การศึกษาความเป็นไปได้ของการใช้ Regenerative Intercooled Cycle สำหรับการขับเคลื่อนในเรือรบขนาดเล็ก

FEASIBILITY STUDY OF REGENERATIVE INTERCOOLED CYCLE FOR MARINE PROPULSION IN SMALL NAVAL SHIP

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Regenerative Intercooled Cycle ได้ถูกนำมาวิเคราะห์เพื่อแสดงให้เห็นถึงประโยชน์ของวัฏจักรดังกล่าวในเชิงของการประหยัดเชื้อเพลิงเมื่อเทียบกับ Simple Cycle รูปแบบทางเทอร์โมไดนามิกส์ ของ Regenerative Intercooled Cycle ได้ถูกนำมาวิเคราะห์ เพื่อหาสมรรถนะ และอัตราการใช้เชื้อเพลิงที่ดีที่สุด การวิเคราะห์ได้แสดงให้เห็นถึงความเป็นไปได้ในการที่จะแทนที่ Simple Cycle ด้วย Regenerative Intercooled Cycle อัตราการใช้เชื้อเพลิงสามารถที่จะลดลงได้มากกว่า 25% การวิเคราะห์เกี่ยวกับตัวแลกเปลี่ยนความร้อนที่ใช้ในวัฏจักรชี้ให้เห็นถึงประโยชน์ของ Plate Fin Counterflow Heat Exchanger ที่เหนือกว่าตัวแลกเปลี่ยนความร้อนแบบอื่น ๆ เมื่อนำมาใช้เป็น Marine Regenerator การวิเคราะห์ยังได้รวมถึงการศึกษาขนาดและน้ำหนักรวมทั้งการจัดวางโรงจักร เพื่อเปรียบเทียบ ระหว่างการลดลงของอัตราการใช้เชื้อเพลิงกับการเพิ่มขึ้นของน้ำหนักจากการศึกษาพบว่า สำหรับอัตราการใช้เชื้อเพลิงลดลง 25.6% น้ำหนักของโรงจักรจะเพิ่มขึ้น 17.84% และ 34.16% สำหรับโรงจักรขนาด 20,000 HP และ 40,000 HP ตามลำดับ

ABSTRACT

The Regenerative Intercooled Cycle was analysed in order to highlight the potential benefits of this cycle in term of fuel economy compared with the Simple Cycle. The thermodynamic model of a Regenerative Intercooled Cycle was conceived in order to find the optimal performance and fuel consumption. The analyses show the possibility in replacing the Simple Cycle with the Regenerative Intercooled Cycle. The Improvement of SFC by more than 25% can be realized. The analysis of heat exchanger demonstrates many advantages of plate fin counter flow heat exchanger over other types of heat exchangers and hence an attractive candidate for marine regenerator. The size and weight including the arrangement of the plant was analyzed in order to compare significant SFC improvement with reasonable weight increase. With 25.6% decrease in SFC for the Regenerative Intercooled Cycle, plant weight increase of 17.84% and 34.16% are achieved for 20,000 HP and 40,000 HP power plant, respectively.

1. Introduction

Gas turbine engine has proved itself to be a well established prime mover with many applications in both transport and stationary duties, especially in marine duties. Up to now it is more than 30 years since the gas turbine first went to the sea and become the most attractive propulsion plant for the naval ships. The attractions of this power plant are such that nearly all major naval vessels currently under construction, or planned, feature gas turbines, either wholly or in part, in their propulsion machinery.

After the Faulkland war, the ideas about the benefits of big naval ship have been eliminated. Many navies in the world are looking for small naval ships with increase effectiveness and cost benefits. Many designs have been dedicated to achieve high levels of performance and effectiveness with ships of significant smaller size. As a primary mission of small naval ships, for example, escort requirement of 4,500 nm (without refueling) and a dash speed in excess of 30 knots are considered essential. A minimum escort mission is considered to be comprised of 2,000 nm 16 knots, 2,000 nm at 21 knots and 500 nm at 30 knots dash speed. For more advanced escort duties a higher speed for the same mission will be expected. Normally these mission profiles can be acheived by using the simple gas turbine engine as a prime propulsion for small naval ships. However, the simple gas turbine engine still faces the problem of high fuel consumption, especially for the operation of the engine at part load, which usually take about 80% of the overall operating time. The simple cycle is inherently less efficient in these circumstances. The deterioration of thermal efficiency at low power level with simple cycle engines represents an undesirable characteristic which is not readily overcome.

Several methods for improving the efficiency of this cycle have been proposed, for example adding of the regenerator or the intercooler into the system. However, these methods also made the system more complex, and difficult to control the system in order to acheive theorectical performance. The engine were both costly and bulky. However, the development in compact heat exchanger and many new heat transfer surfaces have helped reduction the size of heat exchangers and made those methods to be more feasible in the present time. Utilization of the intercooled regenerative engine type offers maximum benefits in term of fuel consumption to small naval ship propulsion, this configuration is selected for advanced cruise engine study in this study and will be compared to the conventional simple gas turbine.

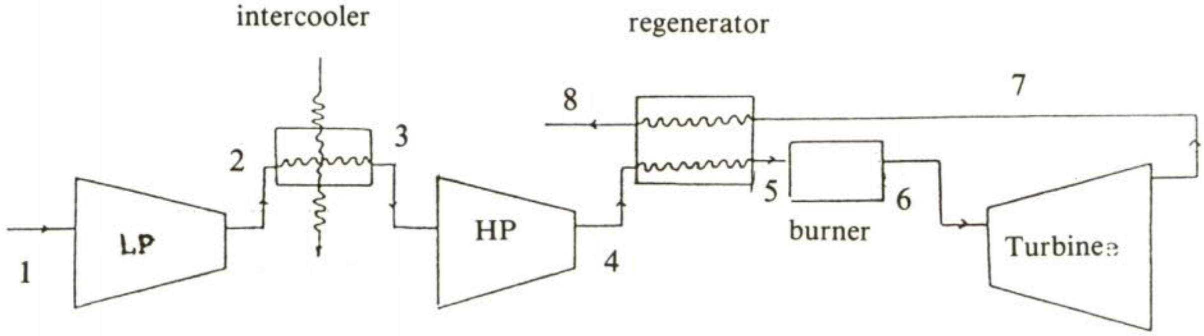

Fig. 1 Thermodynamic model of intercooled regenerative gas turbine

2. The Intercooled Regenerative Gas Turbine Cycle

The basic configuration of this cycle is shown in Fig. 1. It comprises of two compressors, an intercooler, a combustor, a regenerator and a turbine. The cycle will be analyzed based on the following assumptions:

- Heat losses in the system is neglected.
- The working fluid is a perfect gas with semi-constant specific heat throughout the cycle.

The notations used in the analysis are as follows:

P_i = intermediate pressure
P_1 = inlet pressure at low pressure compressor
P_2 = inlet pressure at intercooler
T_1 = inlet temperature at low pressure compressor
T_2 = inlet temperature at intercooler
T_3 = inlet temperature at high pressure compressor
T_4 = inlet temperature at cold side of regenerator
T_5 = inlet temperature at burner
T_6 = inlet temperature at turbine
C_{pa} = specific heat at constant pressure of air
C_{pf} = specific heat at constant pressure of fuel
C_{pg} = specific heat at constant pressure of gas
r = ratio of specific heat = C_p/C_v
n_{c1} = isentropic efficiency of low pressure compressor
n_{c2} = isentropic efficiency of high pressure compressor

Work done by the compressor per unit mass of working fluid can be calculated by assuming isentropic compression

$$W_c = \frac{C_{pa}T_1(T_2 - T_1)}{n_{c1}} + \frac{C_{pa}T_3(T_4 - T_3)}{n_{c2}}$$
$$= \frac{C_{pa}T_1\,[(P_1/P_1)^{(r-1/r)} - 1]}{n_{c1}} + \frac{C_{pa}T_3[(P_2/P_i)^{(r-1/r)} - 1]}{n_{c2}} \quad (1)$$

The required condition for minimum work is:

$$\frac{dW_c}{dP_i} = 0 \quad (2)$$

completing the derivation and simplifying, the intermediate pressure is

$$P_i = P_1P_2\,\frac{T_3 n_{c1}}{T_1 n_{c2}}^{\,r/r-1} \quad (3)$$

Assume a perfect intercooler, equation (3) is reduced to

$$P_1 = P_1P_2 \quad (4)$$

The minimum work done by compressor is

$$W_c = \frac{2C_{pa}T_1[(P_2/P_1)^{r-1/2r} - 1]}{n_c} \quad (5)$$

In general the performance of the intercooler is described by the cooling factor which is defined by:

$$E_{ni} = \frac{T_2 - T_3}{T_2 - T_1} \quad (6)$$

Heat supplied per unit mass of working fluid is given by:

$$Q_s = C_{pf}(T_6 - T_5) \quad (7)$$

where T_5 depends on the regenerator effectiveness

$$E_{rox} = \frac{T_5 - T_4}{T_7 - T_4} \quad (8)$$

The above analysis shows that the mass of fuel supplied in the combustion chamber can be reduced by raising the regenerator exit temperature as high as possible. It is theorectically possible to raise the temperature of compressed air coming out from the compression from T_4 to $T_5 = T_7$ and lower the temperature of the gases coming out from the turbine from T_7 to $T_8 = T_4$ by passing both fluids through a counter flow heat exchanger. But in practice this can not be done because there is a practical limit to the size of heat exchanger.

Work developed by the turbine per unit mass of working fluid is given by:

$$W_t = C_{cx}(T_6 - T_7) \quad (9)$$

W_t remains unchanged for the simple cycle as the inlet temperature to the turbine and overall pressure ratio with or without intercooler or regenerator remains the same since the use of regenerator is for reducing the quantity of fuel supplied.

$$n_{th} = \frac{W_t + W_c}{Q_5} \quad (10)$$

3. Marine Heat Exchanger

Most heat exchangers on a ship are of the "surface" type; that is, there is no direct contact between the cold and not fluid. This means that heat must flow first through one film and then through the solid material of which the surface is made and then through the other film.

Selecting the type of regenerator for any marine application is heavily depend on the gas turbine parameters. However, recent improvement in manufacturing makes plate fin exchangers very competitive with tube shell and become the most attractive regenerator for the naval ship. Another reason is that the regenerator for the naval ship design nowsaday is looking for the high effectiveness ($\omega > 85\%$) plate fin counter flow exchangers, especially in the light of increased fuel cost. Therefore this type of regnerator will be considered in term of design, size and performance.

The design of the regenerator for the intercooled regenerative cycle is based on the following objective constraints:

- The constraints of cycle performance
- Minimum volume of the regenerator

The design process was based on establishing the constant of the total pressure drop in the regenerator in order to maintain the net power output of the gas turbine plant. Therefore the minimum volume for the heat exchanger constitutes the sole criterion for the optimum design [3]. To minimize the volume of the regenerator, several surfaces have to be investigated and compared to determine which is the best to use in our design process. According to Soland [4], he concluded that the wavy fin plate finned surface 17.8 – 3/8 W was the "best" for the specific case. This conclusion was later evaluated by Sheldon [5]. Sheldon also proposed the performance parameter curve which showed that the wavy fin plate finned surface (17.8 – 3/8W) has the good performance at high Reynold number. Therefore this surface was selected to use in this design.

4. Optimization of The Intercooled Regenerative Cycle

From previous analysis, it is shown that the two importance factors in the optimization of the intercooled regenerative cycle are the cycle parameters and regenerator sizing. In this section, it is proposed to determine how these two factors affect the performance of the cycle and to establish the most efficient cycle based on the assumption made by using the thermodynamic modelling computer program of gas turbine.

This program comprises of 4 main subprograms. The description of each subprogram is as followed:

— Subprogram "Comp" represents the thermodynamic performance of the compressor by assuming the perfect gas (with semi-constant of specific heat) as the working fluid.

– Subprogram "Inter" represents the thermodynamic performance of the intercooler using the cooling factor; this subprogram also includes the calculation for the high pressure compressor and correct the total work done by the compressor.

– Subprogram "Burn" represents the thermodynamic performance of the combustor. The calculation in this subprogram is statred by guessing the first amount of fuel and using the reaction equation to calculate the composition of the gas leaving the combustor. An iteration process has been used to refine the amount of fuel from the first guess. The pressure drop in the combustor is also calculated in this subprogram.

– Subprogram "Gtur" represents the thermodynamic performance of the turbine. This subprogram includes cooling the turbine blades. Two methods had been used to check whether the turbine required as additional cooled stage or not. One is the pressure drop check and the other is the temperature check. The amount of cooling air is extracted from the high pressure compressor based on the equation of the curve of coolant flow V.S. cooling effectiveness. To analyze the thermodynamic performance of the regenerator in the intercooled regenerative cycle, the turbine exit temperature is first calculated by assuming a perfect gas and an adiabatic process in turbine. Iteration is performed to refine the exit temperature.

Design Parameter for Cycle Optimization Analysis

Mass fraction of N_2	= 0.77
Mass Fraction of O_2	= 0.23
Mass fraction of H_2o	= 0.00
Mass fraction of SO_2	= 0.00
Mass fraction of CO_2	= 0.00
The compressor inlet stagnation pressure	= 101, 325.00 N/m^2
The compressor inlet stagnation temperature	= 300 K
Polytropic efficiency of compressor	= 0.9
Pressure ratio of combustor	= 0.95
Heating value of fuel	= 10,000 Kcal/Kg
Maximum allowable temperature of blade	= variable K
Inlet stagnation temperature of turbine	= varible K
Polytropic efficiency of turbine	= 0.9
Cooling factor	= 0.9
Pressure ratio of intercooler	= 0.97
Regenerator effectiveness	= 0.9
Pressure drop on hot side of regenerator	= 2,000 N/m^2
Pressure ratio on cold side of regenerator	= 0.97

The following cycle optimization analysis was peformed on the three generations of Gas Turbine engine according to the material improvements and cooling technique which have permitted turbine inlet temperature to rise about 20 °F per year since 1959, starting at 1400 °F. Table 1 below shows the expectation from the Gas Turbine Technology.

Table 1 : Conditions of Parametric Study of Based Load Turbines

Condition	Generation I	Generation II	Generation III
Compressor Efficiency assumed	89	92	93
Turbine Efficiency assumed	90	92	93
Compressor bleed air for turbine inlet temperature below	2200	2400	2800 °F
Air for turbine cooling is precooled to 250 °F if turbine inlet temperature is above	2200	2600	3000 °F
Maximum allowable blade root stress	40000	40000	47000 Psi

The analysis was done to establish the performance trends of the simple and intercooled regenerative cycle as a function of compressor temperature rise based on the three turbine inlet temperatures.

The results were plotted as shown in Fig. 2, 3, 4, and 5.

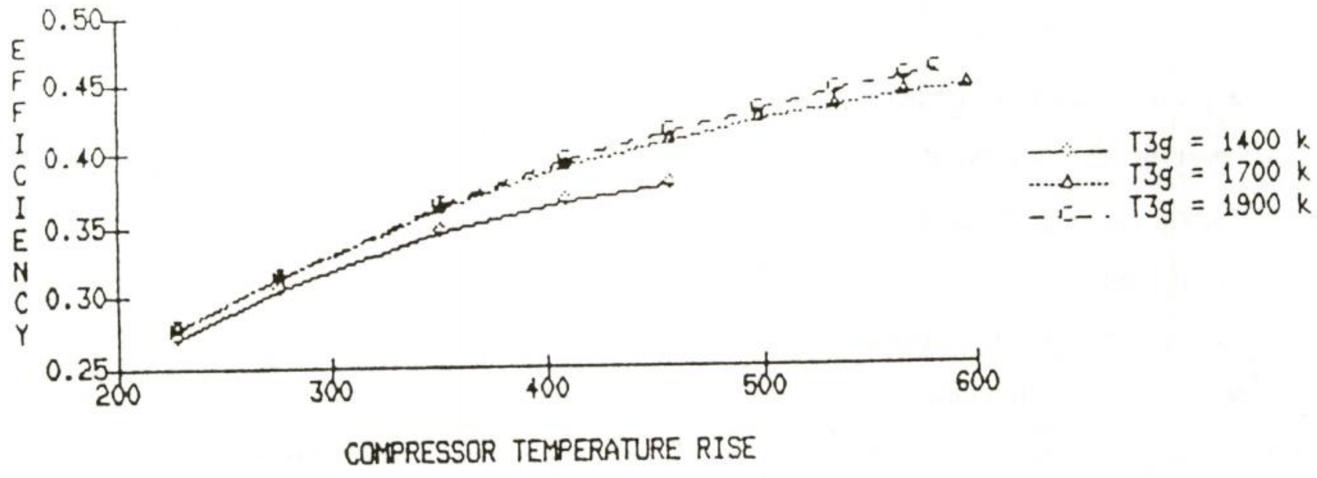

Fig. 2 : Simple Cycle : Efficiency .VS. Compressor Temperature Rise

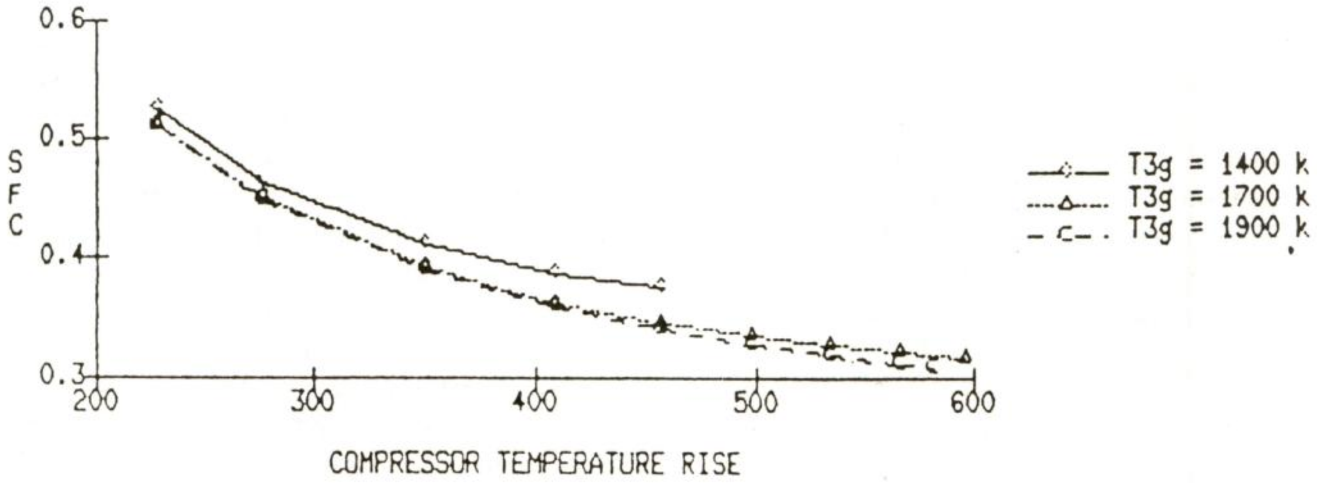

Fig. 3 : Simple Cycle : SFC .VS. Compressor Temperature Rise

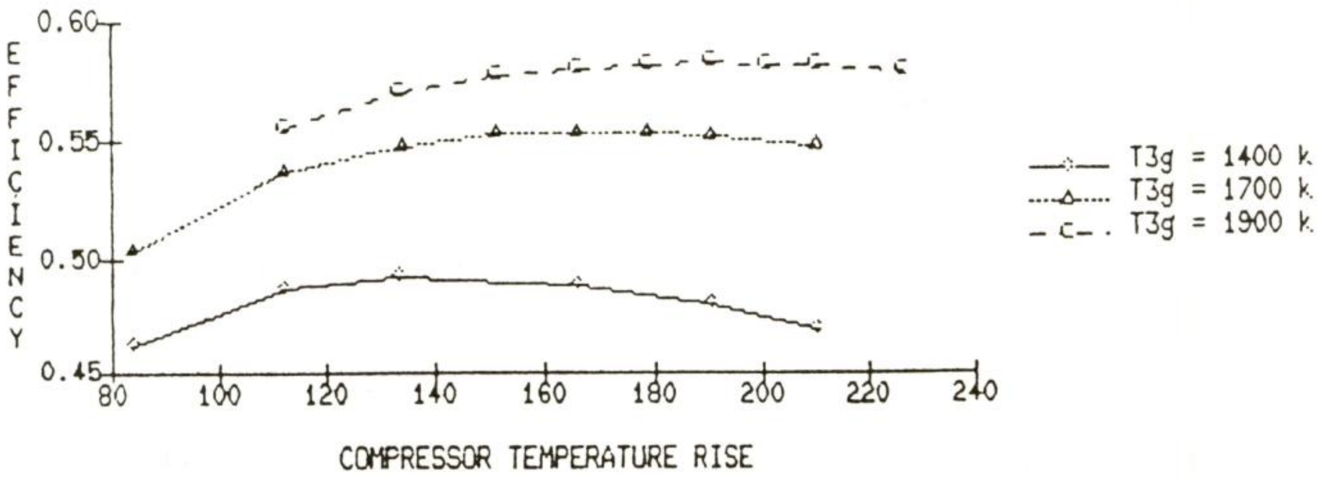

Fig. 4 : Regenerative Intercooled Cycle : Efficiency .VS. Compressor Temperature Rise

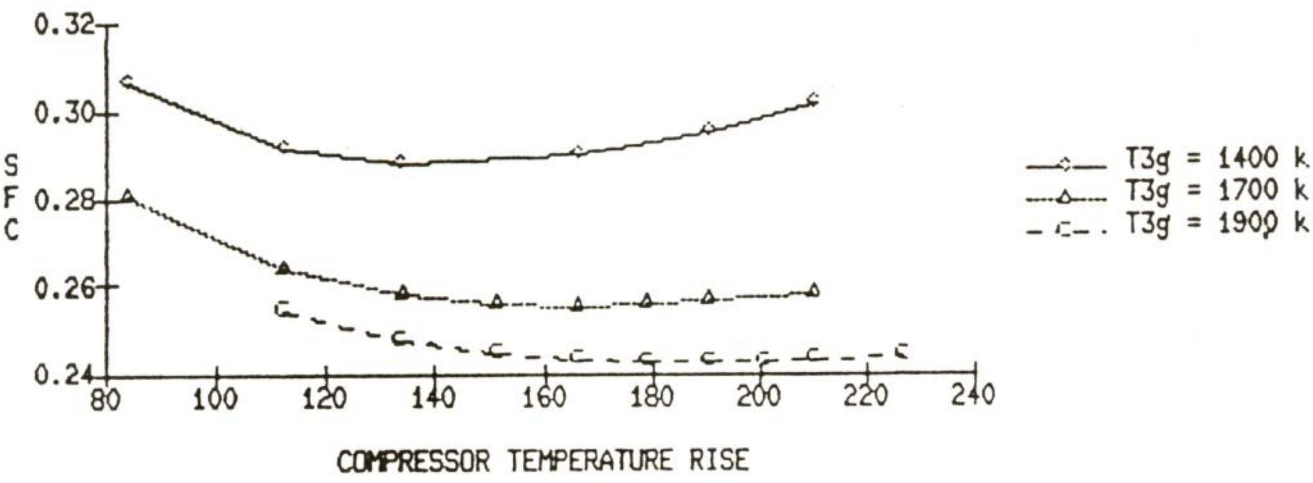

Fig. 5 : Regenerative Intercooled Cycle : SFC .VS. Compressor Temperature Rise

The analysis indicates that the optimum regenerative intercooled cycle for the first generation should have a compression ratio about 8. However, another parameter, regenerator size, stiil plays an important role becuase of space limitation in small naval vessels. The improvement in reducing the regenerator size with a slightly decrease in thermal efficiency can be compromised. With such an advanced gas turbine engine and an effective regenerator will result in reducing not only the fuel cost but also the ship size for given mission profiles.

5. Weight Estimation of the Principal Parts of the Plant

The regenerative intercooled cycle with a compression ratio about 8 was used as a base cycle for calculating the size and weight of the regenerator and intercooler. The data given in Table 2-I and Table 2-II is taken from [7], [8]. Some are based on the empirical formulaes from [9]. The regenerator and intercooler weight were taken from [10] based on the SFC improvement over the simple cycle and frontal area of the regenerator.

The weight comparison in Table 2-I and Table 2-II shows the potential benefit of using the regenrative intercooled cycle over the simple cycle for small propulsion plant.

Table 2-1 : Weight Comparison for 20,000 HP

	Simple Cycle	Regenerative Intercooled Cycle
Propulsion gas turbine	13.9	10.1
Regenerator	--	8.0
Intercooler	--	9.0
Propulsion reduction gear	28.2	28.2
Propulsion shafting	61.3	61.3
Propulsion control system	5.0	7.0
Circulating and Cooling water	8.5	12.0
Miscellaneous	100.0	120.0
Total	216.9	255.8

% increase weight = 17.84 : % decrease SFC = 25.6

Table 2-II : Weight Comparison for 40,000 HP

	Simple Cycle	Regenerative Intercooled Cycle
Propulsion gas turbine	27.8	20.2
Regenerator	--	16.0
Intercooler	--	20.0
Propulsion reduction gear	49.5	49.5
Propulsion shafting	123.0	123.0
Propulsion control system	5.0	7.0
Circulating and Cooling water	10.0	27.3
Miscellaneous	100.0	160.0
Total	315.3	423.0

% increase weight = 34.16 : % decrease SPC = 25.6

** weight in metric ton

6. Plant Arrangement

The plant arrangement is based on the following concepts:

– To provide maximum degree of survivability consistance with the overall space constraints of the ship and machinery.

– To provide the maximum protection for the machinery.

Both concepts are considered to determine the location of the engine room and auxilliary machinery in the naval ship. Normally the engine room is about 3 bulkheads away from the stern and, if there is more than one engine room each engine room should be seperated by at least 3 bulkheads for vulnerability consideration.

The plant layout in the engine room for two engines is represented in figure 6, 7, 8, respectively. The gas turbine engine component should be surrounded by a noise reduction and sirtight enclosure. The enclosure should be mounted on the subbed that is silently mounted on the bedplate foundation. The enclosure will provide engine cooling capability, sound attenuation, internal lighting, view windos, and fire extinguishing capability. Intake and exhaust duct should be constructed in order to provide the minimum pressure drop, flow distortion, or salt ingestion. The reduction gear requires that the shaft and prime mover be in direct connection. The reduction gear should be able to transfer smoothly from one engine to another engine without changing the output horsepower and rpm sufficiently to change the ship speed.

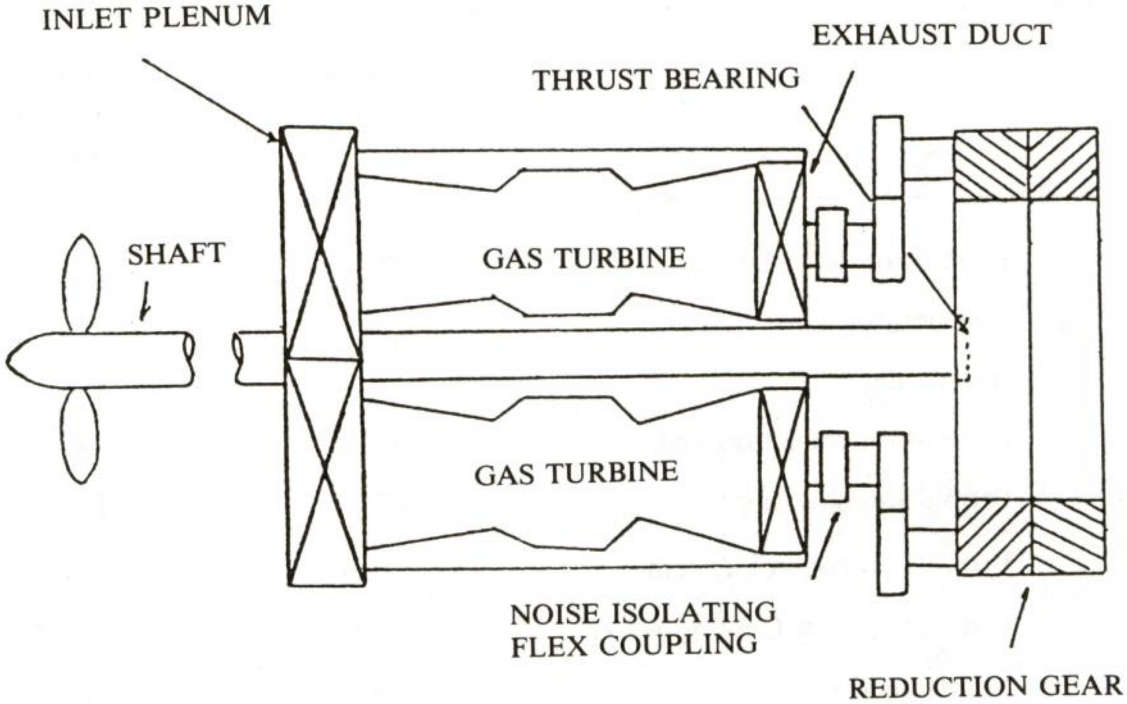

Fig. 6 : Plan view for 2 engines

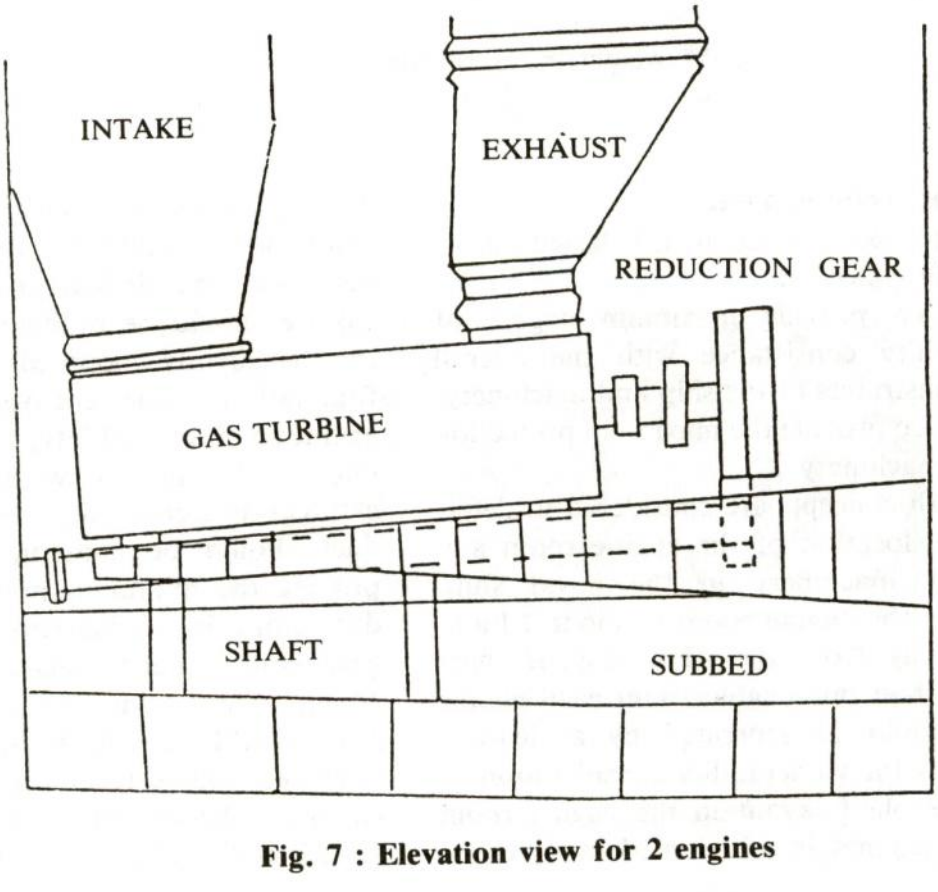

Fig. 7 : Elevation view for 2 engines

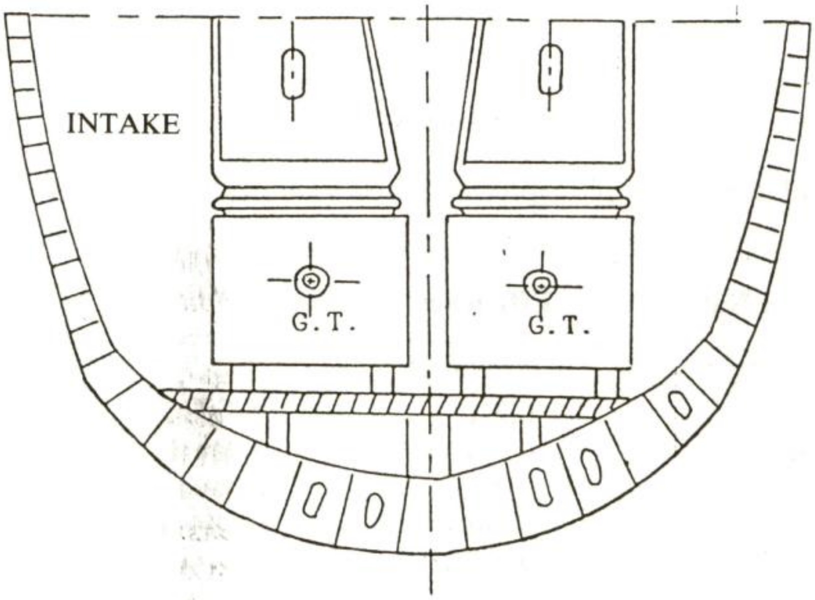

Fig. 8 : Section looking forward for 2 engines

7. Conclusion

The results from the analysis above are presented to illustrate the potential benefit of the regenerative intercooled cycle in terms of fuel economy compared with the simple cycle. It also indicates the possibility of replacing the simple cycle with the regenerative intercooled cycle. The decrease in SFC more than 25% can be realized with the plate fin counter flow regenerator. The size and weight including the plant arrangment were also analyzed to compare the SFC improvement with reasonable weight increase. Further details analysis with new turbine blade cooling technology as well as truely variable specific heat can give more accurate results. This is left for future research to investigate the explicit effect on the results.

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Appendix A

Table A-I: Simple Cycle: Efficiency and SFC for 1^{mt} generation

Compression Ratio	Efficiency	SFC	4 T_c
6	0.27042	0.52300	227.66
8	0.30460	0.46419	275.98
12	0.34449	0.41043	350.61
16	0.36572	0.38660	408.51
20	0.37694	0.37510	456.41

Max. Blade Temp. = 1100 K; Inlet Gas Temp. = 1400 K
$\triangle T_c$ = Compressor Temperature Rise

Table A-II: Simple Cycle: Efficiency and SFC for 2nd generation

Compression Ratio	Efficiency	SFC	$\triangle T_c$
6	0.27657	0.51123	227.66
8	0.31376	0.45063	275.98
12	0.36039	0.39232	350.61
16	0.38910	0.36338	408.51
20	0.40859	0.34604	456.41
24	0.42249	0.33466	497.57
28	0.43262	0.32682	533.84
32	0.44004	0.32131	566.37
36	0.44537	0.31747	595.95

Max. Blade Temp. = 1400 K; Inlet Gas Temp. = 1700 K

Table A-III: Simple Cycle: Efficiency and SFC for 3rd generation

Compression Ratio	Efficiency	SFC	$\triangle T_c$
6	0.27642	0.51151	227.66
8	0.31431	0.44985	275.93
12	0.36265	0.38988	350.61
16	0.39326	0.35953	408.51
20	0.41477	0.34089	456.41
24	0.43078	0.32822	497.57
28	0.44313	0.31907	533.84
32	0.45287	0.31331	566.37
34	0.45698	0.30940	581.49

Max. Blade Temp. = 1600 K; Inlet Gas Temp. = 1900 K

Table A-IV: Regenerative Intercooled Cycle : Efficiency and SFC for 1mt generation

Compression Ratio	Efficiency	SFC	$\triangle T_{c'}$
4	0.46214	0.30595	83.75
6	0.48613	0.29085	112.32
8	0.49150	0.28767	133.36
12	0.48808	0.28969	165.85
16	0.47940	0.29493	190.31
20	0.46980	0.30096	209.88

Max. Blade Temp. = 1100 K; Inlet Gas Temp. = 1400 K

Table A-V : Regenerative Intercooled Cycle : Efficiency and SFC for 2nd generation

Compression Ratio	Efficiency	SFC	$\triangle T_c$
4	0.50448	0.28027	83.72
6	0.53628	0.26365	112.32
8	0.54772	0.25814	133.91
10	0.55203	0.25613	151.16
12	0.55325	0.25556	165.85
14	0.55284	0.25575	178.82
16	0.55150	0.25637	190.31
20	0.54749	0.25825	209.88

Max. Blade Temp. = 1400 K; Inlet Gas Temp. = 1700 K

Table A-VI: Regenerative Intercooled Cycle: Efficiency and SFC for 3rd generation

Compression Ratio	Efficiency	SFC	$\triangle T_c$
6	0.55561	0.25448	112.32
8	0.56975	0.24816	133.36
10	0.57673	0.24516	151.16
12	0.57998	0.24378	165.85
14	0.58137	0.24320	178.82
16	0.58163	0.24309	190.31
18	0.58121	0.24327	200.48
20	0.58034	0.24363	209.88
24	0.57779	0.24471	226.63

Max. Blade Temp. = 1600 K; Inlet Gas Temp. = 1900 K