intense sunlight and allows less intense sunlight, which is not harmful to human and plant life, to the surface of the earth.

The unfiltered sunlight bombards the molecules of CFC refrigerant and they are pushed towards the stratospheric clouds over the poles. Due to this the CFC molecules get disintegrated. The chlorine atom removed from CFCs reacts with ozone molecule (O3) and converts it into oxygen molecule (O2).

Now, the oxygen does not have the capacity to filter the highly intense ultraviolet radiations. So what is happening because of the CFC refrigerants is that the protective ozone layer is getting converted into incapable oxygen. Due to this the amounts of ultraviolet rays reaching the surface of the earth becomes very high and then causes excessive heating in the environment, called the greenhouse effect.

II. SET UP OF α CONFIGURATION STIRLING CYCLE REFRIGERATION SYSTEM

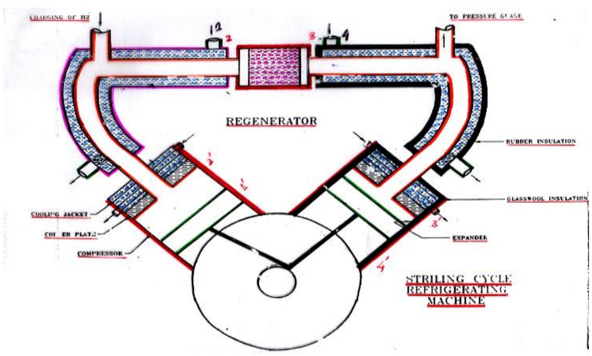


Fig.No.1

In starting the system is charged at predetermined pressure by compressor. The output of this compressor is connected to the air supply of the system. In cylinder 1 air is compressed and the compressed air is passes through the tube to regenerator consisting of a copper tube with copper mesh compacted inside the regenerator, due to the compression of the air the temp increases. The temperature at the entry of the regenerator is designated as T_2 and the temp of outlet is designated as T_3. As the hot air passes through the regenerator, it transfers the heat in the copper mesh and cools. The temp at the exit point is Designated as T_4 which is less than the temp at the inlet i.e. T_2, it passes through the tube goes to the expander. The tube connecting the regenerator and the expander is covered by water jacket through its water circulated where it cools the air further. This heat exchanger reduce the temp of air .The cooled refrigerant is displaced form expander through cooling water Jacket and double pipe heat exchanger where refrigerant absorbs heat from the water flowing through the water Jacket and double pipe heat exchanger to regenerator where it absorbs heat from wire mesh and heated refrigerant passes through the double pipe heat exchanger and water Jacket to compressor, which completes the cycle. The cycle goes on repeating and the temp T_3 and T_2 get stabilize after some time, after that stabilization the temperature of the refrigerating machine remains constant.

2.2 STEPS FOR CALCULATION HTC AND COP

Step 1: Find out the Mass Flow Rate.

Bore Diameter (D) of Cylinder = 70 mm

Total Length of Cylinder = 85mm

$M' = Q \times \rho_{air}$

Step 2 : Find out the Reynolds Number.

$R_e = \frac{4\,m'}{\pi d \mu}$

If $R_e <$ **2300** flow is Laminar

If $R_e >$ **2300** flow is Turbulent

Step 3: Find out the Nusselt Number.

$N_u = 0.023\,(R_e)^{0.3} \times (P_r)^{n}$

n= 0.4 when fluid is being heated

n= 0.3 when fluid is being cold

Step 4 : Calculate the Heat Transfer Coefficient of air passing through Regenerator of 35 mm Diameter filled with copper mesh wire.

$N_u = \frac{h\,D}{k}$

$h = k\, N_u / D$

Step 5 : Calculate the Heat Transfer.

$Q_L = m_w Cp_w \Delta T_w$

Where Q_L is heat lifted in J/sec

m_w is the mass of water in the insulated reservoir.

ΔT_w is the drop in temperature of reservoir water during experimentation

$Q_L = m_w\, Cp_{w\,(T3-T4)}$

Q

$= h\,A\,\Delta T$

Step 6 : Calculate Work Done

$W = P_1 V_1 \ln Pmax / Pmin$

Step 6: Calculate COP.

$COP = \frac{Q}{W} = 4.476$

III. PLANNING OF EXPERIMENTATION

Classical plan of experimentation was used for conducting experimentation on stirling cycle refrigeration system. the design parameters of regenerator are varied over largest possible range. To reduce the variables in experimentation dimension analysis was carried out. The total experimentation was carried out at 25 test points. The table 1 shows the experimental reading at 25 setting of the machines.

The experimental results obtained are shown in Table No.1

Table No.1 Result table for change in $\pi_1, \pi_2, \pi_3, \pi_4$ & π_5

Sr.No	π_1/$L_{R(mm)}$	π_2/$D_{R(mm)}$	π_3/$D_{M(mm)}$	π_4/$V_{R(mm3)}$	π_{5}/$P_{(psi)}$	W	T1 °C	T4°C	COP_{TE}	COP_A
1	2.13/105	0.91/45	0.01826/0.3	0.216/139060	0.156/50	270	49	10	7.256	0.9284
2	2.434/120	0.91/45	0.01826/0.3	0.216/139060	0.156/50	285	44	10	8.3235	1.2144
3	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.156/50	290	38	10	9.4330	1.6091
4	3.043/150	0.91/45	0.01826/0.3	0.216/139060	0.156/50	335	38	10	10.481	1.816
5	3.347/165	0.91/45	0.01826/0.3	0.216/139060	0.156/50	355	35	10	11.32	2.0433
6	2.738/135	0.71/35	0.01826/0.3	0.216/139060	0.156/50	310	44	10	7.075	0.8045
7	2.738/135	0.81/40	0.01826/0.3	0.216/139060	0.156/50	320	44	10	8.323	1.2378
8	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.156/50	315	40	10	9.433	1.6091
9	2.738/135	1.01/50	0.01826/0.3	0.216/139060	0.156/50	310	37	10	10.482	1.84
10	2.738/135	1.11/55	0.01826/0.3	0.216/139060	0.156/50	300	34	10	11.79	2.228
11	2.738/135	0.91/45	0.0183/0.1	0.216/139060	0.156/50	305	40	10	9.433	1.6091
12	2.738/135	0.91/45	0.00406/0.2	0.216/139060	0.156/50	310	41	10	9.129	1.34
13	2.738/135	0.91/45	0.00609/0.3	0.216/139060	0.156/50	305	42	10	8.84	1.11
14	2.738/135	0.91/45	0.00811/0.4	0.216/139060	0.156/50	305	43	10	8.57	0.847
15	2.738/135	0.91/45	0.01014/0.5	0.216/139060	0.156/50	300	44	10	8.32	0.585
16	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.156/50	305	40	10	9.433	1.609
17	2.738/135	0.91/45	0.01826/0.3	0.186/119746	0.156/50	310	41	10	9.129	1.399
18	2.738/135	0.91/45	0.01826/0.3	0.17/109316	0.156/50	305	44	10	8.843	1.156
19	2.738/135	0.91/45	0.01826/0.3	0.147/11619	0.156/50	305	44	10	8.5750	0.91
20	2.738/135	0.91/45	0.01826/0.3	0.123/79315	0.156/50	300	44	10	8.32	0.631
21	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.195/40	310	38	10	10.1	1.93
22	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.1733/45	305	39	10	9.758	1.787
23	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.156/50	305	40	10	9.433	1.6091
24	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.142/55	310	43	10	5.575	1.462
25	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.13/60	305	45	10	8.084	1.379

3.2 THEORETICAL EVALUATION OF PERFORMANCE OF ALPHA CONFIGURATION STIRLING CYCLE REFRIGERATION SYSTEM

The performance of alpha configuration stirling cycle refrigeration systems with 25 different regenerators are evaluated by using fundamental of Heat Transfer. The theoretical coefficient of performance(COP_T) for refrigeration systems are shown in table

Sample calculations for evaluating performance of stirling cycle refrigeration system with first regenerator (Sr.no 1,Table no 2 are as given below)

Evaluation of mass flow rate(M):

$Q=\frac{\pi}{4}(D)^2\ L\ \frac{N}{60}$

Substituting

D= Bore Diameter of compressor cylinder =70mm

L= Stroke of compressor = 85mm

N= number of strokes per minute = 750.

Q = 0.004 m^3/sec

Mass Flow Rate of Refrigerant (M)

M = Q x ρ_{air}

Where ρ_{air} = Density of air

Substituting Q= 0.004 m^3/sec and ρ_{air}= 1.225 kg/m^3

Q = 0.005 kg/sec

Evaluation of Heat Transfer Coefficient of air passing through Regenerator of 35 mm Diameter filled with copper mesh wire.

$N_u = \frac{h\,D}{k}$

h = k N_u/ D

Evaluation of Heat Transfer(Q).

Q = h A ΔT

Evaluation of Work Done(W)

$W=P_1V_1 \ln Pmax/Pmin$

Evaluation of COP(COP_{TH}).

$COP_{TH}=\frac{Q}{W}$=4.4765

3.3 EXPERIMENTAL AND CALCULATED THEORETICAL RESULTS

The table 2 shows the calculated theoretical reading at 25 setting of the machines.

Table No.2 Result table for change in π_1,π_2,π_3,π_4 & π_5

Sr.No.	$\square_{1/LR(mm)}$	$\square_{2/DR(mm)}$	$\square_{3/DM}$	$\square_{4/VR}$	$\square_{5/P}$	T_1	T_4	COP_{TE}	COP_{TH}	% Difference
1	2.13/105	0.91/45	0.01826/0.3	0.216/139060	0.156/50	49	10	7.256	4.47665	38.3041
2	2.434/120	0.91/45	0.01826/0.3	0.216/139060	0.156/50	44	10	8.3235	4.38529	47.3144
3	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.156/50	38	10	9.433	3.94676	58.1600
4	3.043/150	0.91/45	0.01826/0.3	0.216/139060	0.156/50	38	10	10.481	4.38529	58.1616
5	3.347/165	0.91/45	0.01826/0.3	0.216/139060	0.156/50	35	10	11.32	4.22084	62.7134
6	2.738/135	0.71/35	0.01826/0.3	0.216/139060	0.156/50	50	10	7.075	5.92014	16.3230
7	2.738/135	0.81/40	0.01826/0.3	0.216/139060	0.156/50	44	10	8.323	4.93345	40.7250
8	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.156/50	40	10	9.433	4.27566	54.6733
9	2.738/135	1.01/50	0.01826/0.3	0.216/139060	0.156/50	37	10	10.482	3.78231	63.9161
10	2.738/135	1.11/55	0.01826/0.3	0.216/139060	0.156/50	34	10	11.79	3.28896	72.1037
11	2.738/135	0.91/45	0.0183/0.1	0.216/139060	0.156/50	40	10	9.433	4.27566	54.67339
12	2.738/135	0.91/45	0.00406/0.2	0.216/139060	0.156/50	41	10	9.129	4.44011	51.3626
13	2.738/135	0.91/45	0.00609/0.3	0.216/139060	0.156/50	42	10	8.84	4.60456	47.91226
14	2.738/135	0.91/45	0.00811/0.4	0.216/139060	0.156/50	43	10	8.57	4.76901	44.35223
15	2.738/135	0.91/45	0.01014/0.5	0.216/139060	0.156/50	44	10	8.32	4.93345	40.70369
16	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.156/50	40	10	9.433	4.27566	54.67339
17	2.738/135	0.91/45	0.01826/0.3	0.186/119746	0.156/50	41	10	9.129	4.44011	51.3626
18	2.738/135	0.91/45	0.01826/0.3	0.17/109316	0.156/50	42	10	8.843	4.60456	47.91226
19	2.738/135	0.91/45	0.01826/0.3	0.147/11619	0.156/50	44	10	8.575	4.76901	44.35223
20	2.738/135	0.91/45	0.01826/0.3	0.123/79315	0.156/50	44	10	8.32	4.93345	40.70369
21	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.195/40	38	10	10.1	3.94676	60.92315
22	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.1733/45	39	10	9.758	4.11121	57.8683
23	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.156/50	40	10	9.433	4.27566	54.67339
24	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.142/55	43	10	8.575	4.76901	44.38478
25	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.13/60	45	10	8.084	5.0979	36.93838

3.4 THEORETICAL EVALUATION OF HEAT TRANSFER COEFFICIENT ON THE BASIS OF EXPERIMENTAL RESULTS FOR TEMP BASED.

Sample calculations for evaluating performance of stirling cycle refrigeration system with first regenerator (Sr.no 1,Table no 3.are as given below)

Evaluation of Heat Transfer(h).

1) $COP_{TE}=Q/W$, $Q = COP_{TE} *W$

2) Q=hA (Tmax-Tmin) , Q=h (π*D*L)(ΔT)

Where D= Diameter of Regenerator = 45 mm , L= Length of Regenerator = 105 mm

The table 3 shows the calculated heat transfer coefficient on the basis of experimental results for temp based reading at 25 setting of the machines.

Table No.3 Result table for change in π_1,π_2,π_3,π_4 & π_5

Sr.No	$\pi_{1/LR(mm)}$	$\pi_{2/DR(mm)}$	$\pi_{3/DM(mm)}$	$\pi_{4/Vr(mm3)}$	$\pi_{5/P(psi)}$	$Tmax^0K$	$Tmin^OK$	COP_{TE}	HTC_{TE}
1	2.13/105	0.91/45	0.01826/0.3	0.216/139060	0.156/50	322	273	7.256	3464.87
2	2.434/120	0.91/45	0.01826/0.3	0.216/139060	0.156/50	317	273	8.3235	3577.67
3	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.156/50	311	273	9.4330	3773.88
4	3.043/150	0.91/45	0.01826/0.3	0.216/139060	0.156/50	311	273	10.4815	3923.68
5	3.347/165	0.91/45	0.01826/0.3	0.216/139060	0.156/50	308	273	11.32	4029.31
6	2.738/135	0.71/35	0.01826/0.3	0.216/139060	0.156/50	317	273	7.075	3359.73
7	2.738/135	0.81/40	0.01826/0.3	0.216/139060	0.156/50	317	273	8.323	3569.88
8	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.156/50	313	273	9.433	3894.36

9	2.738/135	1.01/50	0.01826/0.3	0.216/139060	0.156/50	310	273	10.482	4143.53
10	2.738/135	1.11/55	0.01826/0.3	0.216/139060	0.156/50	307	273	11.79	4462
11	2.738/135	0.91/45	0.0183/0.1	0.216/139060	0.156/50	313	273	9.433	3770.63
12	2.738/135	0.91/45	0.00406/0.2	0.216/139060	0.156/50	314	273	9.129	3618.47
13	2.738/135	0.91/45	0.00609/0.3	0.216/139060	0.156/50	315	273	8.84	3365.32
14	2.738/135	0.91/45	0.00811/0.4	0.216/139060	0.156/50	316	273	8.57	3186.66
15	2.738/135	0.91/45	0.01014/0.5	0.216/139060	0.156/50	317	273	8.32	2973.83
16	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.156/50	313	273	9.433	3832.44
17	2.738/135	0.91/45	0.01826/0.3	0.186/119746	0.156/50	314	273	9.129	3560.11
18	2.738/135	0.91/45	0.01826/0.3	0.17/109316	0.156/50	317	273	8.84	3213.44
19	2.738/135	0.91/45	0.01826/0.3	0.147/11619	0.156/50	317	273	8.57	3167.14
20	2.738/135	0.91/45	0.01826/0.3	0.123/79315	0.156/50	317	273	8.32	3023.39
21	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.195/40	311	273	10.1	4319.4
22	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.173/45	312	273	9.758	3934.97
23	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.156/50	313	273	9.433	3832.44
24	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.142/55	316	273	5.575	3763.5
25	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.13/60	318	273	8.084	3390.32

3.5 EVALUATION OF HEAT TRANSFER COEFFICIENT ON THE BASIS OF EXPERIMENTAL RESULTS FOR ACTUAL BASED

Sample calculations for evaluating performance of stirling cycle refrigeration system with first regenerator (Sr.no 1, Table no 4 are as given below)

Evaluation of Heat Transfer coefficient (h)

1) $Q = m_w Cp_w \Delta Tw$

2) 2) $Q = hA\,T_S$

The table 4 shows the calculated heat transfer coefficient on the basis of experimental results for actual based reading at 25 setting of the machines.

Table No.4 Result table for change in $\pi_1, \pi_2, \pi_3, \pi_4$ & π_5

Sr.No	$\pi_{1/LR(mm)}$	$\pi_{2/DR(mm)}$	$\pi_{3/DM(mm)}$	$\pi_{4/Vr(mm3)}$	$\pi_{5/P(psi)}$	ΔTW °C	COP_A	HTC_A
1	2.13/105	0.91/45	0.01826/0.3	0.216/139060	0.156/50	3	0.9284	3.895936
2	2.434/120	0.91/45	0.01826/0.3	0.216/139060	0.156/50	4	1.2144	4.328818
3	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.156/50	6	1.6091	4.86992
4	3.043/150	0.91/45	0.01826/0.3	0.216/139060	0.156/50	7	1.816	5.681573
5	3.347/165	0.91/45	0.01826/0.3	0.216/139060	0.156/50	8	2.0433	5.771757
6	2.738/135	0.71/35	0.01826/0.3	0.216/139060	0.156/50	2	0.8045	3.2466
7	2.738/135	0.81/40	0.01826/0.3	0.216/139060	0.156/50	4	1.2378	4.328818
8	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.156/50	6	1.6091	4.86992
9	2.738/135	1.01/50	0.01826/0.3	0.216/139060	0.156/50	7	1.84	5.050287
10	2.738/135	1.11/55	0.01826/0.3	0.216/139060	0.156/50	9	2.228	5.31264
11	2.738/135	0.91/45	0.0183/0.1	0.216/139060	0.156/50	6	1.6091	4.86992
12	2.738/135	0.91/45	0.00406/0.2	0.216/139060	0.156/50	5	1.34	4.638019
13	2.738/135	0.91/45	0.00609/0.3	0.216/139060	0.156/50	4	1.11	4.328818
14	2.738/135	0.91/45	0.00811/0.4	0.216/139060	0.156/50	3	0.847	3.895936
15	2.738/135	0.91/45	0.01014/0.5	0.216/139060	0.156/50	2	0.585	3.246613
16	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.156/50	6	1.609	4.86992
17	2.738/135	0.91/45	0.01826/0.3	0.186/119746	0.156/50	5	1.399	4.638019
18	2.738/135	0.91/45	0.01826/0.3	0.17/109316	0.156/50	3	1.156	3.895936
19	2.738/135	0.91/45	0.01826/0.3	0.147/11619	0.156/50	3	0.91	3.895936
20	2.738/135	0.91/45	0.01826/0.3	0.123/79315	0.156/50	2	0.631	3.246613
21	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.195/40	6	1.93	4.32881

22	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.173/45	6	1.787	4.86992
23	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.156/50	6	1.6091	4.86992
24	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.142/55	6	1.462	4.86992
25	2.738/135	0.91/45	0.01826/0.3	0.216/139060	0.13/60	6	1.379	4.86992

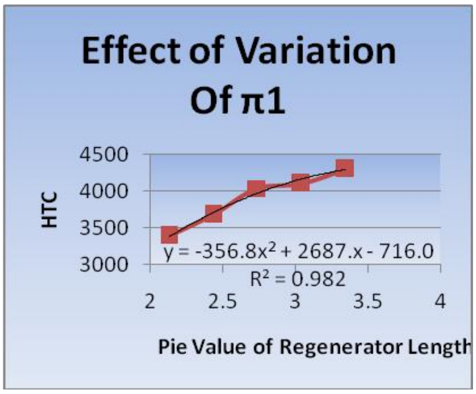
IV. ANALYSIS OF VARIATION OF VARIOUS INDEPENDENT π TERMS ON PERFORMANCE OF REFRIGERATION SYSTEM

a) FORMULATION OF POLYNOMIAL MODEL FOR EXPERIMENTAL. BASED HTC

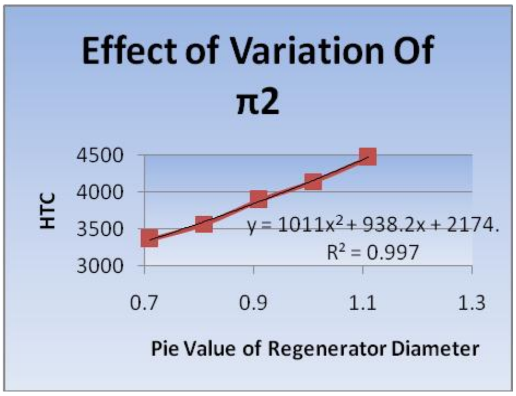
$$HTC_T = K - 46.93\left[\frac{L_R}{(V_E)^{\frac{1}{3}}}\right]^2 + 742\left[\frac{L_R}{(V_E)^{\frac{1}{3}}}\right] + 1011\left[\frac{D_R}{(V_E)^{\frac{1}{3}}}\right]^2 + 938.2\left[\frac{D_R}{(V_E)^{\frac{1}{3}}}\right] - 2E{+}06\left[\frac{D_M}{(V_E)^{\frac{1}{3}}}\right]^2 - 79792\left[\frac{D_M}{(V_E)^{\frac{1}{3}}}\right] + 59357\left[\frac{V_W}{V_R}\right]^2 - 11242\left[\frac{V_W}{V_R}\right] - 26186\left[\frac{Nm_w}{P(V_E)^{\frac{1}{3}}}\right]^2 + 21014\left[\frac{Nm_w}{P(V_E)^{\frac{1}{3}}}\right] \text{---(a)}$$

$$HTC_T = -2380.42 - 46.93\left[\frac{L_R}{(V_E)^{\frac{1}{3}}}\right]^2 + 742\left[\frac{L_R}{(V_E)^{\frac{1}{3}}}\right] + 1011\left[\frac{D_R}{(V_E)^{\frac{1}{3}}}\right]^2 + 938.2\left[\frac{D_R}{(V_E)^{\frac{1}{3}}}\right] - 2E{+}06\left[\frac{D_M}{(V_E)^{\frac{1}{3}}}\right]^2 - 79792\left[\frac{D_M}{(V_E)^{\frac{1}{3}}}\right] + 59357\left[\frac{V_W}{V_R}\right]^2 - 11242\left[\frac{V_W}{V_R}\right] - 26186\left[\frac{Nm_w}{P(V_E)^{\frac{1}{3}}}\right]^2 + 21014\left[\frac{Nm_w}{P(V_E)^{\frac{1}{3}}}\right] \text{--- (b)}$$

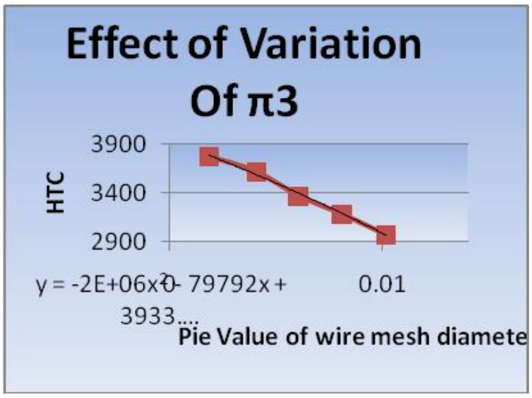
Effect of Variation of π terms On Experimental Based Heat Transfer Coefficient.



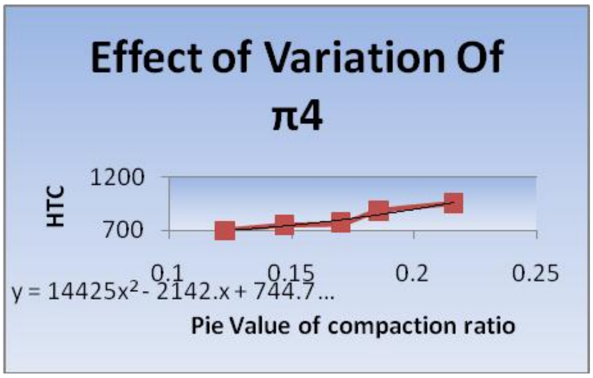
Graph No.1



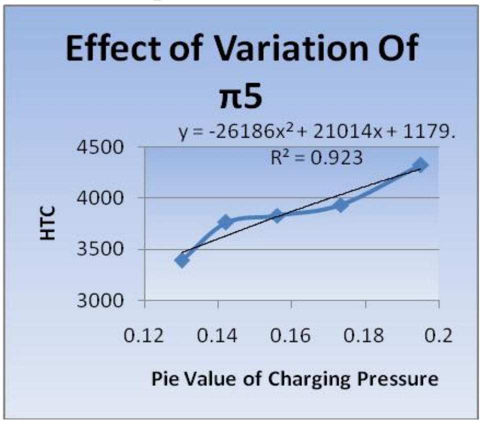
Graph No.2



Graph No.3



Graph No.4



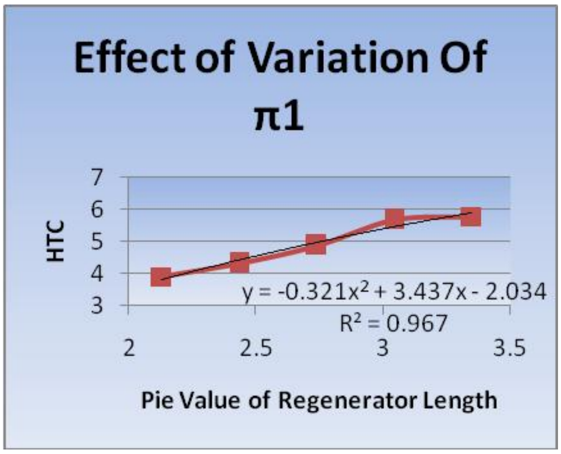
Graph No.5

b) FORMULATION OF POLYNOMIAL MODEL FOR ACTUAL. BASED HTC

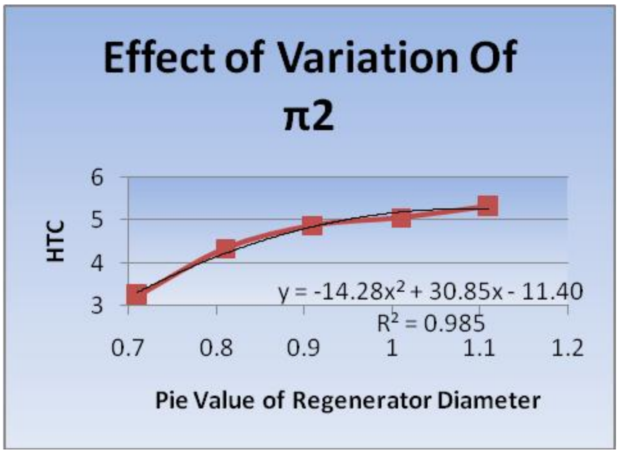
$HTC_A=$
$-14.53-0.321\pi_1^2+3.437\pi_1-14.28\pi_2^2+30.85\pi_2-16986\pi_3^2+10.95\pi_3-36.24\pi_4^2+29.73\pi_4-279.5\pi_5^2+83.64\pi_5$ --- **(a)**

$$HTC_A = -14.53 - 0.321\left[\frac{L_R}{(V_E)^{\frac{1}{3}}}\right]^2 + 3.437\left[\frac{L_R}{(V_E)^{\frac{1}{3}}}\right] - 14.28\left[\frac{D_R}{(V_E)^{\frac{1}{3}}}\right]^2 + 30.85\left[\frac{D_R}{(V_E)^{\frac{1}{3}}}\right] - 16986\left[\frac{D_M}{(V_E)^{\frac{1}{3}}}\right]^2 + 10.95\left[\frac{D_M}{(V_E)^{\frac{1}{3}}}\right] - 36.24\left[\frac{V_W}{V_R}\right]^2 + 29.73\left[\frac{V_W}{V_R}\right] - 279.5\left[\frac{Nm_w}{P(V_E)^{\frac{1}{3}}}\right]^2 + 83.64\left[\frac{Nm_w}{P(V_E)^{\frac{1}{3}}}\right] \text{---.(b)}$$

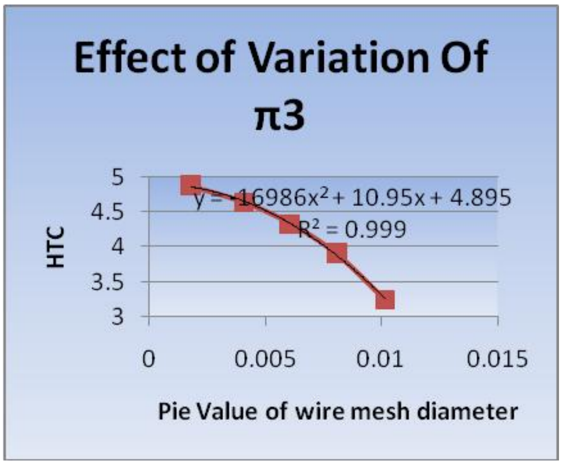
Effect of Variation of π terms On Actual Based Heat Transfer Coefficient.



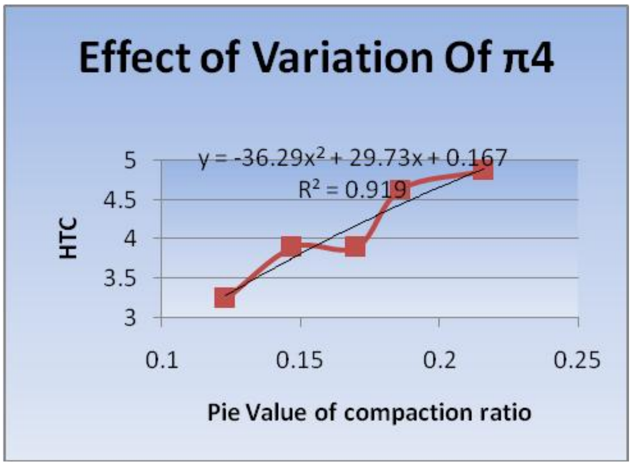
Graph No.6



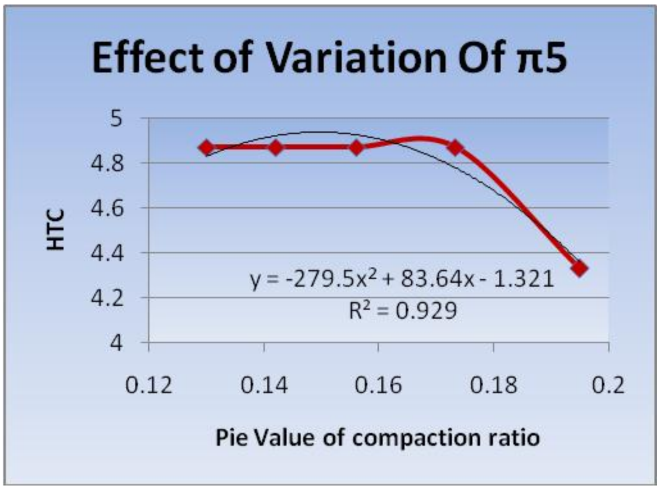
Graph No.7



Graph No.8



Graph No.9



Graph No.10

ABBREVIATIONS USED IN ABOVE TABLE AND CALCULATIONS ARE

LR Length of Regenerator DR Diameter of Regenerator
DM Diameter of Mesh Wire VM Volume of Mesh wire
VR Volume of Regenerator P Charging Pressure
VE Expansion Space Volume W Power consumed
N Speed of the machine TI Compression Space Temperature
T4 Expansion Space Temperature COPT Theoretical Coefficient of Performance
COPA Actual Coefficient of Performance HTCT Theoretical Heat Transfer Coefficient
HTCA Actual Heat Transfer Coefficient mw Mass flow rate of water
HTOA Overall Actual heat transfer coefficient HTOT Overall Temperature based heat transfer coefficient

$\pi 1 = LR/VE^{1/3}$ $\pi 2 = DR/VE^{1/3}$
$\pi 3 = DM/VE^{1/3}$ $\pi 4 = VM/VR$
$\pi 5 = Nmw/PVE^{1/3}$

COPTE = experimental temperature based Coefficient of Performance
COPA = theoretical temperature based Coefficient of Performance

V. CONCLUSIONS

1) It is observed that evaluated theoretical coefficient of performance by assuming the heat transfer in regenerator takes place by convection and experimentally COP are different and varying by large range i.e 40 to 70 %.

2) To improve the performance of model, Heat transfer coefficient must be modified by to account for heat transfer in a regenerator by all three modes i.e conduction, convection & radiation.

3) It is established that as the length of the regenerator goes on increasing the actual Heat Transfer Coefficient and temperature based Heat Transfer coefficient both are increased. The actual heat transfer coefficient is maximum when regenerator length = $5.353 \times V_E^{1/3}$.

4) It is established that as the compaction ratio (i.e. volume of wire mesh in regenerator / regenerator volume) heat transfer coefficient of performance both are increased. The actual coefficient of performance is maximum when compaction ratio (i.e. volume of wire mesh in regenerator / regenerator volume) is 0.410.

REFERENCES

1. G Thombare, D.G, S.K. Verma" Technological development in the Stirling cycle engines. Renewable and sustainable Energy Reviews", 12, 2008, pp. 1-38.
2. G. Walker "Stirling-cycle machines" Book, Oxford, Clarendon Press, 1973 ix, pp156
3. Nitesh B Umale,DhananjayThombre "Theoretical Analysis of Effect of Regenerator Geometry and Material on Stirling Engine Performance",conference paper ,june 2015 pp 38-43
4. M. Tarawneh , F. Al-Ghathian "Numerical Simulation and Performance Evaluation of Stirling Engine Cycle" Jordan Journal of Mechanical and Industrial Engineering Volume 4, Number 5, November 2010 ISSN 1995-6665 pp 615 – 628
5. S.K.Choudhary et.al "Design and performanceof an Eco-Friendly Refrigeratio System working on stirlingcycle"International Conference at deptt of MechEngg.on Energy and Environment,Faculty of Engg&Technology,JamiaMillia Islamic New Delhi 23-24 JAN 2004 pp 496-500.
6. S.K.Choudhary " Experimental Optimization of some sub systems of stirling cycle refrigeration with Non CFC Gases" Thesis, April 2010