DESIGN AND ANALYSIS OF RECUPERATOR IN MINI GAS TURBINE SETUP

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ABSTRACT

The GT85-2 Gas turbine set-up is the two shaft turbine running model. It is produced to understand the basic principle of gas turbine cycle. It has net power output of upto 10kW with compression ratio upto 2 bar. The fuel used is ATF. Power is measured by absorption type eddy current dynamometer. Power produced by the first turbine is totally utilized to drive the compressor. Net power output would be the power produced by the power turbine. The exhaust gases coming out from the power turbine has the temperature range of 400˚C-650˚C whose energy was exhausted to the atmosphere. Here the same energy has been utilized to pre-heat the compressed air before allowing it to enter into the combustion chamber. The device performing the above function has been designed and manufactured keeping in mind many constraints like available area, time period, costing, temperature range, pressure drop. The device is known as REGENERATOR a type of heat exchanger. The comparison between performance of the simple and regenerative cycle has been carried out for the same operating parameters ranging from part load to full load conditions. In order to understand the difference quickly various graphs have been drawn from the actual values of readings. Also what work could be possible for the future for the same set-up has also been suggested.

Keywords: GT85-2 Gas Turbine, Concentric Tube Heat Exchanger, Recuperator, Regenerative Cycle, LMTD

I. INTRODUCTION

Gas turbine performance can be judged with respect to its efficiency, power output, specific fuel consumption and work ratio. There are several parameters that affect its performance, such as the compressor pressure ratio, the combustion inlet temperature and the turbine inlet temperature (TIT). Obviously, these parameters have been actually improved by various gas turbine manufactures. Meanwhile, operating parameters such as ambient temperature, altitude, humidity, inlet and exhaust losses also affect the performance of gas turbine units. Gas turbines that operate in simple cycles have low efficiencies because the turbine exhaust gases come out very hot and this energy is lost to the atmosphere.

II. REMEDY TO OBTAIN BETTER EFFICIENCY IN SIMPLE CYCLE

Better performance is reached with advanced cycles that take advantage of the energy contained in the turbine exhaust gases to improve the cycle performance. In gas-turbine power plant, the temperature of the exhaust gas leaving the turbine is often considerably higher than the temperature of the air leaving the compressor. Therefore, the air leaving the compressor at high-pressure can be heated by transferring heat to it from the hot exhaust gases in a counter-flow heat exchanger, which is also known as a regenerator or recuperator. Gas turbine regenerators are usually constructed as shell- and-tube type heat exchangers using very small diameter tubes, with the high pressure air inside the tubes and low pressure exhaust gas in multiple passes outside the tubes. The thermal efficiency of the Brayton cycle increases as a result of regeneration since the portion of energy of the exhaust gases that is normally rejected to the surroundings is now used to preheat the air entering the combustion chamber. This, in turn, decreases the heat input (thus fuel) requirements for the same net work output. Pressure drop through the regenerator is important and should be kept as low as practical on both sides. Generally, the air pressure drop on the high-pressure side should be held below 2% of the compressor total discharge pressure. The effectiveness of most regenerators used in practice is below 0.85.

III. DESIGN OF CONCENTRIC TUBE COUNTER FLOW HEAT EXCHANGER

A counter-flow, concentric tube heat exchanger is to be used to heat the compressed air coming out from the compressor in a GT85-2 GTP setup. The flow rate of compressed air through the inner tube varying diameter (Di = 53 mm at smaller end to 77.35mm larger end) and the flow rate of exhaust outer annulus (Do = 140 mm) are taken from the reading no.3 of mini gas turbine available in turbo-lab in mechanical department, Faculty of Techo & Engg, M.S.U. Baroda. The compressed air is expected to heat by 20˚C. How long the outer tube must be made if the diameter is taken 0.140 m. Design the concentric tube heat exchanger for the same.

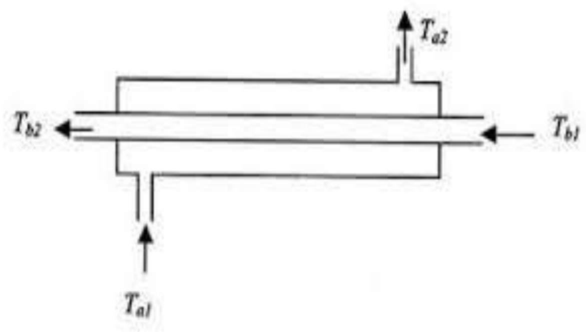

Fig 1: Schematic Diagram Of Concentric Tube Heat Exchanger

IV. OBJECTIVE IS TO RISE THE TEMPERATURE OF COMPRESSED AIR

The following expression has been utilized to find out the performance parameters of mini gas turbine plant.

4.1 Mass Flow Rate Of Air

$$\frac{\dot{m}_a\sqrt{T_1}}{P_1} = 0.3005 \times \sqrt{\frac{\Delta P}{P_1}}$$

Where, $\dot{m}_a$ is in kg/s, T_1 is in K, P_1 is in bar, ΔP is in mm of Hg.

4.2 Properties Of Air And Exhaust Gases

$$\text{Now, } T_{c1} = 98^\circ C = 371K \,\&\, T_{h1} = 815 - 50^\circ C = 492^\circ C = 765K$$

<u>Iteration No.1:-</u> Temperature rise of compressed air is 100˚C

$$\text{So, } T_{c2} = 198^\circ C = 471K \,\&\, T_{h2} = 815\,K - 765K = 465K$$

Assuming reduction of temperature of gas temperature by Δ300˚C because only 1⁄3rd portion of mass flow rate of gases are allowed to enter into the heat exchanger.The properties c_{pa}, μ, Pr are taken at buck mean temperature(T_{cm}) from table [7]

4.3 Heat Gained And Heat Lost In A Concentric Tube Heat Exchanger

$$Q_a = \dot{m}_{air} c_{pa}(T_{c2} - T_{c1}) \,\&\, Q_g = \frac{m_g}{3} c_{pg}(T_{h1} - T_{h2})$$

From above equations calculating heat transfer, if heat gained and heat lost are not equal then again calculate T_{h2} for heat transfer Q_a and use its value for further calculation.

4.4 Logarithmic Mean Temperature Difference (Lmtd)

$$LMTD = \frac{(T_{h1} - T_{c2}) - (T_{h2} - T_{c1})}{\ln\left(\frac{T_{h1} - T_{c2}}{T_{h2} - T_{c1}}\right)}$$

4.5 Heat Transfer Coefficient For Compressed Air (h_i) And Exhaust Gases (h_o)

$$Nu = \frac{hd}{k} = 0.0214\,(Re^{0.8} - 100)Pr^{0.4}$$

Where, the expression of Nusselt number (Nu) is taken from reference [8]

4.6 Overall Heat Transfer Coefficient Based On Inside Area (U_i)

$$\frac{1}{U_i} = \frac{1}{h_i} + \frac{A_i \times \ln\left(\frac{r_o}{r_i}\right)}{2\pi kl} + \frac{A_i}{A_o}\frac{1}{h_o}$$

4.7 Length Required For Obtaining Temperature Rise Of Compressed Air By 100˚C (L)

$$Q = U_i A_i (LMTD) = U_i \times 2\pi r_i L \times (LMTD)$$

Table 2:- The final iterations done for air side heat transfer coefficient

m_a	P_2	T_{c1}	T_{c2}	ΔT_c	T_{cm}	P_a	C_{pa}	μ	k	Pr	V_a	Q_a	Re_a	Nu_a	h_a
0.1927	171325	371	471	100	421	1.418	1.017	2.37E-05	0.0351	0.686	40.96	19.59	159219.5	265.18	143.28
0.1927	171325	371	421	50	396	1.507	1.014	2.29E-05	0.0337	0.689	38.52	9.77	165125.8	273.55	141.61
0.1927	171325	371	401	30	386	1.547	1.013	2.23E-05	0.0327	0.691	37.55	5.85	169045.1	279.09	140.58
0.1927	171325	371	391	20	381	1.567	1.012	2.21E-05	0.0320	0.692	37.07	3.90	170804.4	281.59	138.63
0.1927	171325	371	381	10	376	1.588	1.012	2.18E-05	0.0318	0.693	36.58	1.95	173154.9	284.87	139.54

Table 3:- The final iterations done for gas side heat transfer coefficient

m_g	P_2	T_{h1}	ΔT_h	C_{pg}	μ	k	Pr	T_{h2}	T_{hm}	ρ_g	V_g	Re_g	Nu_g	h_g
0.19519	110953	765	300	1.0576	3.07E-05	0.0475	0.681	480.26	622.63	0.621	39.63	57000.2	115.19	77.01
0.19519	110953	765	150	1.0752	3.33E-05	0.0523	0.684	625.34	695.17	0.556	44.25	52432.6	107.84	79.44
0.19519	110953	765	90	1.0777	3.36E-05	0.0530	0.685	681.50	723.25	0.535	46.03	51995.7	107.14	79.98
0.19519	110953	765	60	1.0814	3.42E-05	0.0540	0.685	709.57	737.29	0.524	46.93	51083.5	105.64	80.34
0.19519	110953	765	30	1.0856	3.48E-05	0.0551	0.686	737.40	751.20	0.515	47.81	50202.7	104.21	80.88

Table 4:- The final iterations done for length requirement

SR NO.	ΔTc	ha	hg	Ui	LMTD	ri	Q	L
1	100	143.28	77.01	50.09	186.63	0.062	19.59	5.402
2	50	141.61	79.44	50.89	296.92	0.062	9.77	1.660
3	30	140.58	79.98	50.98	336.54	0.062	5.85	0.876
4	**20**	**138.63**	**80.34**	**50.86**	**355.99**	**0.062**	**3.90**	**0.553**
5	10	139.54	80.88	51.20	375.13	0.062	1.95	0.261

From table 4 it is seen that if we want to increase the temperature of compressed air by 20˚C, the required length of concentric tube heat exchanger is 0.553m. but the limitation is 0.45 m. Hence the outer tube having 0.140 m outer diameter and 0.45 m length is manufactured.Temperature rise by the compressed air can be achieved by interpolation

$$\Delta T_c = 10 + \frac{(0.45 - 0.261)}{(0.553 - 0.261)}(20 - 10) = 16.5^{\circ}C$$

V. TEST RESULTS AND DISCUSSION

The GTP setup was run for four different pressures i.e. 0.5, 0.6, 0.7, and 0.8 after attaching the recuperator. And keeping 6000 rpm speed of power turbine constant torque produced in kg was measured. Also properties of air at all the sections were measured. The performance evaluation of the simple and regenerative cycle has been carried out using the following performance formulae.

5.1 Power Output Of Power Turbine Indicated By Dynamometer (W_{actual})

$$W_{actual} = \frac{2\pi N_2 T}{60000}$$

Where, $T = F \times R = mg \times R$

5.2 Overall Thermal Efficiency (η_o)

$$\eta_o = \frac{W_{actual}}{\dot{m}_f \times CV}$$

5.3 Specific Fuel Consumption (SFC)

$$SFC = \frac{\dot{m}_f \times 3600}{W_{actual}}$$

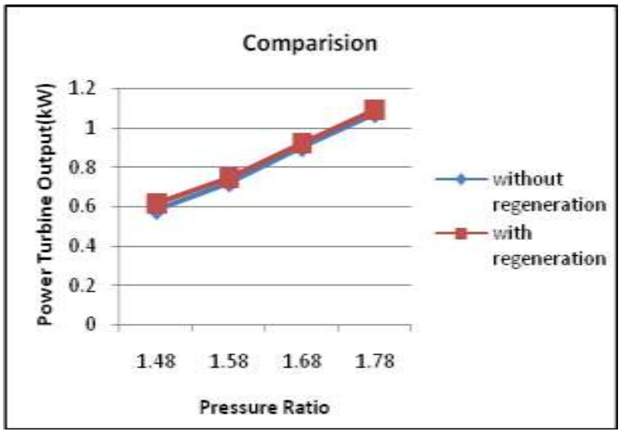

Fig 2: Variation of Actual Power Output with Pressure Ratio

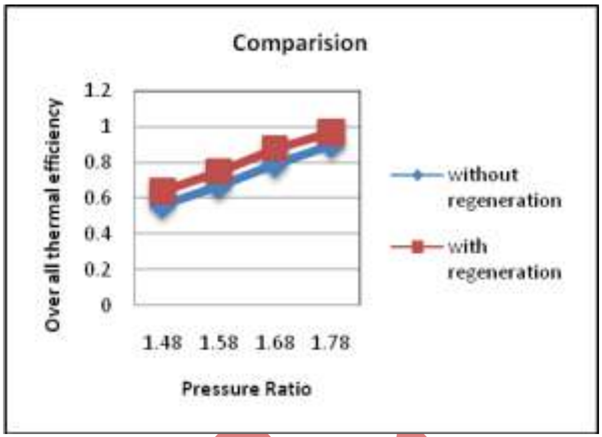

Fig 3: Variation of Overall Thermal Efficiency with Pressure Ratio

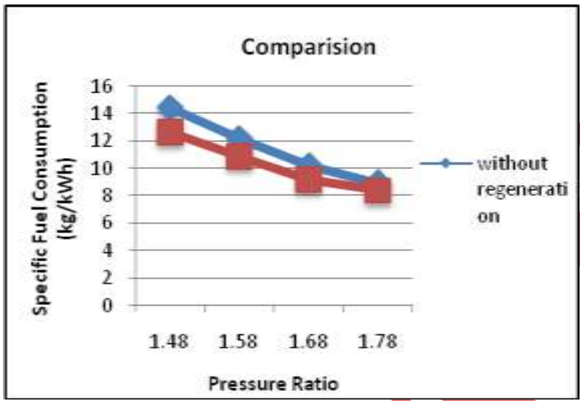

Fig 4: Variation of Specific Fuel Consumption with Pressure Ratio

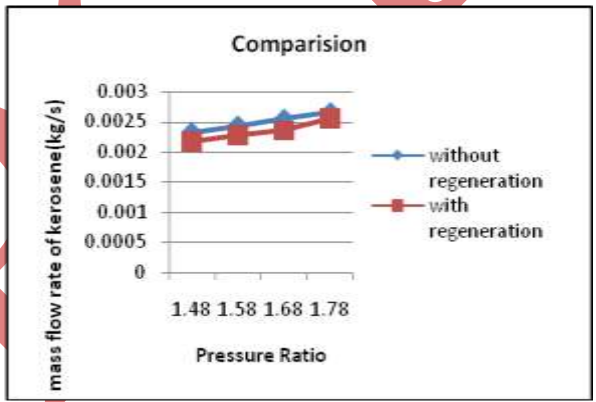

Fig 5: Variation of Mass Flow Rate of Kerosene with Pressure Ratio

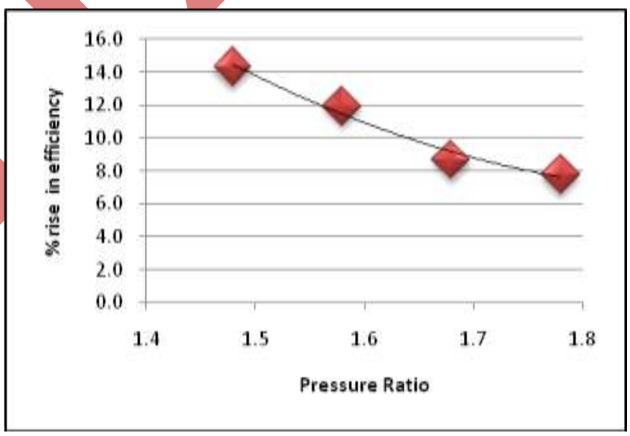

Fig 6: Variation of % rise in efficiency with Pressure Ratio

VI. CONCLUSIONS

Power turbine output increases with pressure ratio. The rate of increase in power turbine output with regenerator is slightly higher than that of without regenerator.Overall thermal efficiency increases with pressure ratio in both with and without regenerator. It gets higher overall thermal efficiency for with regeneration than that without regeneration.

Specific fuel consumption decreases with pressure ratio in both with and without regenerator. It gets lower Specific fuel consumption for with regeneration than that without regeneration.Mass flow rate of kerosene used is less for regenerative cycle than that in simple cycle for the same speed of power turbine.

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[7] Table A-5&6 Properties of air and gases at atmospheric pressure, Appendix A-Tables, "HEAT TRANSFER" by J.P.Holman. 8th Edition.

[8] Nusselt Number for tubes when $10^4 < Re < 5 \times 10^6$&$0.5 < Pr < 1.5$, Heat Transfer by J.P.Holman, 10th Edi. Equation 6.4b,