"STUDY OF COMBINED GAS AND STEAM TURBINE CYCLE WITH PARTIAL STEAM INJECTION AND WITH SOLAR INPUT"

A dissertation

Submitted For Partial Fulfillment Of The Requirements For The Degree Of

Master of Technology

In

THERMAL ENGINEERING

By

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CANDIDATE DECLARATION

I **Harshit Gupta**, roll no. **2K18/THE/07** student of **M.TECH** with specialization in **"Thermal Engineering"** hereby declare that the work, which is being presented in this dissertation, entitled **"Study of combined gas and steam turbine cycle with partial steam injection and with solar input"** and submitted by me for the partial fulfillment of the requirement for award of the degree of Master of technology in Thermal Engineering, from Delhi Technological University, Delhi **,** is an authentic record of my own work carried out under supervision of **Shri P.V. Ram Kumar**, Associate Professor, Department of Mechanical Engineering, Delhi Technological University, Delhi. And not copied from any source without proper citation. This work has not previously formed the basis of the award of any Degree, Diploma Associateship, Fellowship or other similar title or recognition to the best of my knowledge.

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ACKNOWLEDGEMENT

I am immensely grateful to my supervisor **Shri P.V. Ram Kumar** Associate Professor at department of mechanical engineering, DTU for his guidance, unwavering support and encouragement. This major dissertation could not have attained its present form, both in content and presentation, without his active interest, direction and guidance. His personal care has been the source of great inspiration. He has devoted his invaluable time and took personal care in motivating me wherever I was disheartened.

I am also indebted to all the faculty members, my colleagues, and my family for their unwavering love and support.

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ABSTRACT

The Oil Crisis in the 1970's pressurized the world to seek alternatives to conventional sources of energy. But the ascending rate of air pollution and the issue of global warming have made it the need of the hour to start relying on various non-conventional resources for power generation. To check the situation, efforts are still being made to increase the efficiency of currently employed power cycles while at the same time trying to reduce the particulate emissions. STIG is a well-known principle to suppress the air pollutants (specifically the NO_x emissions) or to increase the power output of a gas turbine. In the present work performance of a combined gas and steam cycle with partial steam injection is evaluated. For this purpose, the simple STIG cycle is modified in a way so as to inject a fraction of steam directly into the combustion chamber of a GT cycle and rest is allowed to expand through the steam turbine. An attempt has also been made to discuss the effect of utilizing solar energy in the system for steam generation along with that generated from heat recovery. The thermodynamic energy analysis has been carried out in this project work. For evaluating the configuration, various parameters were varied and their graphs have been obtained to observe their consequences on the work output, efficiency and specific fuel consumption etc. of the proposed cycle. The simulation of the proposed configuration in this work has been carried out in EES software.

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NOMENCLATURE AND GREEK SYMBOLS

Nomenclature

GT	Gas Turbine
TIT	Turbine Inlet Temperature
HRSG	Heat Recovery Steam Generator
IAC	Inlet Air Cooling
STIG	Steam Injection in a Gas Turbine
HP	High Pressure
RSTIG-1	Regenerative STIG cycle 1
RSTIG-2	Regenerative STIG cycle 2
POGT	Partial Oxidation Gas Turbine
LP	Low Pressure
POSTIG	Partial Oxidation STIG cycle
A-STIG	Advanced STIG
CCHPW	Combined Cooling Heating Power and Water Production Cycle
SAR	Steam to Air Ratio (kg/s of steam to kg/s of air)
LHV	Lower Heating Value (kJ/kg)
IAT	Inlet Air Temperature (℃)
sfc	Specific fuel consumption (kg/kW.hr)
$\dot{W}$	Work (kW)
$\dot{m}$	Mass flow rate (kg/s)
T	Temperature (K)
r	Pressure ratio
h	Enthalpy (kJ/kg)
$\dot{Q}$	Rate of heat transfer (kW)
Cp	Specific heat at constant pressure (kJ/kg.K)
$\dot{E}$	Exergy flow rate

Greek Symbols

η	Efficiency
γ	Specific heat ratio
$\sum$	Summation

Subscripts

I	First law
II	Second law
0	Ambient
1, 2, 3 etc.	State point on the cycle
A, B	State point on the cycle
C	Compressor
CC	Combustion Chamber
CH_4	Methane
CV	Control volume
Con	Condensor
Des	Destroyed
ev	Evaporator
f	Fuel
GT	Gas turbine
gas	Flue gases or exhaust gases
P	Pump
rej	Rejection
S	Isentropic
ST	Steam turbine
$steam_{inj}$	Injected steam
sat_{steam}	Saturated Steam

CHAPTER 1

INTRODUCTION

1.1 CURRENT WORLD ENERGY SCENARIO

The power generation in the world before the oil crisis in the 1970s was mainly dominated by conventional fuels like coal, petroleum, natural gas, etc. Though the oil crisis pressurized the world to seek alternatives to conventional sources of energy yet it is the ascending rate of air pollution, the issue of global warming and the limited availability of the fossil fuels that it has become the need of the hour to start relying on various non-conventional resources for power generation.

The world has already started adopting different unconventional energy resources like biomass, wind, solar, tidal, geothermal, nuclear etc. and high percentage of these energy sources fall into the category of renewables. But the generation of power from these resources varies throughout the world relying upon the existing infrastructure and technology present for judiciously utilizing these resources.

The current pandemic situation of COVID-19 has affected the demand for power consumption throughout the world. Figures 1.1(a), 1.1(b), 1.1(c) shown below represent variation in power consumption from various resources for the first quarter of the year 2020 in China, EU, and US respectively to provide a comparison of recent power scenario.

Relative to the first quarter of 2019, the first quarter of 2020 witnessed about 8% decrease in coal demand globally [1]. China was hit the hardest in terms of coal demands in the first quarter; the reasons for this decline are;

1. Growth in demand for renewable resources and cheap gas.
2. The use of coal was also hampered by mild weather [1].

From figure 1.1 (a) it is clear that in China, while there is a decreasing trend in power consumption from coal, yet at the same time the curve in that from renewable resources is moving upwards. These two trends remain the same before and even during the lockdown period in china. The trend from nuclear power is uneven but it can be observed that its consumption has more or less increased during the lockdown; also the power utilization gaseous fuel remained negligibly affected during this period. Despite the decrease in demand from coal-based power, the major portion of power requirement is still fulfilled by it which is more than 60% of the total power consumed in china.

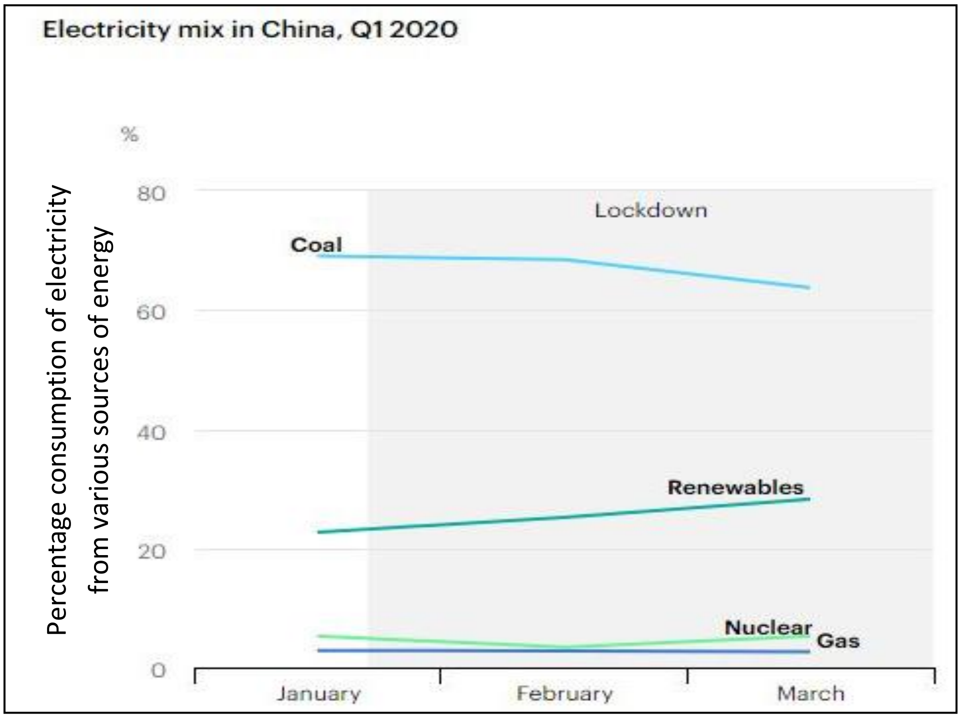

Figure 1.1(a) Power consumption from various energy resources in China for the first quarter of 2020 [2]

Figure 1.1(b) shows the trends of power consumption from various resources in European Union (EU). The trends here are different than that of china. The major portion of total power consumption here is somewhat derived mainly from the renewables, which is approximately 48% of the total, while a significant decrease is there in coal consumption and gas based energy specially during the lockdown period due to COVID-19. Energy consumed from nuclear resources is nearly constant at about 28%. It can be safely said that EU has majorly shifted to renewable means of energy for power generation.

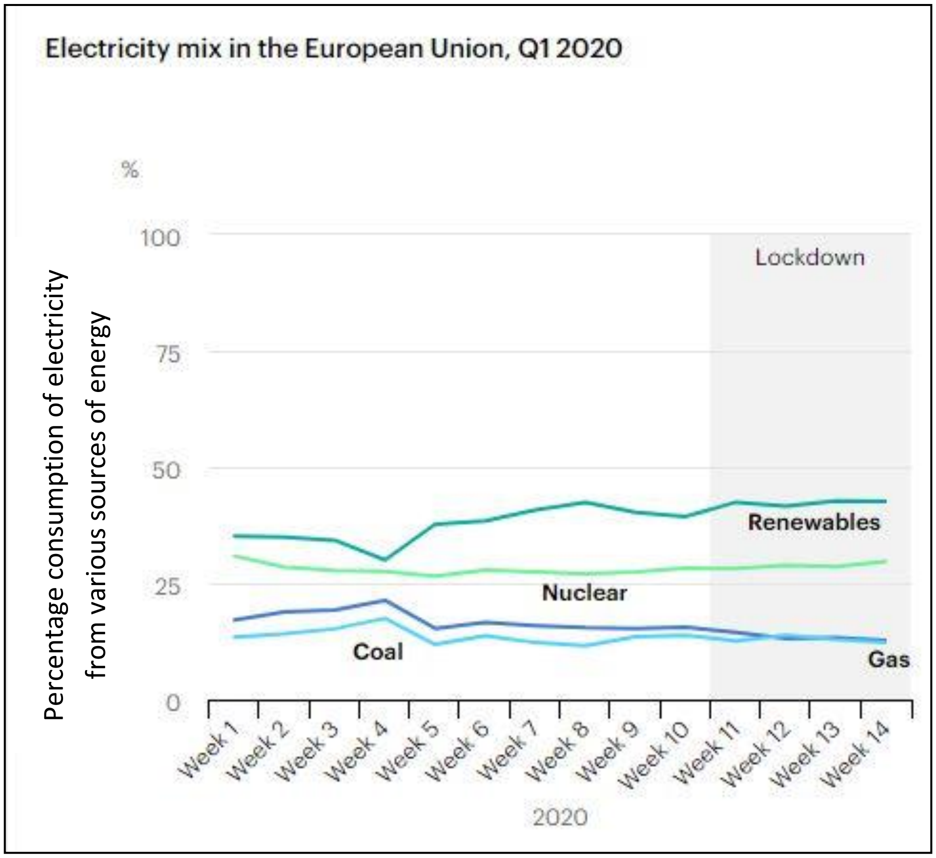

Figure 1.1(b) Power consumption from various energy resources in European Union for the first quarter of 2020 [3]

The case in United States is quite different from the others as the major source of power consumption is from gaseous fuels. The percentage consumption from nuclear power is somewhat constant in the first quarter of 2020 and the consumption from renewable resources is slightly increasing while there is a fairly decreasing trend in power consumption from coal especially during the lockdown period in the United States. Figure 1.1(c) shows the power consumption from various resources in United States.

From the figures 1.1(a), 1.1(b), 1.1(c) it can be clearly seen that the world is shifting towards the use of renewable sources of energy and though the demand from fuels like coal and gas is decreasing yet to completely replace the existing conventional fuel plants will take a lot of time and capital.

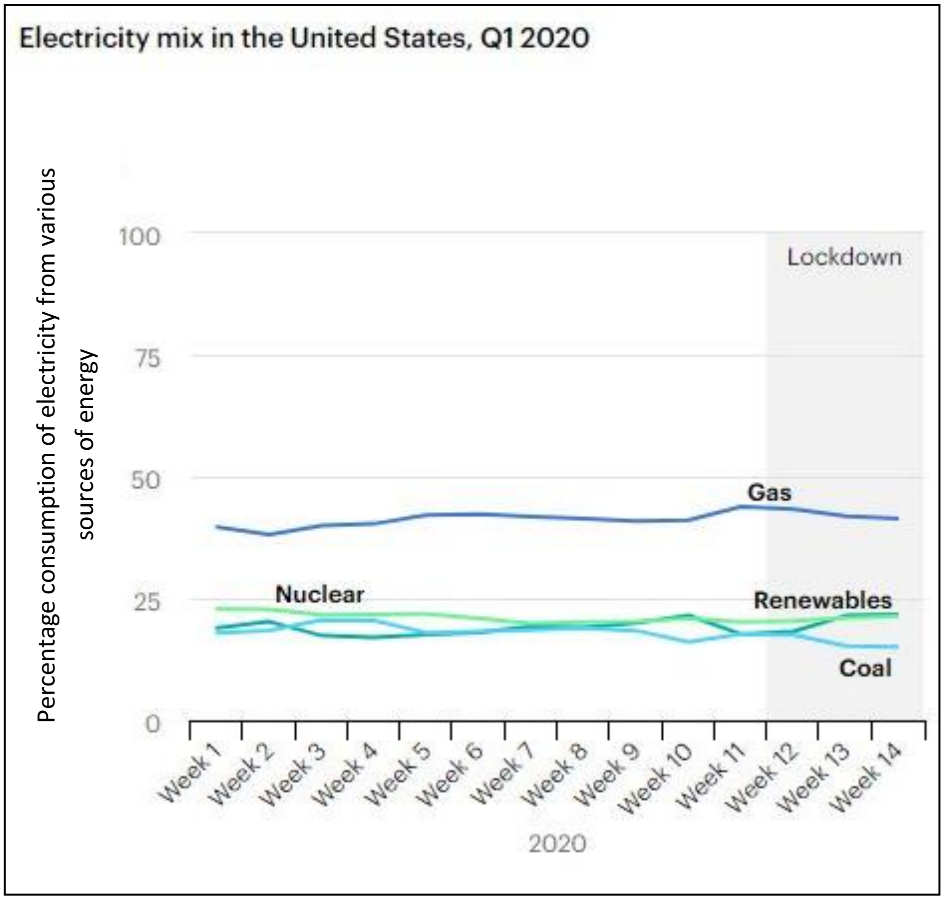

Figure 1.1(c) Power consumption from various energy resources in United States for the first quarter of 2020 [4]

Due to lockdown measures electricity demand has been significantly affected, impacting the power mix greatly [1]. Only the renewable resources can be witnessed to follow an increasing trend in demand [1].

Though the demand of residential electricity has increased yet it is unable to compensate the decline of industrial and commercial demand for electricity. During lockdown period, some countries witnessed the demand to decrease by even 20% or more [1].

1.2 CURRENT ENERGY SCENARIO IN INDIA

Due to the abundantly increasing population, the demand for electricity is increasing day by day. In terms of population India is only second to china and so it becomes quite difficult to fulfill the country's power requirements. But India also has the vast variety of resources for its electricity production. From hydro based power plant to coal and natural gas based thermal power plants, India tends to utilize every available resource effectively. Not only this, India also has great potential for utilizing solar and wind power too. Figure 1.2(a) shows the current percentage of electricity consumption in India from various resources. It is evident from the figure that the major portion of electricity generation in the country is derived from coal based plants which contributes to nearly 70% of the net electricity generation. Though, the power consumption from other resources viz. renewable, nuclear, gas and oil has increased a bit during the first quarter of 2020 but still there is dominance of coal based power plants in India.

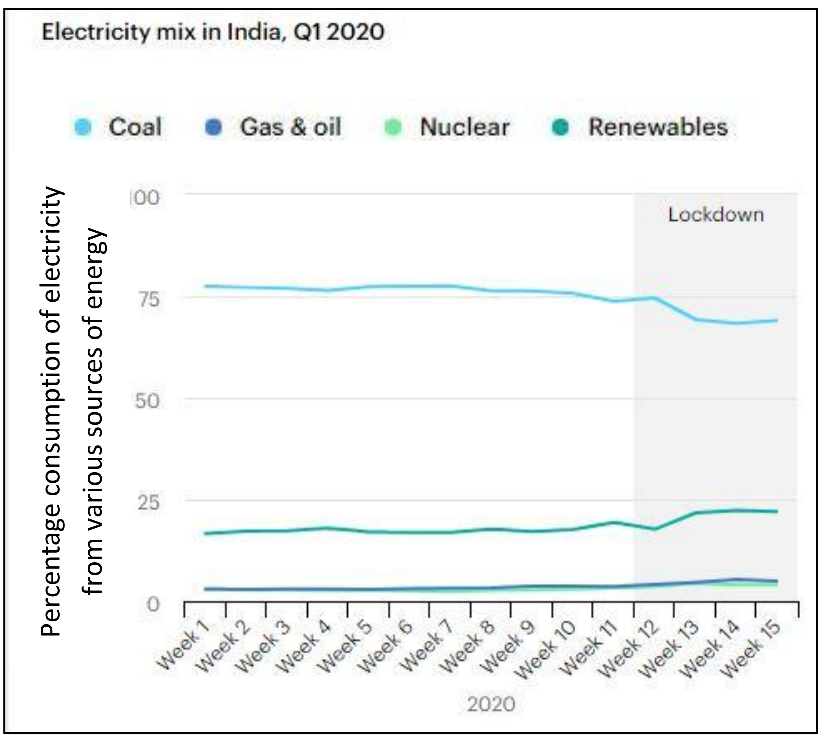

Figure 1.2(a) Power consumption from various energy resources in India for the first quarter of 2020 [5]

The installed capacity of India from various resources as on till 31.03.2020 was nearly 370.1 GW (refer figure 1.2(b)). The percentage wise breakup of the installed capacity* is given below in figure 1.2(b). It becomes evident that major share of the power production is from thermal power plants which 62.8% of the total followed by variety of renewable energy resources (RES) and their combined share stands at 23.5% which is followed by the share of hydro power plants at 12.4% and production of electricity still from nuclear power plants still lacks at about 1.9%.

Fuel	MW	% of Total
Total Thermal	2,30,600	62.8%
Coal	1,98,525	54.2%
Lignite	6,610	1.7%
Gas	24,937	6.9%
Diesel	510	0.1%
Hydro (Renewable)	45,699	12.4%
Nuclear	6,780	1.9%
RES* (MNRE)	87,028	23.5%
Total	**370,106**	

*Installed capacity in respect of RES (MNRE) as on 31.03.2020

RES (Renewable energy resources) include Small Hydro Project, Biomass Gasifier, Biomass Power, Urban & Industrial Waste Power, Solar and Wind Energy [6]

Figure 1.2 (b) Installed capacities from various energy resources in India till 31.03.2020 [6]

Establishing a power plant requires a lot of capital investment and time and even when the world is adopting new technologies and inclining towards the renewable energy resources it would be quite cumbersome to immediately shut the existing power plants and shift to other means of energy production, though it should be done but it will take time, especially in a country like India where the majority of power production takes place from thermal power plants.
So whilst the change is taking place efforts must also be made in the direction of augmenting the output from existing power cycles, increasing their efficiency, reducing the exhaust pollutants and also reducing the fuel consumption in the existing cycles. Steam injection in a GT cycle is one such technique that is employed for the same.

1.3 STEAM INJECTION IN A GAS TURBINE

For controlling NO_x emissions from a GT cycle, water injection was the technology used in the earlier days. It is not only an efficient and verified technique for decreasing NO_x but it also provides an augmentation in output from the cycle. But injecting water requires massive amount of highly pure feed water, and if the purity of this feed water is compromised even for a temporary period, the gas turbine is vulnerable enough to sustain huge damages [9].

To solve this problem steam injection is used, apart from the fact that before injecting, the water has to be boiled, it is somewhat same as injecting water [9].Due to vaporization of water, almost all of the pollutants present in it are retained, concentrated and removed from the boiler [9]. Steam of satisfactory quality is generated by the water in the high pressure boiler for the sole purpose of injection. Steam of such quality is appreciated for use in the industry [9].

STIG is a well-known principle to suppress the air pollutants (specifically the NOx emissions) resulting from the combustion products or to further the output of a GT cycle [9].

In a simple STIG cycle, the steam is directly injected into the combustion chamber of a gas turbine. Both, the injected steam and air after the compression process consume the energy from the fuel in the combustion chamber and pass through the GT and thus raising the flow rate(by mass) available at turbine inlet and hence the output of the GT.

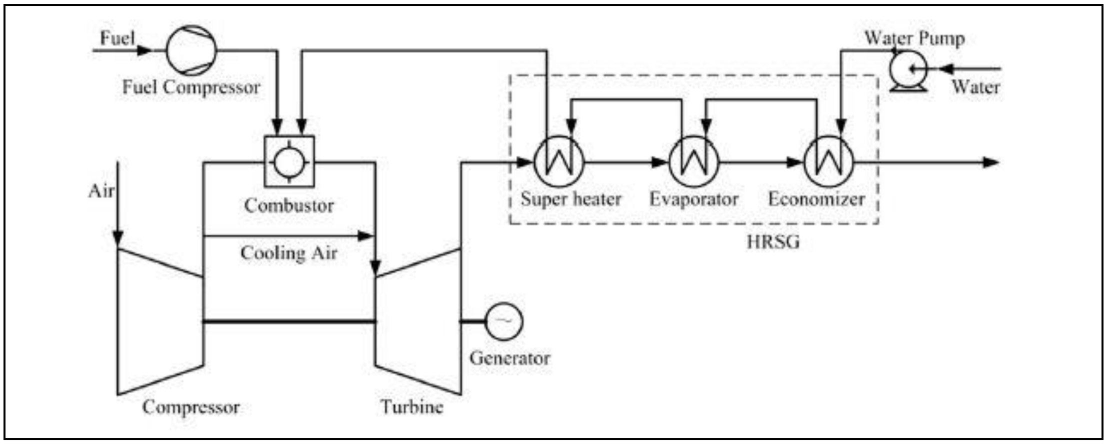

Figure 1.3 A Simple STIG Cycle [21]

1.4 MODIFICATIONS IN A SIMPLE STIG CYCLE

i) TURBO S.T.I.G.-THE TURBOCHARGED STEAM INJECTED GAS TURBINE CYCLE

The Turbo-STIG system makes the use of the steam turbine to power a topping air compressor in a gas turbine cycle to increase the pressure ratio of the gas turbine and thus increasing its efficiency [7].

The process of injecting steam into a GT cycle raises the output without raising the consumption of power in the compressors. To bring out the most efficient results, the output shaft must receive this raised power directly, which necessitates the use of a single shaft GT system because when the steam injection is performed in a multi-shaft GT system, then there is mismatch of speed and power of the compressors and turbines [7]. Because of such restrictions of multi-shaft GT systems the efficiency reduces and thus the concept of Turbo STIG has been restricted to single shaft GT systems. For maximum power generation efficiency, all of the steam which that is generated from the heat recovery must be used for injection. Also injection of the additional amount of steam generated by supplementary firing results in reduction of efficiency Thus using the Turbo STIG system increases the efficiency of the cycle when compared with simple

STIG cycle, by raising the pressure ratio of both the ST and GT while, maintaining the low capital cost of normal STIG systems. [7]

ii) STEAM REHEAT IN ADVANCED STEAM INJECTED GAS TURBINE CYCLES

The thermal efficiency of simple GT cycle is usually low due to the fact that a considerable amount of exhaust gas energy is lost which could have been utilized. By using STIG methodology higher values for the specific work output can be attained, but still the rise in thermal efficiency would be lower when measured relative to that of the combined cycle [14]. To integrate the benefits of the comparatively simpler STIG cycle with the combined cycle which has greater caliber of having higher thermal efficiency, STIG cycle was altered in following manner:

1. By considering the installation of another combustion chamber. The reheating of the gas at any point during its expansion obviously increases the outlet temperatures resulting in the exhaust gases to carry higher energy than before. The same effect is produced by this sequential combustion, so the benefits of this altered condition can be reaped too.

2. Another modification can be done by raising the pressure, of the steam generated. This raised pressure could be used to produce more power by employing back pressure ST.

The flow carried out through this back pressure ST brings the steam down to the pressure required for suitable injection, it also causes decline in steam temperature, which results in the steam entering the combustion chamber to be at relatively lower temperature, which increases the need of use additional amount of fuel.

The performance of this cycle can be further increased by first heating the steam in the boiler instead of leading it from the exit of back pressure steam turbine straightly into the GT cycle [14]. Nevertheless, total volume of the generated steam would be less due to the steam reheat, which also decreases the specific work performance [14].

iii) STEAM INJECTION GAS TURBINE WITH INLET AIR COOLING

A massive loss of heat energy occurs in the exhaust of the GT cycle. But this heat energy can be utilized for steam production. This generated steam serves a number of useful purposes (heating, drying, separation etc.), also it can be used back for enhancement of power generation capability and efficiency [13]. Methods like STIG and IAC are very effective in utilizing the generated steam for the same purposes [15].This generated steam by waste heat recovery usually surpasses the amount of steam required for steam injection [13].

When the atmospheric pressure rises and/or ambient temperature descends, the density of air increases, which creates a rise in flow rate of inlet air (by mass). With the resulting air mass flow, the net heat generated due to combustion increases and power available at the output shaft also increases [13]. Therefore the engine can produce higher power.

Decreased ambient temperature results in increasing the air density which thus results in an increased pressure ratio. But during the warmer months, there is a reduction in thermal efficiency and electrical capacity because of the increase in ambient temperature. So, using an IAC system at the compressor in a STIG cycle would enhance the thermal efficiency and capacity of power plant. [13] The reason is definitely the rise in flow rate (by mass) across the GT.

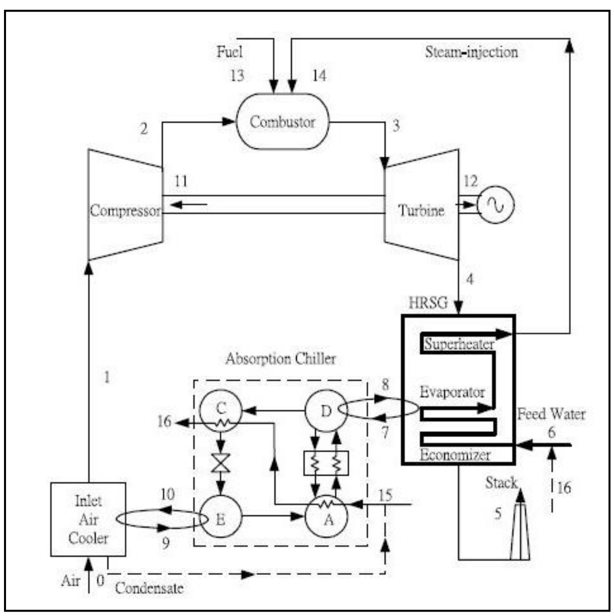

Figure 1.4.1 STIG with Inlet Air Cooling [15]

iv) REGENERATIVE STEAM- INJECTIONS GAS-TURBINE SYSTEMS

Not only can the thermal efficiency of the GT but that of regenerative gas turbine cycles also be improved with the help of steam injection [16]. Thus integrating the regenerative cycles with STIG is a good proposal. Mainly two types of regenerative STIG systems are available namely RSTIG-1(regenerative STIG-1) and RSTIG-2 (regenerative STIG-2) [16].

In the first configuration (refer fig. 1.4.2(a)) the regenerative cycle is integrated with STIG in such a manner that the total volume of generated steam after the HRSG is straightly supplied to the GT cycle. Resulting exhaust gas temperature for RSTIG-1 is lowered down to about 100° C [16].

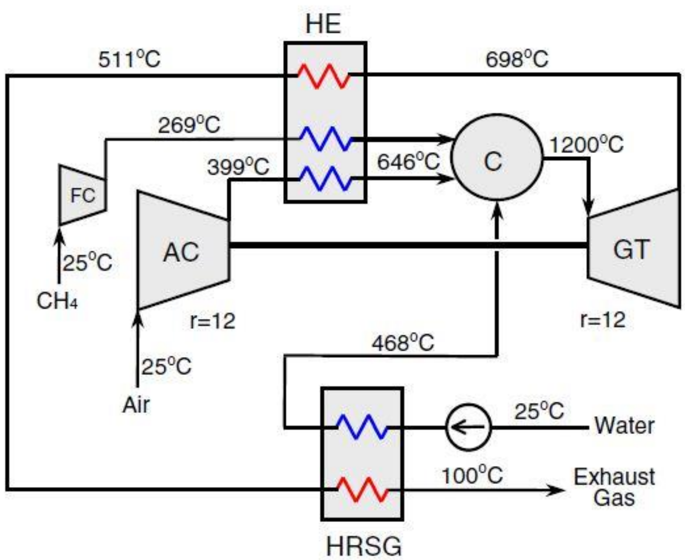

Figure 1.4.2(a) RSTIG-1 [16]

The configuration of RSTIG-2 (refer figure 1.4.2(b)) differs from that of RSTIG-1 system because the total volume of generated steam in HRSG is injected after compressor [16]. In both the configurations a heat exchanger has to be employed for regeneration and since exhaust heat recovery can be done at both the HRSG and heat exchanger by the water supplied to the HRSG, therefore the steam flow rate of the RSTIG-1 is higher than that of the RSTIG-2 [16].

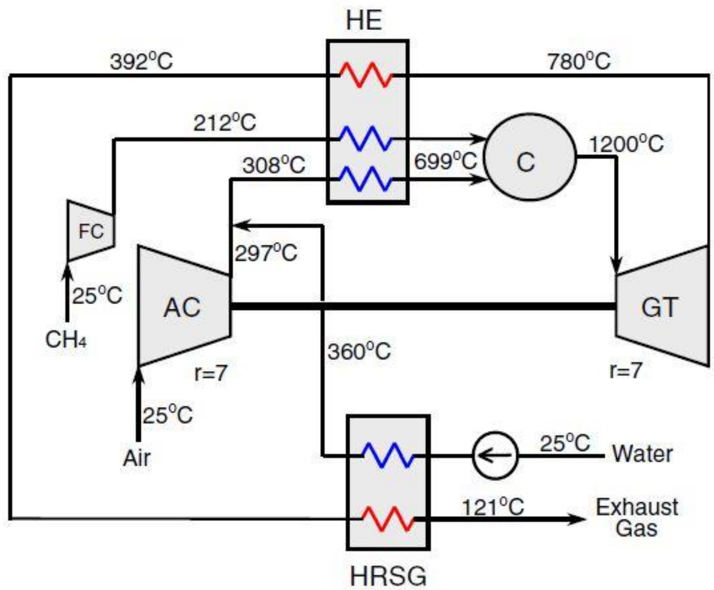

Figure 1.4.2(b) RSTIG-2 [16]

v) STIG BASED COMBINED CYCLE WITH DUAL PRESSURE HRSG

Steam expansion in GT is carried out up to the atmospheric pressure however steam exits at much lower pressures in a combined cycle plant, in turn generating greater output. Therefore, STIG system has a relatively lesser efficiency than that in combined cycle system. The output of the steam cycle declines when steam injection is used in the combined cycle however it causes a rise in the GT output. But by carefully selecting the parameters so that the loss in ST output is compensated by gain in GT cycle output, the overall output for the combined cycle is raised with the technique of steam injection [17].

vi) SOLAR HYBRID STEAM INJECTION GAS TURBINE (STIG) CYCLE

In the simple STIG arrangement the steam flow rate varies directly with the extent of wasteful heat energy available from the exhaust of GT at the HRSG and thermodynamically it is feasible to increase the amount of injected steam to the cycle but it requires an extra heat source. Conventionally, production of steam is from the thermal energy of the exhaust gases, utilizing solar energy as another heat source results in generation of more steam [23]. This additional heat required for steam generation can be produced with the solar energy using concentrating collectors. This results in a solar hybrid STIG cycle, which takes heat input from fuel in the combustor as well as from solar collectors [18]. Solar STIG cycle hybridization leads to solar power intermittency and economic issues being solved [20].

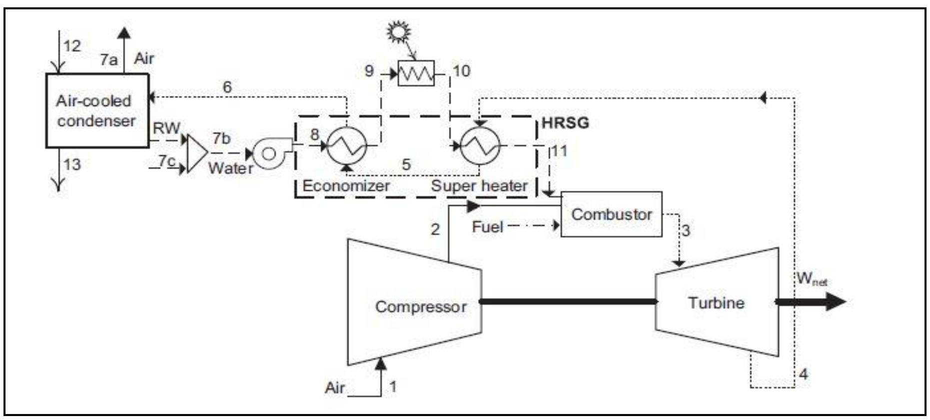

Figure 1.4.3 Layout of a solar hybrid STIG [19]

vii) PARTIAL OXIDATION STEAM INJECTED GAS TURBINE CYCLE

In the Partial Oxidation Gas Turbine (POGT), the conventional combustion chamber is replaced by a partial oxidation reactor, in which the air quality is lower than the stoichiometric value needed for complete combustion of fuel. A synthesis gas consisting of primarily carbon and hydrogen is produced by this partial oxidation reactor which has rather greater specific heat than the gases produced during total fuel combustion [21].

Unlike STIG cycle, in POGT cycle the steam is injected in the partial oxidizer and mixer of the topping gas turbine cycle [21]. Also compressor of bottoming GT cycle raises the pressure of the ambient air, a part of which is then sent for an extra raise in pressure nearly 6 MPa and this is part which supposedly enters the partial oxidizer [21].

Under the conditions of high temperature and pressure, a secondary fuel gas is generated in the partial oxidizer, when the fuel reacts with air and the injected steam. Steam is mixed with this secondary fuel gas and both are allowed to enter the expander. After exiting from the expander the secondary fuel gas enters the combustor of the bottoming GT cycle, turbine cooling is not considered in the topping cycle [21]. Figure 1.4.4 shows a schematic diagram of POSTIG cycle.

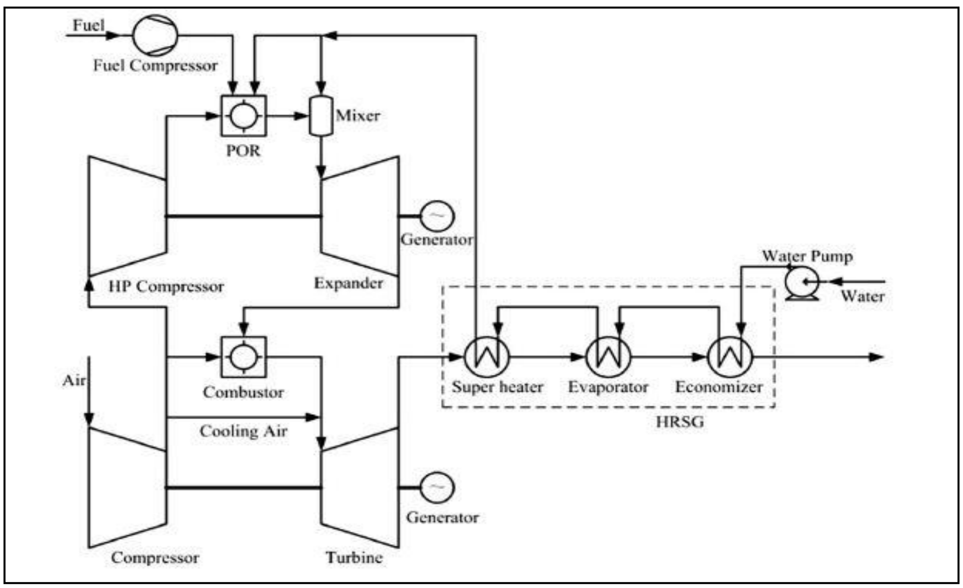

Figure 1.4.4 Schematic Diagram of POSTIG [21]

CHAPTER 2

LITERATURE REVIEW

2.1 SUMMARY OF PAST WORK

In around 1949 **Cheng** invented six patents in relation to application of STIG methodology [7]. But it is only from about 1960 that the technique of steam injection is being used to increase the output of industrial gas turbines [7].

Around 1965, papers discussing steam injection started trending [7].

Foster-Pegg (1989) [7] analyzed the turbocharged STIG cycles and stated that the turbocharged STIG removes the problem of the surge margin by making the use of a topping compressor. He also concluded that the turbo STIG removes most of the problems of simple STIG as:

1) Its efficiency is comparable to that of combined cycle while that of STIG is lesser
2) It requires lesser modification of already installed engine
3) Due to lower capital cost and somewhat equal heat rate when compared with other combined cycles, the cost of power output obtained from the Turbo-STIG is lower [7].

There is a great loss of water and energy available from the exhaust steam, to recover this loss **Nguyen** and **Otter** [8] portrayed and examined a "closed loop" water recovery cycle with steam injection that was created for STIG applications. Their procedure was expected to help in areas where treatment of water can be quite complicated, squandering of water cannot be afforded. The system described in the study helps in decreasing the exhaust gas temperature to a specific temperature, and it is therefore it becomes reasonable to regain all of the injected steam. To cool off the exhaust gases any cooling medium can be used, be it the ambient air or be it the large water bodies. The Study [8] asserted that the main factors affecting the size and design of the water recovery system are the flow rates(by mass) of air, fuel rather than the volume of steam injection, or size, the power rating of the turbine.

They [8] concluded that to recover water more effective results are produced in case of gas turbines having higher ratio of steam injection or for the units that have high specific output. Also they asserted that the boiler described in their study can produce steam much more than is generally necessary for the purpose of injecting steam. And the extra steam is further utilized for alternative applications too like the cogeneration and thus will help in reducing the emissions from exhaust gases, enhancing the work output and at various loading and ambient conditions provide flexible operation [8].

Rice [9] realized the potential of steam injection for both power augmentation and cooling and around 1993 presented a series of 3 papers for analyzing the STIG cycle. He compared the increase in power by varying the amount of injected steam. To calculate the increase in output he divided the flow rate(by mass) of injection by steam rate for a given TIT [9]. He found the steam rates for various STIG cycles including STIG integrated with reheat and intercooling with and without a topping steam turbine cycle. Though he suggested that increasing the TIT will help in increasing the power output yet he did not take some considerations into account. Parameters including requirements of nozzle-area, changes in expansion efficiency, variation in compressor pressure-ratio and in compressor efficiency, changing air flow rates, blade-cooling requirements, increases in loading for rotating blade were among such considerations [9].

He also compared the cycle efficiency for a fixed amount of injected steam, whose temperature was increased and was expanded in a gas turbine to that of this steam when it flowed through a condensing steam turbine [10]. The steam rates for cycles having larger TIT and greater pressure ratio of GT were found to be significantly lesser than that for the condensing steam turbine [9]. His results also proved that the STIG cycle with topping steam cycle and with reheat requires the least steam rate at a given TIT [9]. Steam cycle efficiency becomes more important parameter when the cycle is evaluated for a fixed amount of injected steam [10]. He suggested that even if the flow of air is dropped by a very tiny proportion, compressor efficiency can still be improved for a given nozzle area provided that the ambient inlet air temperatures are high [9].

To optimize the STIG, certain thermodynamic considerations must be accounted. Keeping this in mind **Noymer** and **Wilson** [11] concentrated their study on the thermodynamic factors effecting STIG cycles. Such factors may include the state of steam injection i.e. the temperature and the pressure, the amount and rate of injected steam and the locations where the steam should be injected [11].

The authors [11] studied effects of these factors on a normalized basis on a engine called the dual spool turbo shaft engine, and TIT was maintained keeping in mind the design point of a simple cycle [11]. Each of these factors affects the capacity of steam generation. The volume of generated steam is the upper limit for the system's possible set of design points [11].

Economics also influences the selection of the design point. It is obvious that however advantageous the steam injection may be, it must initially top the total expense occurred during the installation and the amount of resources consumed.

The device will perform better if the difference between temperature of the steam and that of the exhaust gases is less, but the size and cost of the heat exchanger are inversely proportional to this temperature difference, so if this difference is large, then both the size and cost will increase [11].

They authors [11] asserted that though the mentioned factors are important yet in the end it is the needs of the end user which usually facilitates design criteria for which less amount of steam is required. The structural integrity of the engine may also restrict the amount of steam injection [11]. They also asserted that variable steam temperatures will make the steam injection more complex and may also require altering the gas turbine structure mainly the casings and bearings for improving the endurance capability and these considerations should be made the part of initial economic analysis of the installation cost [11].

Bartolini et al. [12] developed a code or computerized model for evaluating the off design performance of the STIG turbines. Various factors were taken into account including the effect of the cooling process on the power output and also the real fluid dynamic conditions required for turbine operation. The code proved to be a valuable tool for the prediction of off design characteristics of STIG plants, and it could also be used for the analysis of performance of plants which are already in existence so as to analyze the practicality of different modes of operating the plants [12].

To further increase the performance of STIG cycle, **Bartolini** and **Salvi** [13] explored the possibility of cooling down the air at compressor inlet, by utilizing the energy which was not utilized to generate the maximum amount of steam that could have been injected. Their study

[13] presented the techno-economical benefits of cooling down the inlet air by incorporating an absorption chiller in a STIG cycle. The authors [13] asserted that the modification proposed in the study made it possible to boost the turbine's output, which resulted in optimized recovery of wasteful heat in the plants with cogeneration and also giving district heating. They concluded that highest performance levels can be achieved in these plants whilst generating:

1) Maximized volume of steam for steam injection.
2) District heating from steam or hot water.
3) Chilled water for cooling down inlet air into the compressor

Hofstadter et al. [14] compared the performance of advanced STIG (A-STIG) cycles with the combined cycle power plants. For this purpose, the STIG cycle was altered in following manner:

1. By considering the installation of another combustion chamber. The reheating of the gas at any point during its expansion obviously increases the outlet temperatures resulting in the exhaust gases to carry higher energy than before. The same effect is produced by this sequential combustion, so the benefits of this altered condition can be reaped too.
2. Another modification can be done with a raise in the pressure of generated steam. The increased pressure is converted into extra output with employment of a back pressure ST.

He concluded that since the combined cycle has a thermal efficiency of about 58%, it is justified that it has a dominating position in base load power generation. Nonetheless, it may be worth considering an A-STIG cycle with significantly lesser installation costs and relatively high specific work performance but at a low thermal efficiency, for designing a power plant with peak or medium load demands.

Wang and Chiou [15] based their research on the fact that for a simple GT cycle the exhaust gases commonly have certain high temperatures of the magnitude of around 500° C, and though there are many techniques proposed for converting the otherwise lost potential of the exhaust gases, they asserted, integrating STIG with IAC(inlet air cooling) will provide the best results because utilization of waste heat recovery for generation of steam, which in turn, provides power to an absorption chiller is inherently compatible with Steam injection[15]. In their study [15], a simple power generation cycle was considered and its performance was compared with that of simple cycle integrated with STIG and simple cycle integrated with both the STIG and IAC. The required steam is produced as an outcome of conversion of otherwise lost potential of exhaust gases into beneficial energy of the steam. They concluded that advantages of incorporating STIG can significantly increase the output the efficiency of power generation [15].

In a study presented by **Nishida et al.** [16] The property of STIG to further the outputs of simple and even regenerative GT was analyzed in the form of two regenerative STIG systems namely RSTIG-1 and RSTIG-2. He inspected the proposed cycles in terms of their specific power output and their thermal efficiencies. Also, these proposed configurations were matched with various other configurations like water injected, regenerative, and STIG integrated systems on account of their performances in exergy analysis. They concluded that:

1. The Regenerative STIG system performs better than simple regenerative system in terms of specific power output.
2. RSTIG is supposed to be relatively better than water injection when matched with in terms of first law efficiency.
3. To obtain maximum efficiency, the optimum pressure ratio of STIG and RSTIG-1 is higher than that of RSTIG-2. Also, the performance of RSTIG system can be further enhanced by the injecting steam just after the compressor.

Srinivas et al. [17] aimed at finding the combined cycle system variables which improve net exergy of the system when integrated with technology of steam injection. For this reason, he conducted a sensitivity analysis with effect of fuel mass ratio, steam injection, temperature ratio of the deaerator, steam reheat pressure ratio, inlet pressure of the steam turbine, pressure ratio of the gas cycle and temperature of the combustion chamber on exergy output of the combined cycle.

Livshits and Kribus [18] evaluated the utility of a solar hybrid STIG cycle within optimal circumstances to find solutions for higher overall conversion efficiency of heat energy into electricity, along with medium to high Solar Fraction (up to 50 percent), they also compared performance of the cycle in terms of combustion power, heat supplied, turbine work, compressor work, cycle capacity, specific work output, pump work, solar fraction and steam air ratio [18].

Polonsky et al. [19] performed an evaluative study for validating the effectiveness of the solar STIG hybrid cycle on account of its annual performance at various sites across the globe. They concluded that the climatic factors majorly determine the performance of solar STIG hybrid cycle at each location and that further technical and economical study can be performed on this topic.

To evaluate the economical feasibility of solar hybrid STIG plants **Selwynraj et al.** [20] researched the cycle in various power generation scenarios for the cities of Jaipur and Indore. He concluded that for the plant to profitable economically the key factor is the market price of electricity.

Jiezhang et al. [21] compared the POSTIG cycle with STIG on the basis of their performance by taking the parameters such as specific work output, and efficiency, concluded that POSTIG cycle is somewhat better than the STIG cycle in terms of the exergetic efficiency.

Solar hybrid STIG cycle largely depends on the solar power, so it is mainly efficient during daytime but at night it is not that effective, to counter this problem **Polonsky and Kribus** [22] investigated the possibility of storing the heat energy in a latent phase changing material.

Selwynraj et al. [23] evaluated the annual performance and environmental impact of solar hybrid STIG cycle for different parts of India namely Jaipur and Indore. The study showed that that CO_2 emissions were lesser than that of simple gas turbine cycles but were nearly same as those of the combined cycle.

Kalathakis et al. [24] assessed various solar hybrid cycle integrated with to determine the most effective output from basic STIG cycle. Of all the different types of configuration examined in the study, the recuperated engines proved to be best in performance with respect to fuel reduction which is about 25% at the cost of reduction in energy output which was about 2.5%. It was concluded in the results that a STIG cycle with solar input shows better results when the energy from the sun is utilized in the evaporator rather than the superheater.

In the study presented by **Sanaye et al.** [25], a combined cooling, heating, power and water (CCHPW) plant integrated with IAC and steam injection was evaluated on basis of energy, exergy, environmental impact and economic feasibility. Apart from STIG arrangement the absorption refrigeration system was utilized for inlet air cooling. The system was optimized using the genetic algorithm.

CHAPTER 3

RESEARCH GAP AND OBJECTIVE

3.1 RESEARCH GAP

Various modifications have been carried out for the development of simple STIG cycles varying from integrating it with regenerative or reheat cycles to including the concept of solar hybrid STIG. The recent developments include the integration of IAC to a combined steam and gas cycle generating power along with incorporation of steam injection while also producing district heating and cooling **.**

But there is still scope of study in this field because not much research has been done on partial steam injection with combined gas and steam cycle and by utilizing solar hybrid STIG with it and with every other possible configuration.

3.2 OBJECTIVE

i) To vary the different input parameters for a combined gas and steam cycle with partial steam injection and compare its sfc, net work output and efficiency with those for a simple GT, combined gas and steam cycle and conventional STIG cycle.

ii) To integrate solar hybrid STIG with the proposed cycle and see its effect on various output parameters.

CHAPTER 4

THERMODYNAMIC MODELLING OF THE SYSTEM

4.1 INTRODUCTION

STIG technology has proved to be efficient in increasing power and reducing emissions, but still continuous efforts are being made to integrate the methodology into various configurations. To investigate one such prospect and for the further study of STIG, one of configurations discussed by **Livshits and Kribus** [18] and the study presented by **Sanaye et al.** [25] are used as a reference to individually validate the solar hybrid STIG and combined cycle respectively and then initially the parametric analysis of combined gas and steam cycle with partial steam injection is carried out and then solar input is added to it to see effect of the two reference configurations combined together.

4.2 SYSTEM DESCRIPTION

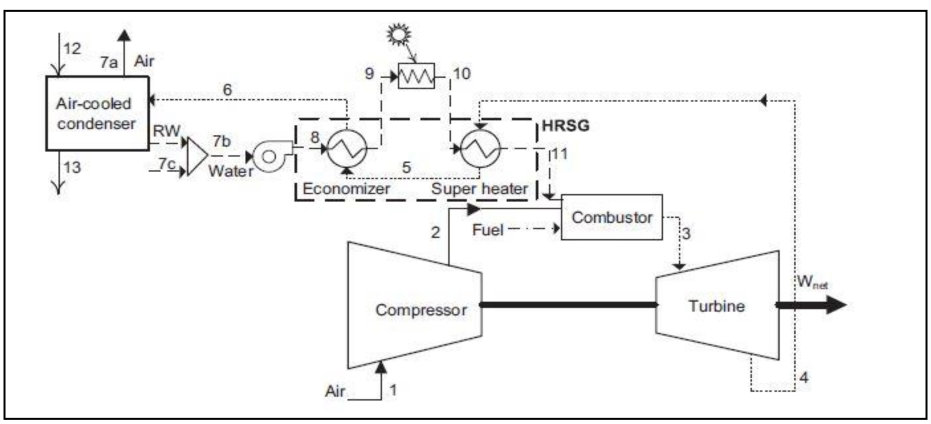

Figure 4.2.1(a) Layout of a solar hybrid STIG (reference cycle for solar hybrid) [18]

The reference cycle has taken from the study of **Livshits and Kribus** [18]. This cycle is shown in figure 4.2.1(a). There are various configurations of solar hybrid STIG and analysis of each configuration is necessary but for the sake of this work the reference configuration used is one in which solar energy is utilized in HRSG, the other possible configuration would be to utilize the solar heat at the combustion chamber. For generating an appreciable amount of heat the energy from the sun must be of high temperatures in order to do so solar receiver are used which concentrates the solar energy and thus increases the amount of heat. Even for utilizing the solar heat at the HRSG there are two possible configurations, which are as following:

a) To place the solar receiver with the existing evaporator in parallel and then adding the steam generated in the HRSG evaporator to the saturated steam from the solar input [24].

b) To place the solar receiver so as to produce the superheated steam and then adding it to the superheated steam generated in the HRSG [24].

In the reference cycle a single shaft GT cycle is employed, and then an HRSG. This design is used for the conventional STIG. So, to make the design solar hybrid, the solar part, consisting of a solar collectors used for utilizing the input from sun to evaporate the water is added which increases SAR. Water enters the economizer which heats water to a temperature which is different from the saturation temperature (generally lower than it) and this difference is the approach temperature. Then the water enters the evaporator and where the conversion of water into saturated steam takes place which is then allowed to enter the superheater thus the resulting steam is in superheated state. Finally, this steam is allowed enter the combustion chamber GT cycle for the purpose of steam injection. The exhaust gases do not transmit their energy to the evaporator rather it only utilizes solar energy as shown in figure 4.2.1(a).

Another cycle has been taken from **sanaye et al. [25]** for validating the performance of combined cycle. This cycle consisted of dual pressure HRSG with lower and higher pressure drums and also an absorption chiller. Along with STIG inlet air cooling is also employed in this cycle. But for the sake of validation only the data for combined gas and steam cycle has been taken from this study. The schematic diagram for this study has been depicted in figure 4.2.1 (b).

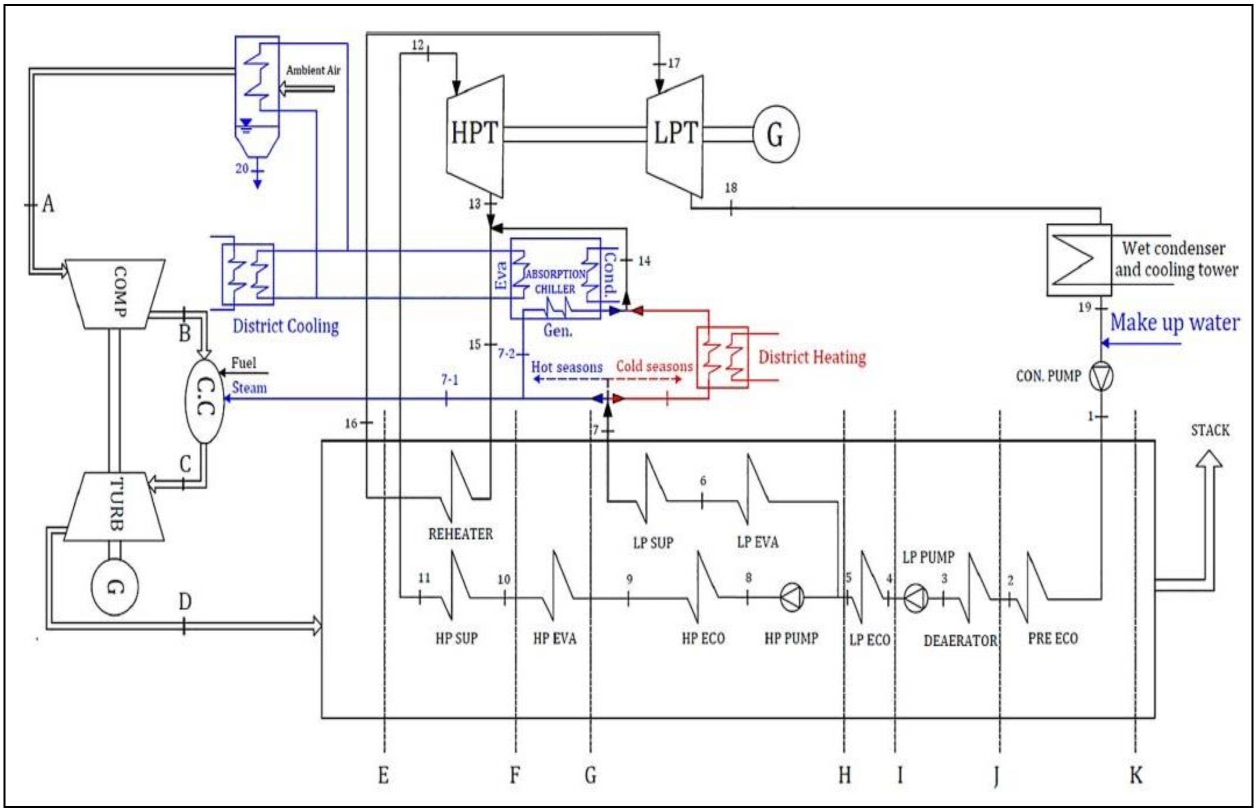

Figure 4.2.1(b) Reference cycle combined gas and steam cycle [25]

For avoiding the risk of ice formation, the lowest temperature of inlet air temperature was considered as 15 °C [25]. For designing the HRSG, the approach temperature and the pinch point temperature difference was considered as 15 °C & 10 °C respectively. The gas TIT was kept constant (1060°C) and the injection temperature of the steam was taken as 320°C [25]. A part of steam was expanded in two ST(one HP and one LP). The combined output of the GT and ST was subtracted from the power input to the compressor and the result was considered as the net work output for the study [25].

For the work in this project the reference cycle [25] is modified by removing the absorption chiller and inlet air cooling from the cycle and considering a single pressure HRSG. Also instead of two, a single turbine is considered. Thus it has become a combined gas and steam cycle with partial steam injection. This proposed cycle is shown in figure 4.2.2(a). When integrated with solar input the proposed cycle is shown in figure 4.2.2(b). Further analysis of this work has been carried to compare the efficiency, specific fuel consumption and net power output of the proposed cycle with those of simple GT cycle, conventional STIG cycle, combined gas and steam cycle.

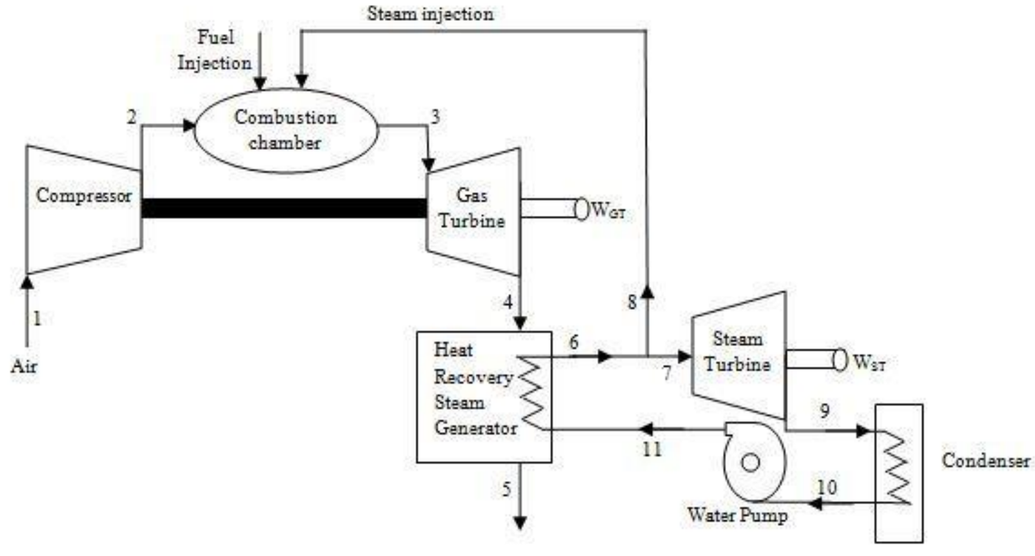

Figure 4.2.2(a) Combined steam and gas turbine cycle with partial steam injection (proposed cycle)

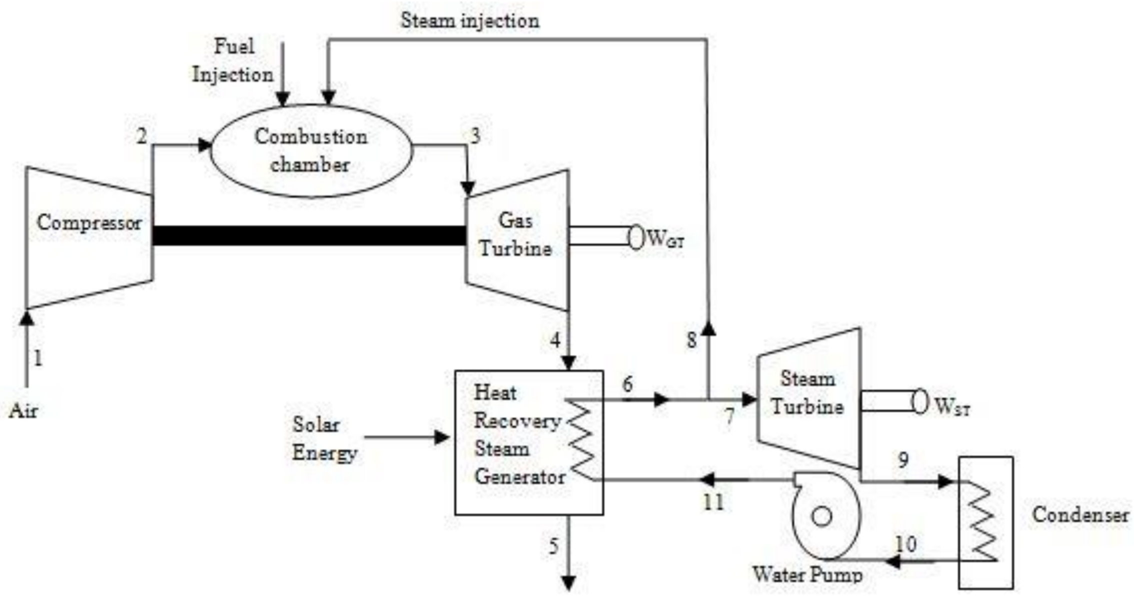

Figure 4.2.2(b) Proposed cycle with solar input

4.3 SYSTEM ASSUMPTIONS:

The assumptions made for modeling and analyzing of all the systems under consideration are as follows:

1. All of the processes in the proposed cycle are steady state processes.
2. The compressor and the turbines (both gas and steam) are working adiabatically.
3. Lower heating value of the fuel (methane) is 50000 kJ/kg [25].
4. Inlet air and exhaust gases are considered to be ideal gases so that principles of ideal gas mixtures can be used.
5. The pressure drops at combustion chamber, HRSG and pipes are 2% and 3%, 3% respectively [25].
6. HRSG is single pressure and outlet state the after HRSG is saturated steam.
7. To avoid the formation of corrosive acid and water vapor condensation in the stack the temperature of exhaust gases in HRSG must be >120 °C [25].
8. Potential and kinetic energy and exergy are ignored.

4.4 THERMODYNAMIC MODELLING

Taking the above assumptions into account the thermodynamic energy analysis of the proposed cycle is given below while only the exergy balance equations of each component is provided, all the equations are with reference to figure 4.2.2(a) and figure 4.2.2(b):

4.4.1 ENERGY ANALYSIS

The energy analysis of various components is given as below:

a) **COMPRESSOR**: Air at ambient temperature is compressed with the help of the compressor as depicted in figure 4.2.2(a). Stage 1 is the inlet air at ambient conditions and stage 2 is air after compression. The mass flow rate is constant during the compression process. This process 1-2 raises the pressure of the inlet air and thus raises the temperature too. Equations 4.4.1 and 4.4.2 represents the calculation of temperature at the outlet of compressor and energy balance for calculation of work input required by the compressor respectively.

$$T_2 = T_1 * \left[1 + \left\{\left(\frac{1}{\eta_{C_S}}\right) * \left(r_C^{\left(\frac{\gamma_{air}-1}{\gamma_{air}}\right)} - 1\right)\right\}\right] \quad (4.4.1)$$

$$\dot{W}_C = \dot{m}_{air} * (h_2 - h_1) \quad (4.4.2)$$

b) **COMBUSTION CHAMBER**: Air after compression is further heated inside the combustion chamber with added energy of fuel after combustion. A part of steam is also added to the combustion chamber while the additional of fuel is taking place and being mixed with the air from the discharge of the compressor. Thus it is there is addition of heat at this point and the increased heat from point 2 to 3 is considered to be input for the proposed cycle. Afterwards products of combustion i.e. the flue gases and the injected steam are considered as the energy carrier and carries heat from point 3 onwards. Equation 4.4.3 shows balance of energy from point 2-3 (with reference to figure 4.2.2(a).

$$\dot{m}_f * \left(LHV_{CH_4}\right) + \dot{m}_{air} * h_2 + \dot{m}_{steam_{inj}} * h_{steam_{inj}} = \dot{m}_{gas} * cp_{gas} * T_3 \quad (4.4.3)$$

$$\dot{m}_f + \dot{m}_{air} + \dot{m}_{steam_{inj}} = \dot{m}_{gas} \quad (4.4.4)$$

c) **GAS TURBINE**: The GT is used to generate power output as per the requirements of the end user, also the power developed is due to the expansion of exhaust gases from combustion chamber, the product of flow rate (by mass) at state point 3 and the change in enthalpy across the gas turbine from stage 3 to stage 4 gives the power output delivered by GT. The power output from the GT is also used to provide power to the compressor, so the net amount of work obtained from the gas turbine becomes less. The flue gases from the GT at point 4 are further sent to HRSG. The temperature relation for the gas turbine is given by equation 4.4.5 and the work output from it can be taken from equation 4.4.6.

$$T_4 = T_3 * \left[1 - \left\{ \eta_{GT_S} * \left(1 - r_{GT}^{\left(\frac{1-\gamma_{gas}}{\gamma_{gas}}\right)} \right) \right\} \right] \quad (4.4.5)$$

$$\dot{W}_{GT} = \dot{m}_{steam_{inj}} * (h_3 - h_4) + \dot{m}_{gas} * cp_{gas} * (T_3 - T_4) \quad (4.4.6)$$

d) **HEAT RECOVERY STEAM GENERATOR (HRSG)**: The HRSG considered for the proposed cycle is a single pressure HRSG consisting of a superheater, evaporator and an economizer. The heat from the exhaust gases after the gas turbine at point 4 is transferred to the water coming after the feed water pump at point 11. This exhaust heat is utilized to heat up the water in the economizer to point 11a which is then passed to evaporator to convert the water to steam at point 6a.The steam coming out of the evaporator is saturated steam and then at superheater the enthalpy of steam is increased. This result in decrease in the enthalpy of exhaust gases which are further discharged to the atmosphere at a lower temperature at point 5. The enthalpy at point 6 is increased due to the heat consumed from the HRSG this steam further partially injected into the combustion chamber and rest of it is allowed to expand through the steam turbine. The detailed sections of HRSG are shown in figure 4.4.1. Equations (4.4.7), (4.4.10), (4.4.11) are respectively energy balance equations at economizer, evaporator and super heater respectively. In case of solar input, only the evaporator receives the sun’s energy and the energy of the exhaust gases is utilized only the superheater and economizer, evaporator does not receive energy from exhaust gases(for the proposed cycle).

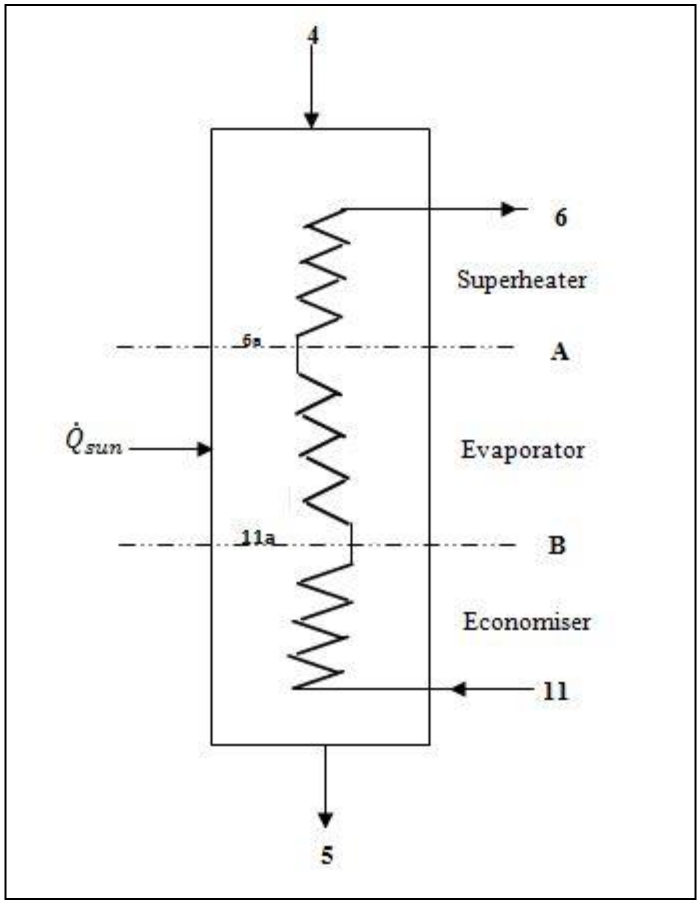

Figure 4.4.1 Detailed sections of HRSG with reference to figure 4.2.2(b) [solar input is only at evaporator]

$$\dot{m}_{gas} * cp_{gas} * (T_B - T_5) = \dot{m}_{11} * cp_{water} * (T_{11a} - T_{11}) \quad (4.4.7)$$

$$T_{sat_{steam}} - T_{11a} = \text{Approach Temperature Difference}$$

(4.4.8)

$$T_A - T_{sat_{steam}} = \text{Pinch Point Difference} \quad (4.4.9)$$

$$\dot{m}_{gas} * cp_{gas} * (T_A - T_B) = \dot{m}_{11} * (h_{6a} - h_{11a}) \quad (4.4.10)$$

$$\dot{m}_{gas} * cp_{gas} * (T_4 - T_A) + \dot{Q}_{sun} = \dot{m}_{11} * (h_6 - h_{6a}) \quad (4.4.11)$$

e) **STEAM TURBINE**: The amount of injected steam at state point 7 is passed through the ST to generate the power output. The output from ST is equal to product of flow rate (by mass) across the ST at point 7 and the enthalpy change form point 7 to point 9. The exit products of steam turbine are further allowed to pass through a condenser. To relation for isentropic efficiency and the work done by the steam turbine are given by equations (4.4.12) and (4.4.13) respectively.

$$\eta_{ST_S} = \frac{h_7 - h_9}{h_7 - h_{9s}} \tag{4.4.12}$$

$$\dot{W}_{ST} = \dot{m}_7 * (h_7 - h_9) \tag{4.4.13}$$

f) **CONDENSOR**: An air cooled condenser is employed for condensing the steam at point 9 from the exit of steam turbine and then converting it to water at point 10.

$$Q_{rej} = \dot{m}_7 * (h_9 - h_{10}) \tag{4.4.14}$$

g) **WATER PUMP**: The water pump is employed to further raise the pressure of the water after condenser to point 11. But this requires some work to be done for pumping the water. Thus it will result in decrease in net work output. Also make up water is added to pump to maintain the mass flow rate. The equation for the energy required by the water pump is given below as (4.4.14).

$$\dot{W}_P = \dot{m}_{11} * (h_{11} - h_{10}) \tag{4.4.15}$$

Under the steady state conditions for the evaluation of energy of the proposed system the steady state steady flow energy equation becomes:

$$\dot{Q} + \sum \dot{m}h = \dot{W}_{CV} \qquad (4.4.16)$$

And the mass balance at any state is considered to follow law of conservation of mass as:

$$\sum \dot{m}_{inlet} = \sum \dot{m}_{outlet} \qquad (4.4.17)$$

The summary of the equations for the energy balance for various main components present in the proposed system is given in table1.

Table 1: Energy balance equation summary for the proposed cycle
(Source: Reference [25])

Component	Energy Balance Equation
Compressor	$\dot{W}_C = \dot{m}_{air} * (h_2 - h_1)$
Combustion Chamber	$\dot{m}_f * (LHV_{CH_4}) + \dot{m}_{air} * h_2 + \dot{m}_{steam_{inj}} * h_{steam_{inj}} = \dot{m}_{gas} * cp_{gas} * T_3$
Gas Turbine	$\dot{W}_{GT} = \dot{m}_{steam_{inj}} * (h_3 - h_4) + \dot{m}_{gas} * cp_{gas} * (T_3 - T_4)$
Economizer	$\dot{m}_{gas} * cp_{gas} * (T_B - T_5) = \dot{m}_{11} * cp_{water} * (T_{11a} - T_{11})$
Evaporator	$\dot{m}_{gas} * cp_{gas} * (T_A - T_B) = \dot{m}_{11} * (h_{6a} - h_{11a})$
Superheater	$\dot{m}_{gas} * cp_{gas} * (T_4 - T_A) = \dot{m}_{11} * (h_6 - h_{6a})$
Steam Turbine	$\dot{W}_{ST} = \dot{m}_7 * (h_7 - h_9)$
Condenser	$Q_{rej} = \dot{m}_7 * (h_9 - h_{10})$
Pump	$\dot{W}_P = \dot{m}_{11} * (h_{11} - h_{10})$

4.4.2 EXERGY ANALYSIS

The exergy analysis of main processes in the system is given as below:

Table 2: Exergy balance equation summary for the proposed cycle (Source: Reference [25])

Component	Exergy Balance Equation
Compressor	$\dot{E}_1 + \dot{W}_C = \dot{E}_2 - \dot{E}_{Des,C}$
Combustion Chamber	$\dot{E}_2 + \dot{E}_f + \dot{E}_{Steam_{inj}} = \dot{E}_3 + \dot{E}_{Des,CC}$
Gas Turbine	$\dot{E}_3 - \dot{E}_4 = \dot{W}_{GT} + \dot{E}_{Des,GT}$
Economizer	$\dot{E}_B - \dot{E}_7 = \dot{E}_{11a} - \dot{E}_{11} + \dot{E}_{Des,E}$
Evaporator	$\dot{E}_A - \dot{E}_B = \dot{E}_{6a} - \dot{E}_{11a} + \dot{E}_{Des,Ev}$
Superheater	$\dot{E}_4 - \dot{E}_A = \dot{E}_6 - \dot{E}_{6a} + \dot{E}_{Des,E}$
Steam Turbine	$\dot{E}_7 - \dot{E}_9 = \dot{W}_{ST} + \dot{E}_{Des,ST}$
Condenser	$\dot{E}_9 - \dot{E}_{10} + \left(1 - \frac{T_0}{T_{con}}\right) * Q_{con} = \dot{E}_{Des,con}$
Pump	$\dot{E}_{10} - \dot{W}_P = \dot{E}_{11} + \dot{E}_{Des,P}$

The first law and second law efficiencies can be calculated as:

$$\eta_I = \frac{\dot{W}_{net}}{(\dot{m}_f * LHV_{fuel}) + \dot{Q}_{sun}} \qquad (4.4.18)$$

$$\eta_{II} = \frac{\text{Exergy recovered}}{\text{Exergy Available}} \qquad (4.4.19)$$

4.5 MODEL VALIDATION

The two models for combined steam and gas cycle and solar hybrid STIG are validated individually and then were incorporated together to form the proposed cycle.

i) For validating and comparing the simple STIG and solar hybrid STIG, a cycle described in the study of **Livshits and Kribus [18]** is used. The parameters of comparison are given below:

Table 3: Parameters used in the modeling Solar hybrid system

(Source: Ref [18])

Parameters used	**Numerical value**
Inlet Air Temperature (℃)	25
Compressor Pressure Ratio	30
Inlet Pressure (kPa)	101
TIT(℃)	1200
η_C(%)	85
η_{GT} (%)	90
Drop in pressure at heat exchangers (%)	4
Flow rate (kg/s) for inlet air	1

The model was prepared in EES software and the performance was compared with the result obtained in the previous work **Livshits and Kribus [18]**, the comparison is shown in table 4.

Table 4: Relative performance of this work and that in reference [18]

Parameters	Conventional STIG [18] (SAR=0.15)	This work (SAR=0.15)	Solar STIG [18] (SAR=1.2)	This work (SAR=1.2)
Compressor Work(kW)	587.5	584.7	587.5	584.7
Turbine Work(kW)	1102	1106	2679	2701.7
Net Output(kW)	513.9	521	2086	2117
$\dot{Q}_{sun}$(kW)	0	0	2186	2133
Thermal Efficiency[%]	49.8	50.56	41.4	40.23
Turbine outlet temp. [℃]	470	472.3	529.9	513.6

From table 4 it can be seen that the relative performance the of the modeled system is nearly same as that in the study of **Livshits and Kribus [18]** with difference in each calculation to be quite low, for Conventional STIG cycle and solar STIG cycle.

ii) For validating the performance of combined cycle the reference system is taken from **sanaye et al.** [25] The input parameters and the and the comparison of performance is shown in table 5 and 6 respectively.

Table 5: Input parameter in reference [25]

Parameters	**Value**
Ambient air temperature(℃)	48
Ambient pressure(bar)	1.01
TIT(℃)	1060
Fuel	CH_4
Pressure ratio	11.3
η_{c_S}	87
η_{GT_S}	91
η_{ST_S}	90

Table 6: Comparison of performance of this work and that in reference [25]

Parameters	Reference cycle [25]	This work
Gas Turbine Output [MW]	124.7	124.39
Steam Turbine Output [MW]	39.78	37
Fuel Consumption(kg/s)	7.40	7.7
Combined cycle thermal efficiency (%)	43.50	41.38

From table 6 it is clear that the output performance is nearly same with the difference in this work and reference to be very low. So, the mathematical model is correct.

Thus now the proposed cycle figure 4.2.2(a) can be subjected to parametric analysis. The input parameters of the proposed cycle and the range of variation of different input variable parameters are as given in table 7. The constant input variable parameters are given in table 8.

Table 7: Input parameters and the range of variation

Input Variable Parameters	Range of variation
Pressure ratio(bar)	5 – 40
TIT(K)	1000-2000
Inlet Air Temperature(℃)	5-50
Pinch Point Temperature(℃)	10 – 50
Steam Saturation Pressure(bar)	5-30
Inlet air mass flow rate(kg/s)	200-500

Table 8: Constant parameters and their value

Constant Parameters	**Value**
Isentropic Compressor Efficiency (%)	89
Isentropic Turbine Efficiency (%)	92
Inlet air mass flow rate(kg/s)	484.3
Pinch point temperature (℃)	18.6
Approach (℃)	15
Steam injection (% of generated steam)	5
SAR	0.15
IAT (℃)	25
Compressor Pressure Ratio	11.3
TIT(℃)	1060

The final model for the proposed cycle is prepared in EES software and the results of variation of these input parameters and their effects on the proposed system are discussed in next chapter.

CHAPTER 5

RESULTS AND DISCUSSIONS

Following results are obtained after the parametric analysis of combined cycle with partial steam injection:

5.1 Variation of parameters with inlet air temperature or compressor inlet temperature

5.1.1 Variation in net work output

The variation of net work output in the proposed system with IAT is shown in figure 5.1. The amount of steam injection is considered to be 5% of the total generated steam. Also the SAR is kept at 0.15 for the proposed configuration (GT+ ST+ STIG cycle). The TIT and the compressor pressure ratio are kept constant.

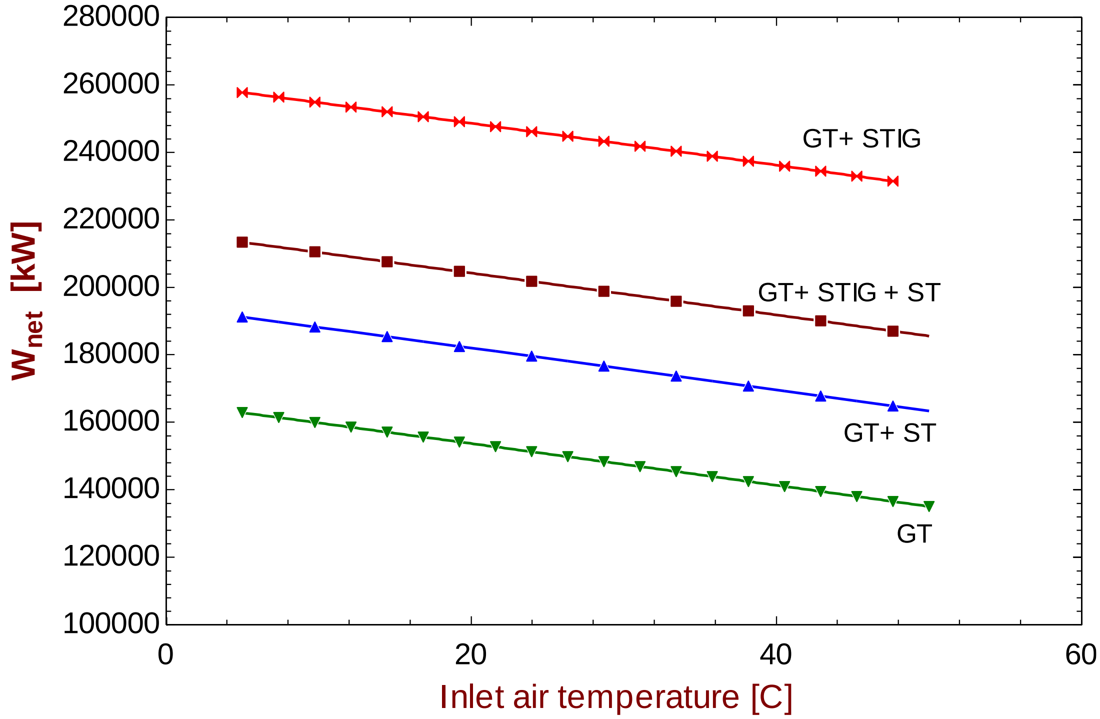

Figure 5.1 Net work output variation of different schemes with IAT

Observation 5.1

From figure 5.1 it is clear that the work output declines with the rise in inlet air temperature for various possible configurations whether it be simple GT cycle, gas turbine combined with steam generation, combined system with partial steam injection or the conventional STIG cycle. Also it becomes quite clear that the work output has increased from simple gas cycle to conventional STIG, thus justifying the use of steam injection for power enhancement.

5.1.2 Variation in first law efficiency of the cycle

First law efficiency variation for the proposed system with IAT is as depicted in figure 5.2, the amount of injected steam is considered to be 5% of the total generated steam. Also the SAR is kept at 0.15 for the proposed configuration (GT+ ST+ STIG cycle). Other parameters are kept constant.

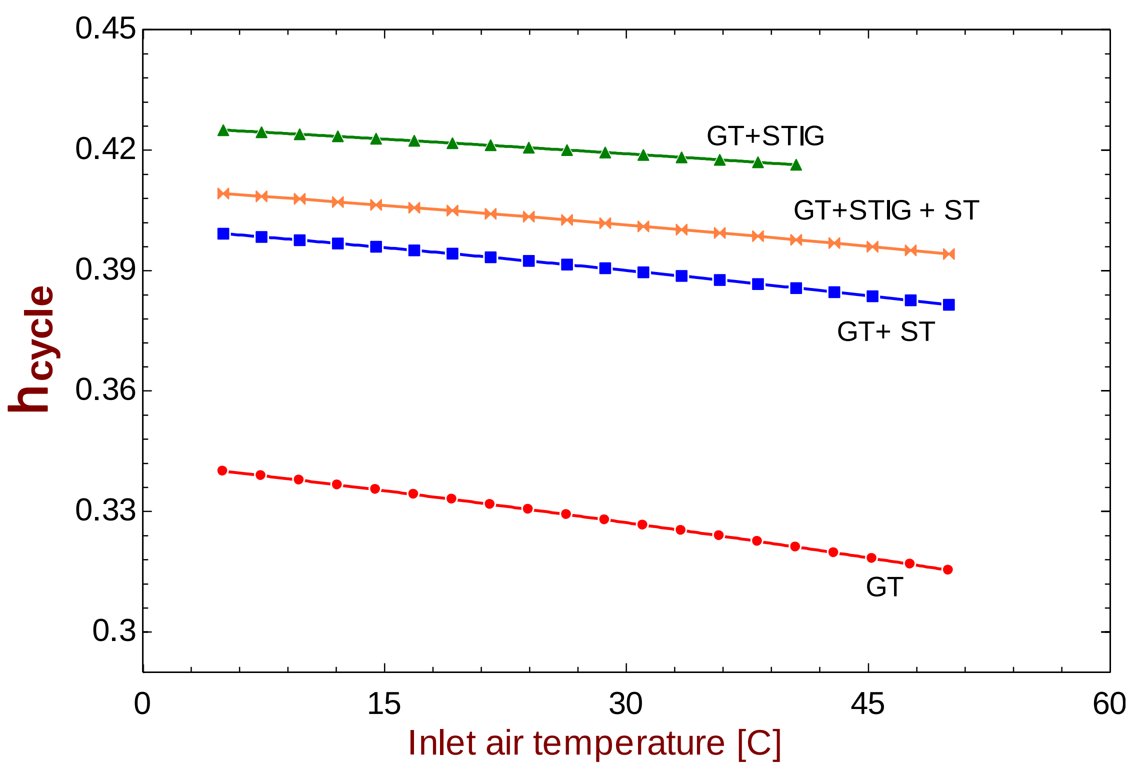

Figure 5.2 First law efficiency variation of different schemes with IAT

Observation 5.2

After observing the variation of first law efficiencies of various cycles with IAT, it is not wrong to say that the former descends with rise in the latter. Also for a given inlet air temperature, the STIG system has the highest efficiency while that of the GT system is the lowest. The gap amongst the efficiencies of the GT system and the other arrangements is quite significant. This difference is less at lower inlet air temperatures (up to 10 degrees) while it somewhat increases with increase in temperature.

5.1.3 Variation in specific fuel consumption (sfc)

Specific fuel consumption is the quantity of fuel consumed to produce a net output of 1 kW in hour. So it is an important parameter to study for variation. The variation of sfc for the proposed system is depicted in figure 5.3, the amount of injected steam is considered to be 5% of the total generated steam.

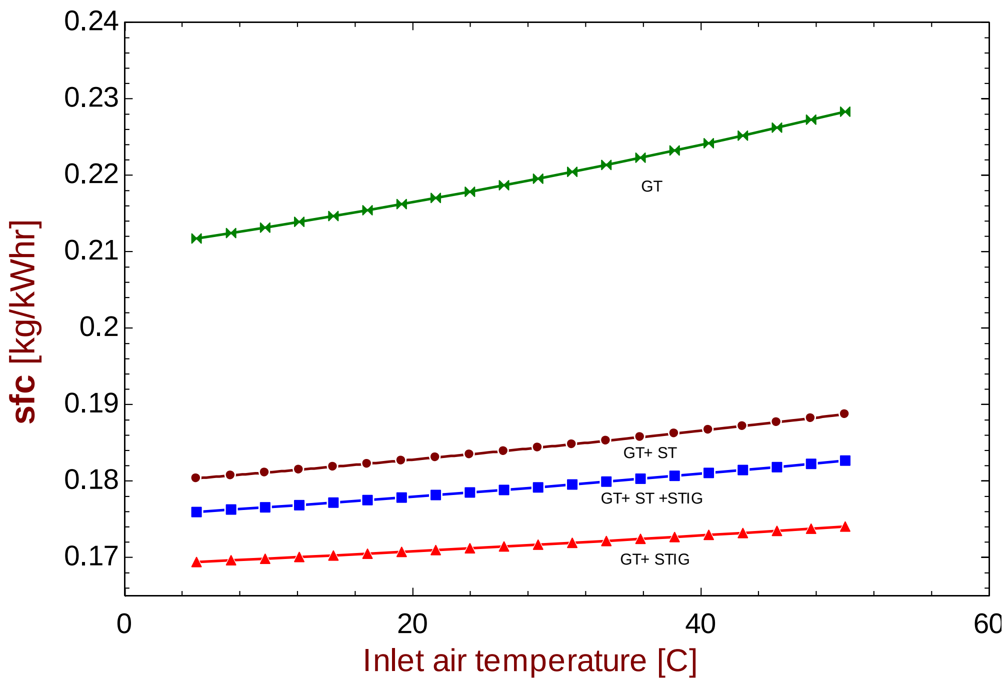

Figure 5.3 Sfc variation of different schemes with IAT

Observation 5.3

The gas turbine has the highest specific fuel consumption as is evident from figure 5.3 while that of the conventional STIG is the lowest. The reason is because there is no extra fuel flow across the GT in simple GT arrangement while the mass of steam provides extra work through the turbine in other cases. Also even the combined cycle with no steam injection has a much lower sfc than the simple gas turbine. Adding to that, with the rise in the IAT the requirement of fuel flow increases and the gap between the sfc of various schemes also increases.

5.2 Variation of parameters with compressor pressure ratio

5.2.1 Variation in net work output

Work output variation for the proposed configuration with compressor pressure ratio is depicted in figure 5.4.

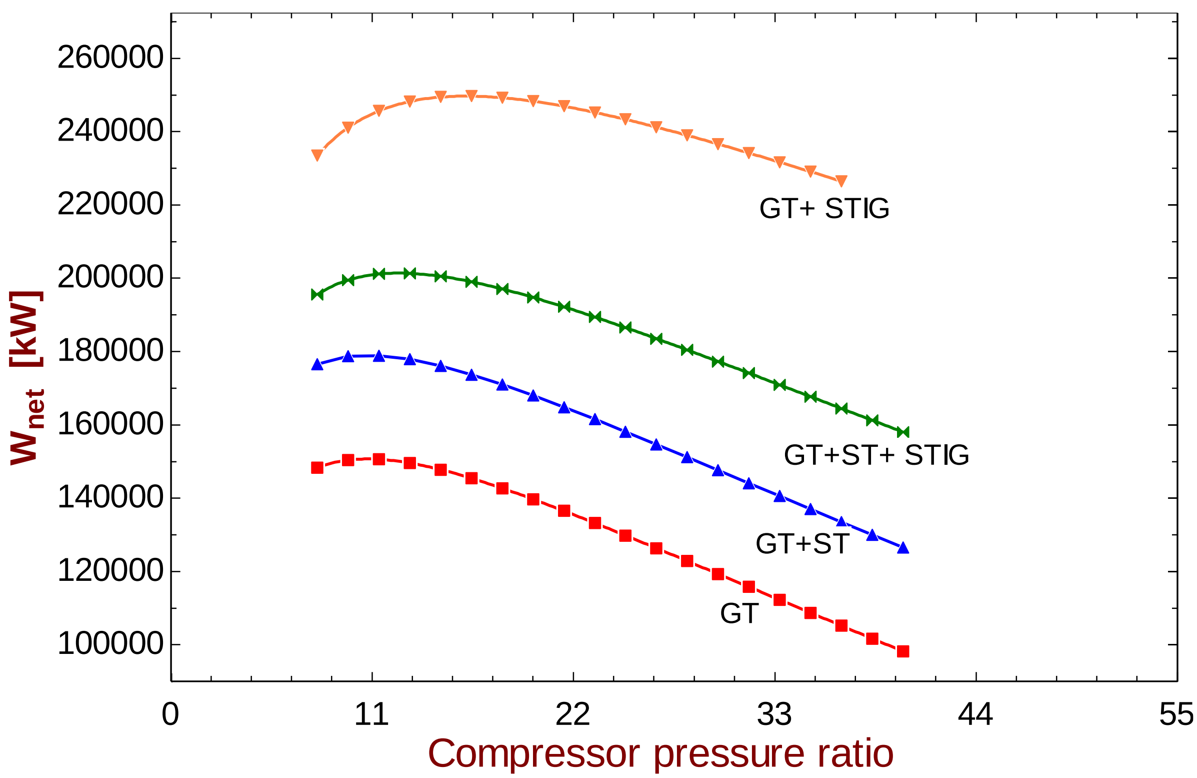

Figure 5.4 Net work output variation of different schemes with compressor pressure ratio

The amount of steam injection for the variation is considered to be 5% of the total generated steam. Also the SAR is kept at 0.15 for the proposed configuration (GT+ ST+ STIG cycle). The TIT and the inlet air temperature are kept constant at 1060℃ and 25℃ respectively.

Observation 5.4

Net work output variation of compressor pressure ratio follows a peculiar behavior as is observed from figure 5.4. It is clear that the conventional STIG cycle has the highest network output while the simple STIG cycle has the lowest output. The gap amongst the work output between the various schemes is less at lower pressure ratio while the same increases significantly as higher pressure ratios, also the work output for every scheme initially increases then attains a maximum and then finally decreases. The maximum work output for every configuration shown in the figure is attained when the pressure ratio between the range of 10 to 14.

5.2. 2 Variation in first law efficiency of the cycle

The variation of first law efficiency with compressor pressure ratio for the proposed cycle and the other configuration is shown in figure 5.5.

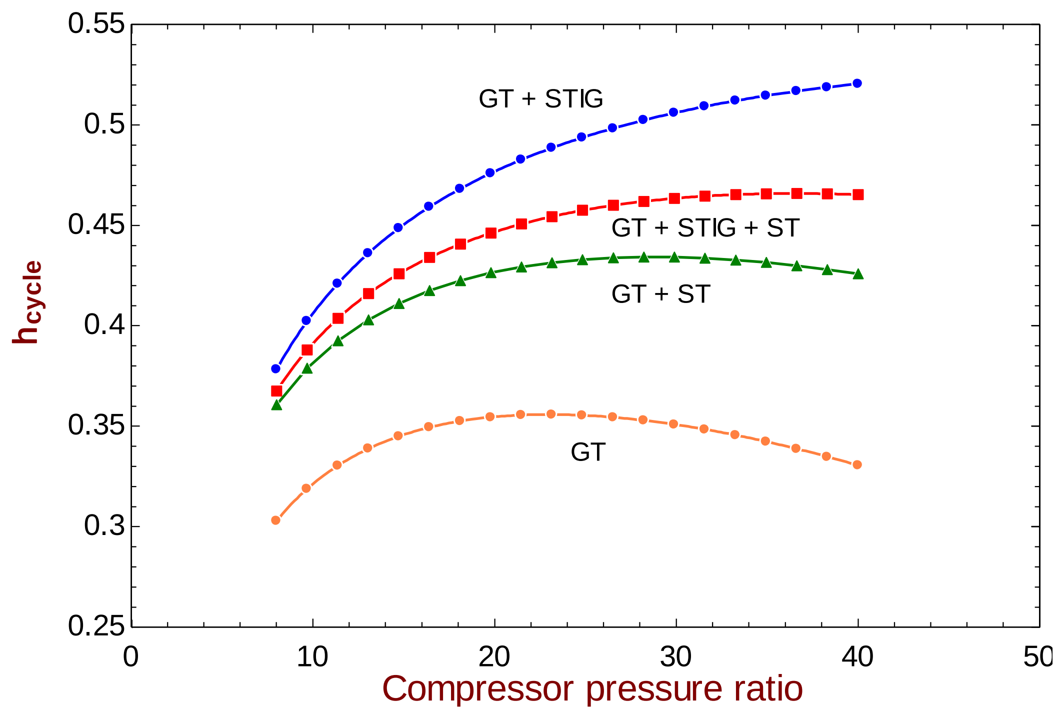

Figure 5.5 First law efficiency variation of different schemes with compressor pressure ratio

The amount of steam injection for the variation is considered to be 5% of the total generated steam. Also the SAR is kept at 0.15 for the proposed configuration (GT+ ST+ STIG cycle). The TIT and the inlet air temperature are kept constant at 1060℃ and 25℃ respectively.

Observation 5.5

The efficiency of conventional STIG is better than all the other schemes but the variation in efficiency with pressure ratio is somewhat different for different schemes as depicted in the figure 5.5. The GT which is having the least efficiency of all the schemes follows the trend of increase in efficiency then attaining a maximum followed by a decrease in efficiency. The combined steam and gas cycle follows the similar trend, but the proposed cycle has increase in its efficiency followed by a nearly constant trend. The conventional STIG cycle follows only an increasing trend within the given range of variation of pressure ratio but the rate of increase decreases after a certain range of pressure ratio variation.

5.2.3 Variation in specific fuel consumption (sfc)

Figure 5.6 depicts the variation in specific fuel consumption with the pressure ratio. All the other parameters are same as used in above analysis.

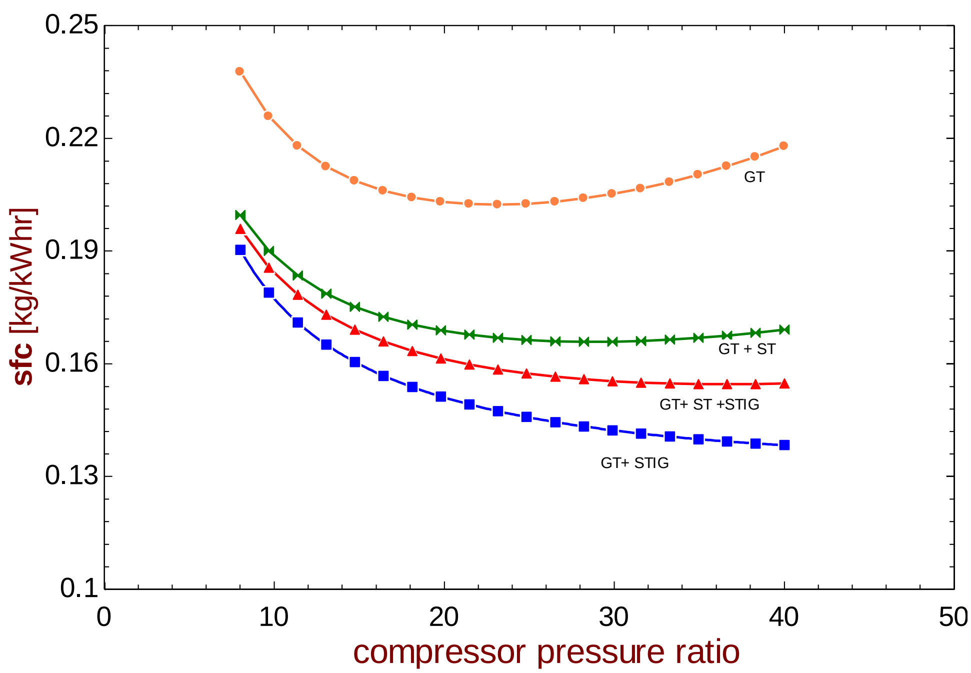

Figure 5.6 Sfc variation of different schemes with compressor pressure ratio

Observation 5.6

The variation of sfc with pressure ratio follows a peculiar trend too. While the sfc is highest for the GT arrangement, it is least for the conventional STIG. But the trends are different for the various configurations are different. The simple gas cycle initially has a decreasing trend followed by attaining a minimum and then in increasing trend for sfc. The combined gas and steam cycle follows a similar pattern but with lower values of sfc then the simple GT cycle. The proposed cycle has a decrease in sfc initially followed by maintaining a nearly constant trend with increasing pressure ratios. The conventional STIG follows a constantly decreasing trend of sfc with rise in pressure ratio in the given width of variation.

5.3 Variation of parameters with gas turbine inlet temperature

5.3.1 Variation in net work output

Net work output variation with gas turbine inlet temperature for various cycles in shown in figure 5.7.

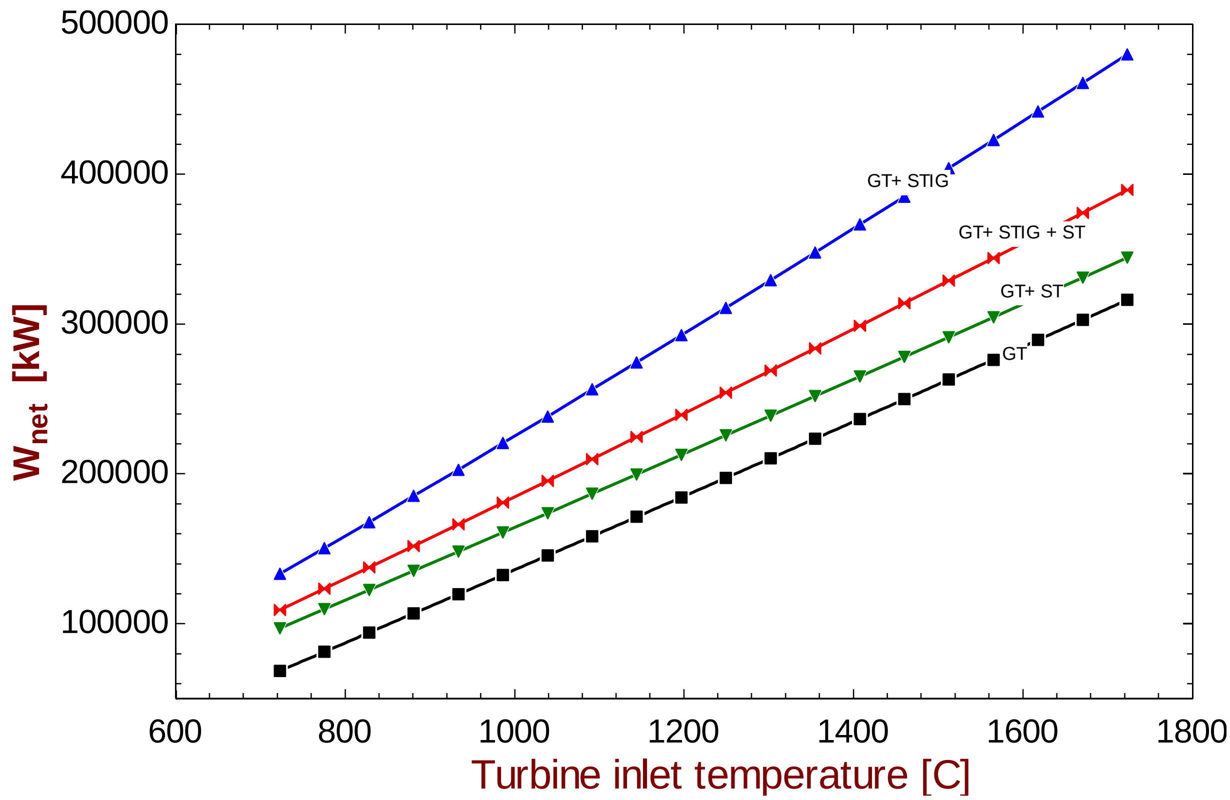

Figure 5.7 Net work output variation of different schemes with TIT

The amount of steam injection is considered to be 5% of the total generated steam. Also the SAR is kept at 0.15 for the proposed configuration (GT+ ST+ STIG cycle). The inlet air temperature and the pressure ratio are kept constant at 25℃ and 11.3 respectively.

Observation 5.7

The net work output rises with the rise in TIT of gas turbine. For simple gas cycle and the combined cycle of steam and gas the variation is linear as is evident from figure 5.7. The variation in conventional STIG and partially injected steam is linear but slope increases somewhat in the range of TIT between 1000℃ - 1200℃ after which the increasing linear trend is followed.

5.3.2 Variation in first law efficiency of the cycle

The variation of first law efficiency with TIT of gas turbine for the proposed combined cycle and the other configurations is shown in figure 5.8. All the other parameters are same as used in above analysis.

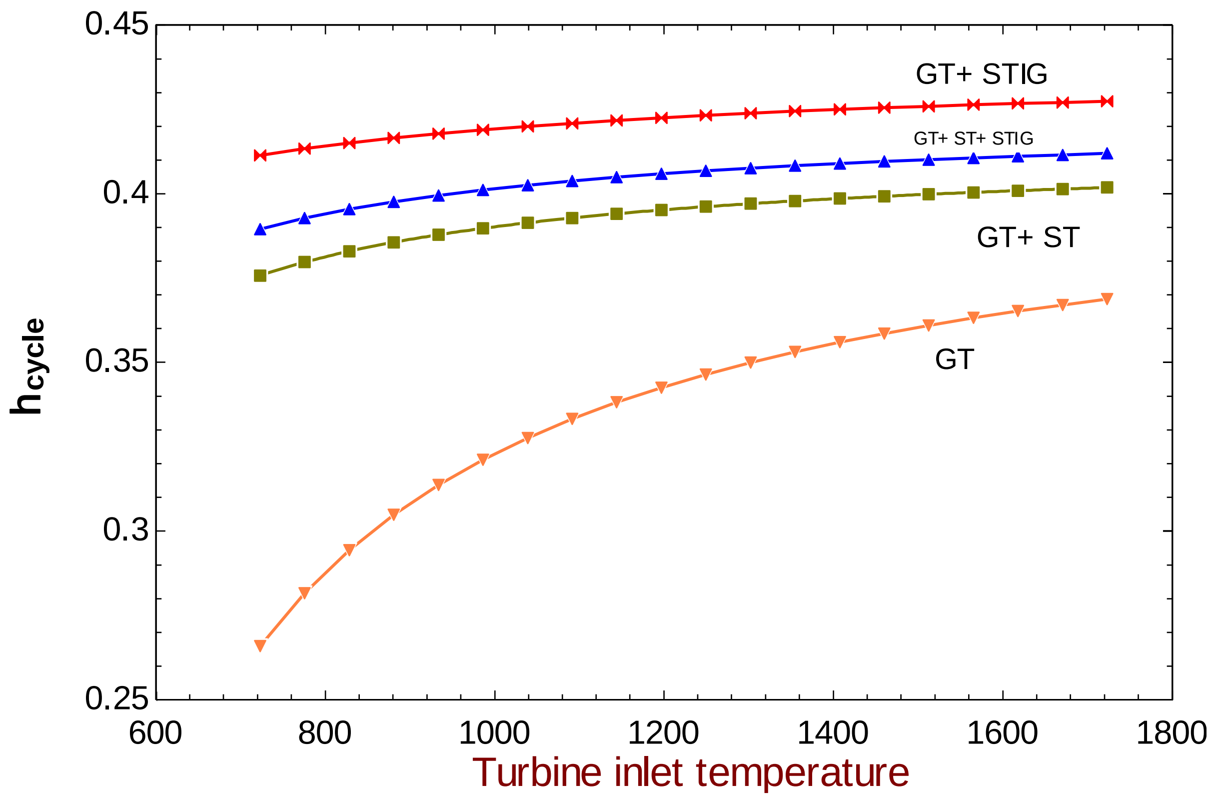

Figure 5.8 First law efficiency variation of different schemes with TIT

Observation 5.8

The increasing trend of cycle efficiency can be observed in figure 5.8 with variation in gas TIT. There is a drastic rise in efficiency of GT up to nearly 1200 ℃ after which the rate of increase decreases. A similarity in trends can be witnessed for all the other configurations. The difference between the values of cycle efficiency up to the TIT of 1200℃ of the simple GT and all the other cycles is quite large initially while it lesser afterwards.

5.3.3 Variation in specific fuel consumption (sfc)

Figure 5.9 depicts the variation in sfc with the gas TIT for the proposed configuration and the other configurations. All the other parameters like the compressor pressure ratio and the inlet air temperature are same as used in above analysis i.e. 11.3 and 25℃ respectively.

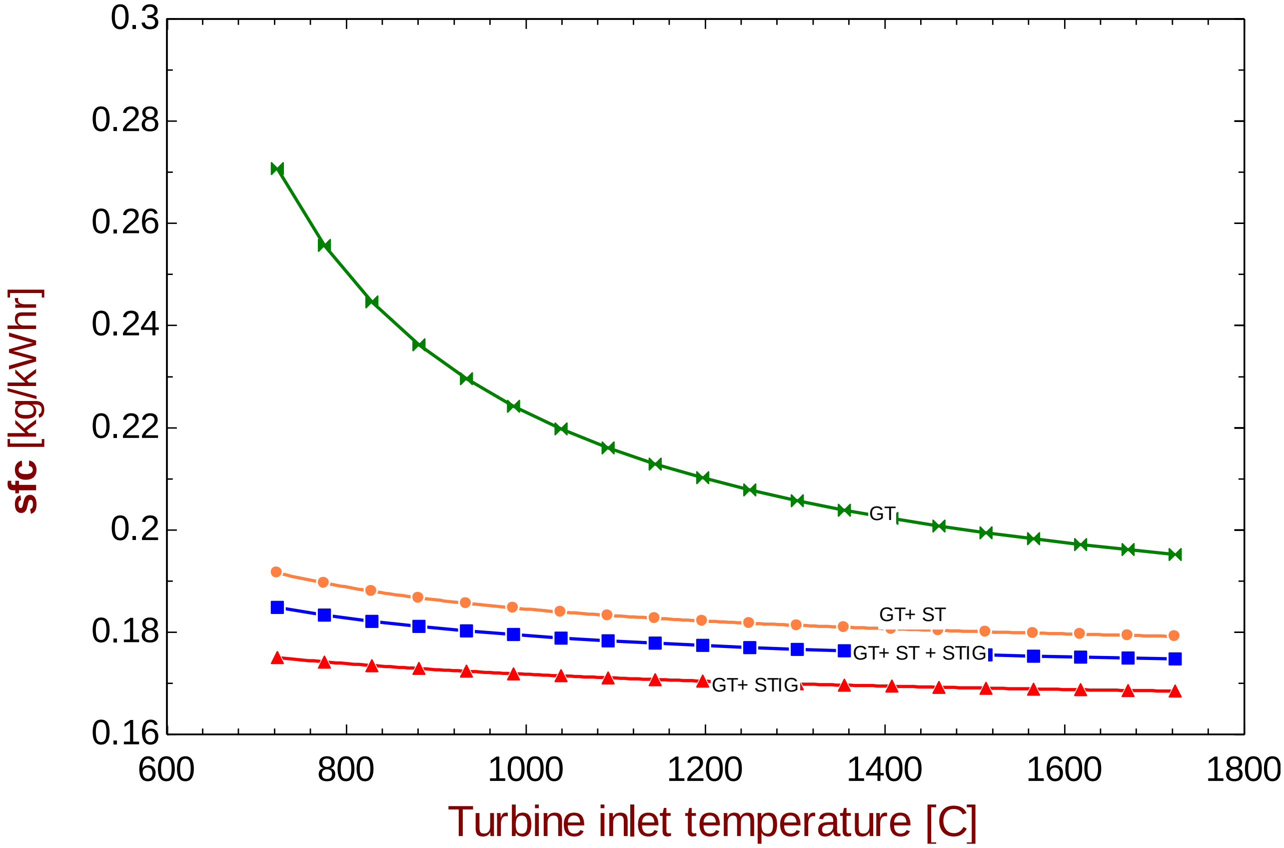

Figure 5.9 Sfc variation of different schemes with TIT

Observation 5.9

After carefully observing the figure 5.9 it is witnessed that the sfc declines with rise in TIT for every configuration mentioned figure. Also the simple gas turbine cycle has the highest sfc but it decreases significantly with increase in TIT within the given range of variation. For all the other configurations the sfc decreases but the rate of decrease is quite less than that of the other configurations. The difference in the values of sfc between the simple gas turbine and the other cycles is quite large initially while it keeps on decreasing with the increase in TIT.

5.4 Variation with Steam to air ratio (SAR)

The analysis for the above parameters was carried out at the constant SAR of 0.15 which means that the amount of steam generated for the analysis was only 15% of the amount of air. Only a part (5%) of this generated steam was supplied to combustion chamber and the rest was allowed to expand through the steam turbine. So it also becomes necessary to see the effect of variation of SAR on the efficiency of cycle.

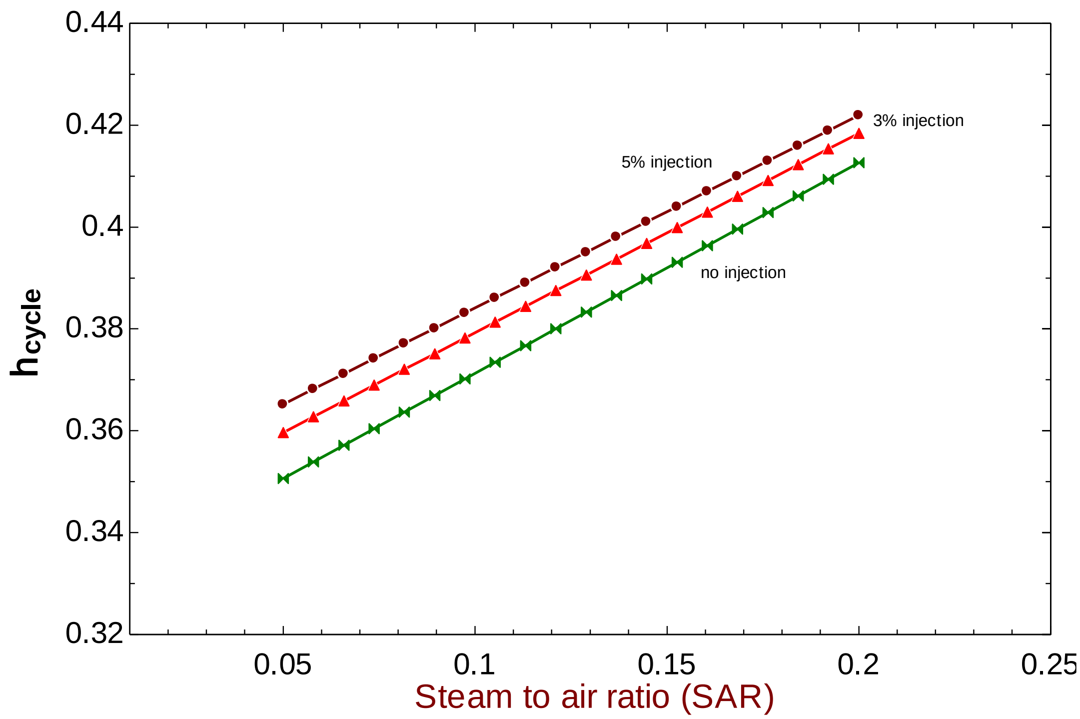

Figure 5.10 Efficiency variation of proposed cycle with SAR at different steam injection rates

Observation 5.10

The variation in SAR of the proposed cycle has a tremendous effect on the efficiency of the proposed configuration. The observed trend can be evidently said to be increasing in nature. Not only this, but the amount of injected steam also affects the efficiency of the configuration as with the rise in percentage of injected steam, the efficiency also increases. No injection condition shows the configuration of combined gas and steam cycle while that at 100% injection (not shown in figure 5.10) accounts for conventional STIG.

5.5 Variation in Steam saturation pressure

5.5.1 Variation in net work output

Another important parameter worth considering for the analysis is the steam saturation pressure. The steam injection temperature is fixed at 320°C for this analysis. All the other parameters like pressure ratio, IAT, TIT also remain same as 11.3, 25°C, 1060°C respectively. Since there is no steam generation in a simple GT cycle, it has been excluded from this analysis.

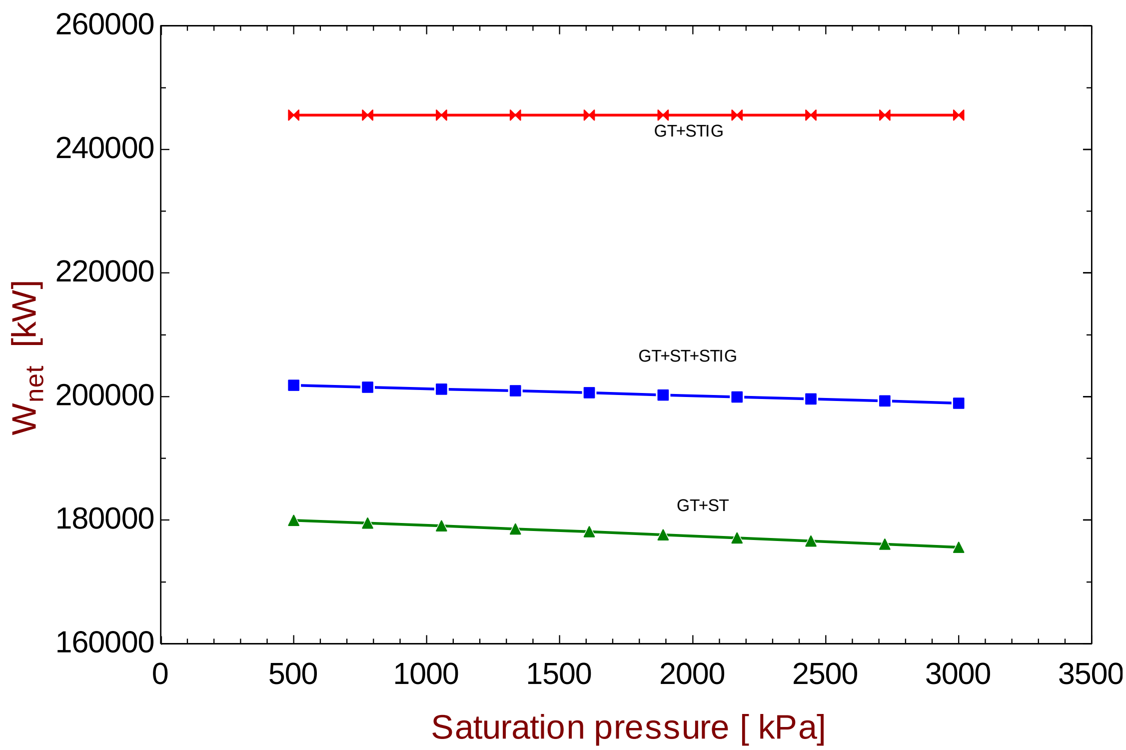

Figure 5.11 Net work output variation of different schemes with steam saturation pressure

Observation 5.11

Observing figure 5.11 it becomes clear that with the rise in saturation pressure of the steam net work output decreases but the rate of decrease is quite low for combined steam and gas cycle, and is even lower for the proposed cycle but for the STIG cycle though there is a slight decrease in output yet the rate of decrease is so low that it can nearly be assumed constant.

5.5.2 Variation in first law efficiency of the cycle

Figure 5.12 depicts the variation in efficiency of different configurations with the steam saturation pressure. For evaluation purpose of this parameter too, the temperature for injecting steam is fixed at 320°C and all the other parameters like pressure ratio, IAT, TIT also remain same as 11.3, 25°C, 1060°C respectively. Since there is no steam generation in a simple GT cycle, it has been excluded from this analysis.

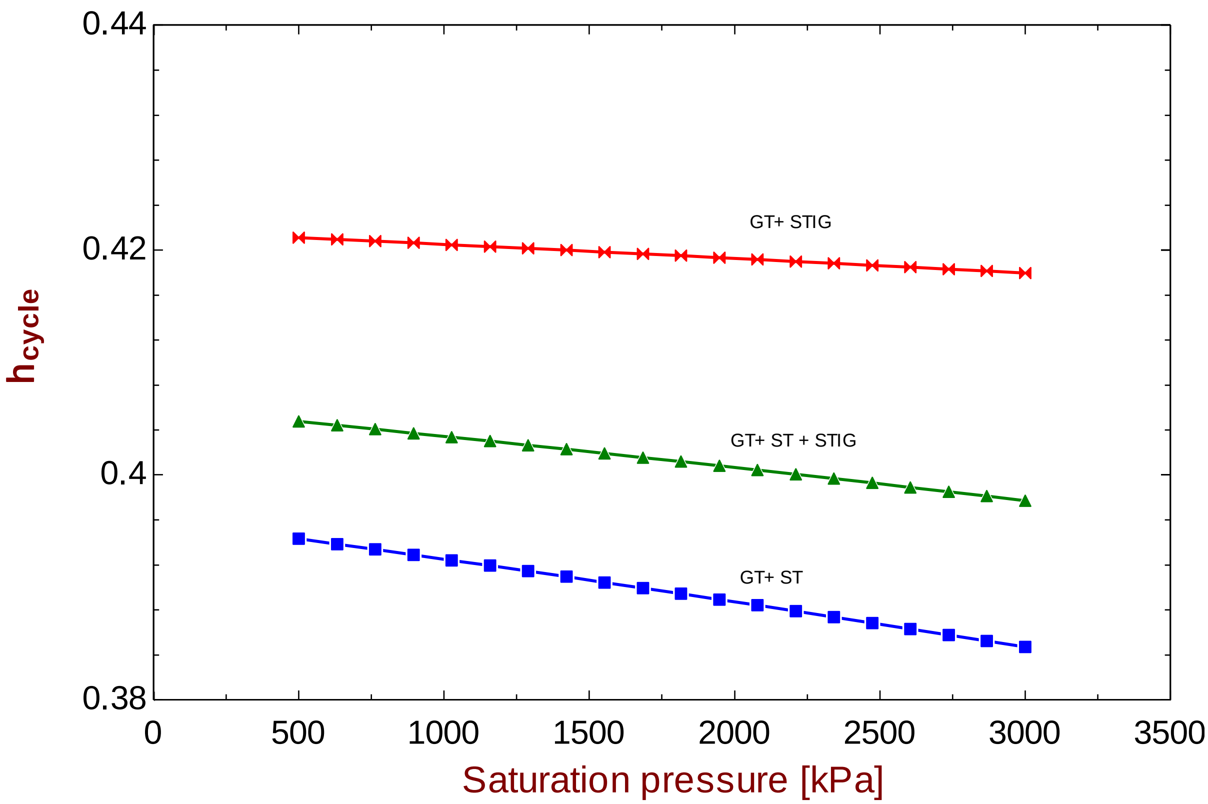

Figure 5.12 Efficiency variation of different schemes with steam saturation pressure

Observation 5.12

The cycle efficiency declines with rise in saturation pressure of the steam. From figure 5.12 it is witnessed that significant declination occurs in case of combined steam and gas cycles and the proposed cycle but the rate of decrease for STIG system is lower than both.

5.5.3 Variation in specific fuel consumption (sfc)

The result of varying the saturation pressure on the sfc is depicted in figure 5.13. All the other parameters remain same as that in the analysis of net work output and efficiency variation with steam saturation pressure.

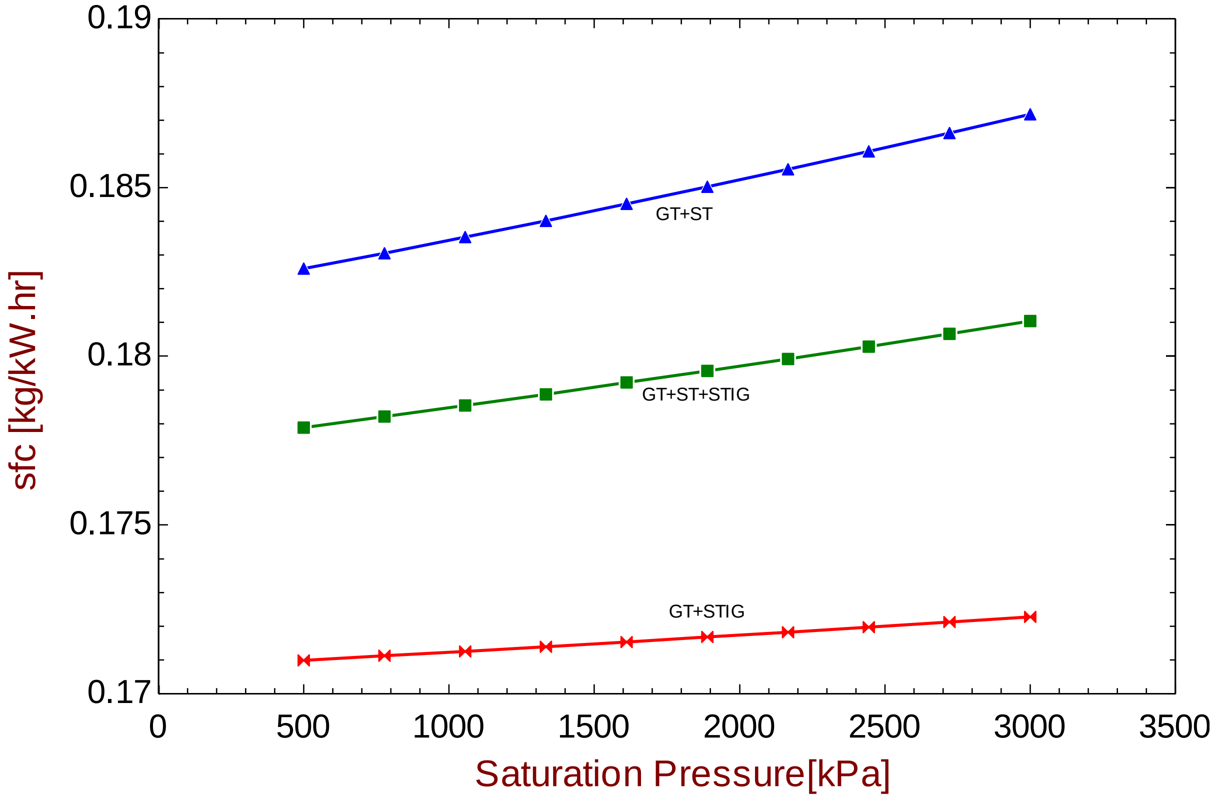

Figure 5.13 Sfc variation of different schemes with steam saturation pressure

Observation 5.13

With the rise in saturated steam pressure, the sfc rises for the schemes shown in figure 5.13. The conventional STIG has the least sfc and the combined GT and ST cycle has the most sfc when compared with each other.

5.6 Variation with pinch point temperature difference

After utilizing all the heat energy of the exhaust gases they are transferred to stack after which are discharged into air. Normally the exhaust gas temperature of gas turbine is quite high which results in waste of exergy so their energy is utilized by modifying the GT cycle in various ways but the recuperation of wasteful heat energy from the exhaust of a GT system due to application of these various techniques is only appreciable if the resulting stack temperature is at least lower than 170℃, also the stack temperature cannot be lower than 120℃ to avoid condensation and formation of corrosive substances [25]. Thus variation in stack temperature is an important parameter for analysis. The pinch point temperature affects the stack temperature and its variation is shown in figure 5.14. The variation is shown for the proposed cycle only at a steam injection rate of 5%. All the other parameters are same as in the above analysis.

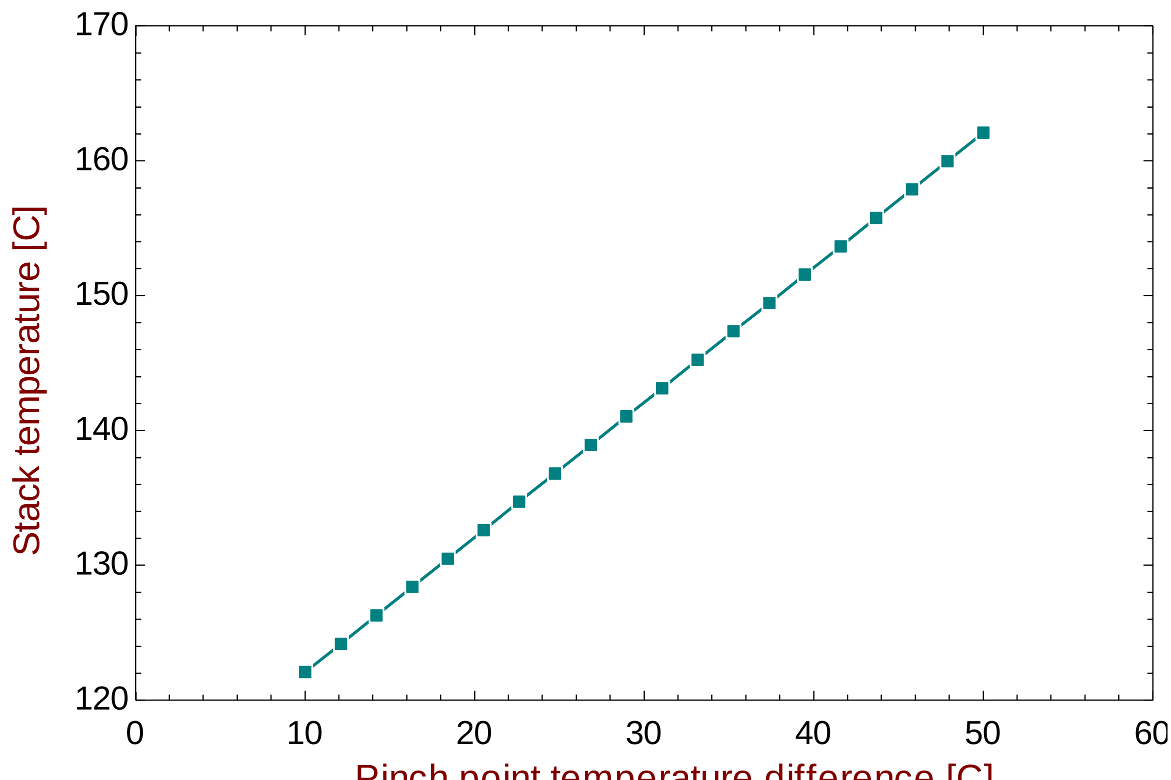

Figure 5.14 Stack temperature variation of proposed cycle with pinch point

Observation 5.14

From figure 5.14 it can be witnessed, with the rise in pinch point temperature the stack temperature increases. Thus it is necessary to keep the pinch point temperature as low as possible.

5.7 Variation with mass flow rate of air

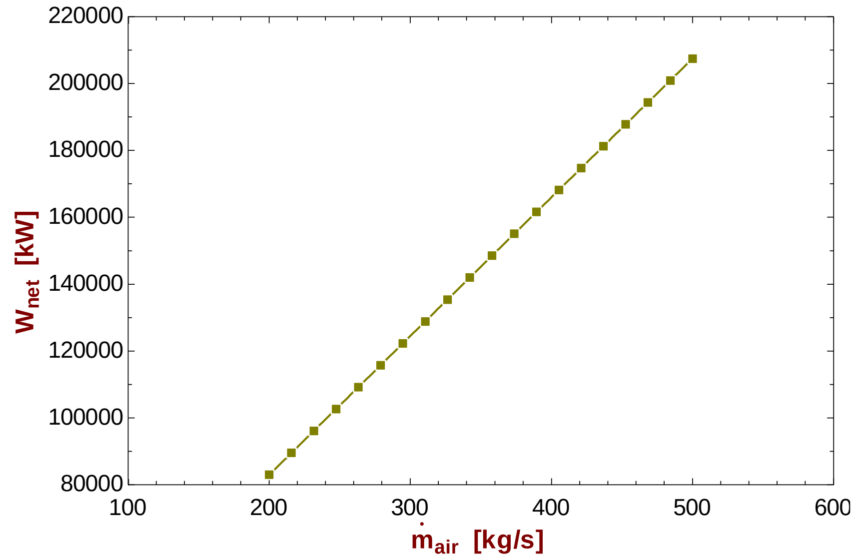

Figure 5.15(a) Net work output variation of proposed cycle with inlet air flow rate (by mass)

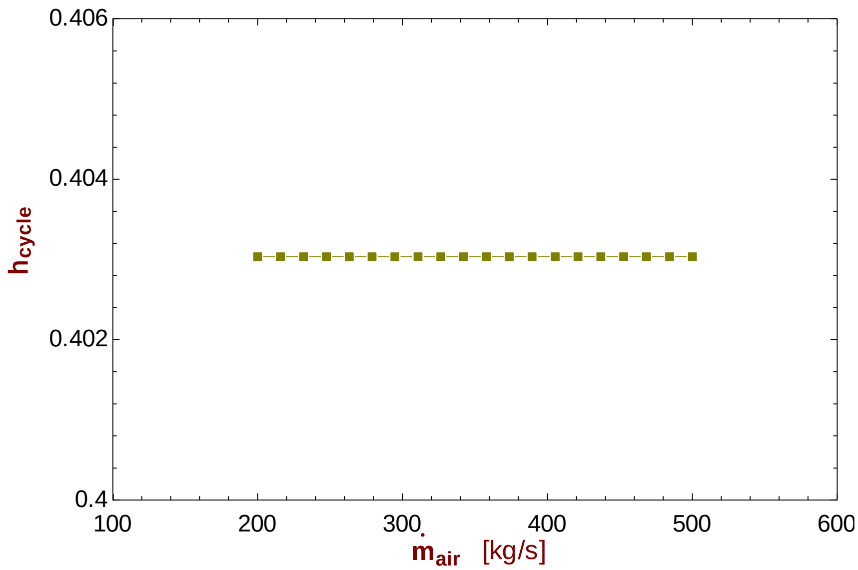

Figure 5.15(b) Efficiency variation of proposed cycle with inlet air mass flow rate

So far the analysis is done by considering constant inlet air mass flow rate. For the analysis in figure5.15(a), figure5.15(b), figure5.15(c), figure5.15(d) the flow rate(by mass) of inlet air of the proposed cycle is varied keeping the other parameters constant.

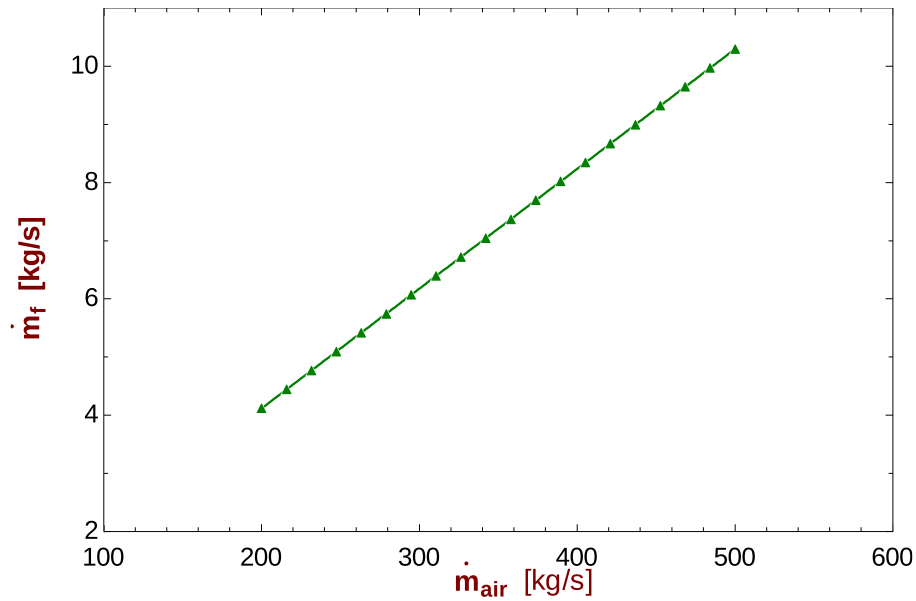

Figure 5.15(c) fuel flow rate variation of proposed cycle with inlet air flow rate(by mass)

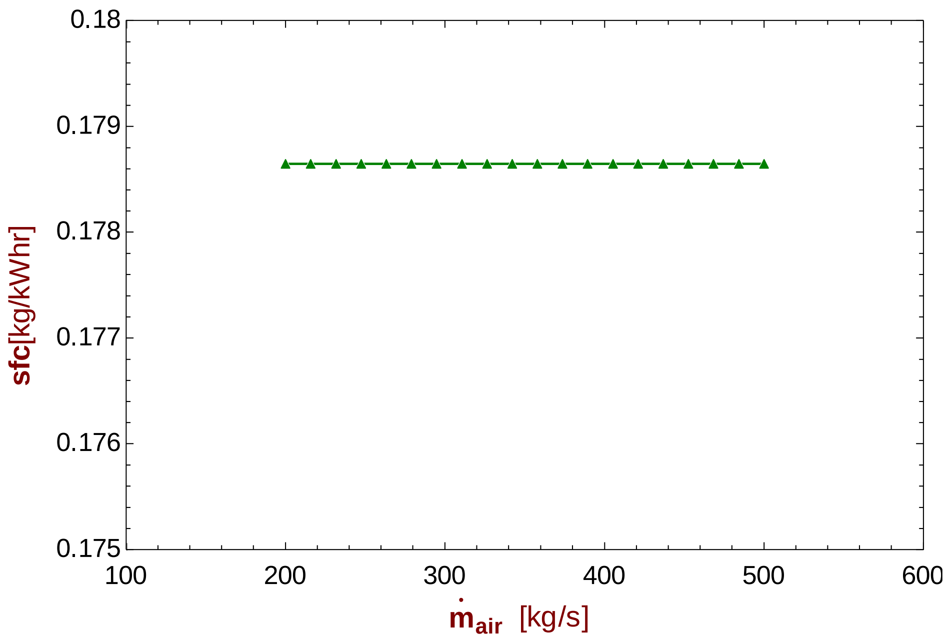

Figure 5.15 (d) sfc variation of proposed cycle with inlet air mass flow rate

Observation 5.15

From figure5.15(a), figure5.15(b), figure5.15(c), figure5.15(d) it can be collectively interpreted that the efficency and sfc of the proposed configuration does not depend upon the inlet mass flow rate air, they remain constant with it, it is rather the work output and fuel flow rate(by mass) of the configuration that rises with rise in inlet air flow rate(by mass). So it is not wrong to say that the net work output and the fuel flow rate(by mass) adjust themselves according to the flow rate(by mass) of inlet air so as to, keep the efficiency and sfc of the configuration constant.

5.8 Variation in SAR with constant solar input

From the analysis carried out above it is clear that amount of rise in quantity of steam injection escalates the output of proposed configuration with solar input, but the amount of steam that can be generated is constrained by the quantity of energy present with the exhaust gases. In order to generate more steam heat energy from can be used as an alternative to generate more steam. For this scheme only the energy of the sun is utilized, the exhaust gas energy is only utilized in super heater and economizer. Figure 5.16 shows the variation in SAR with inlet mass flow rate at constant solar input of 2086 kW [18].

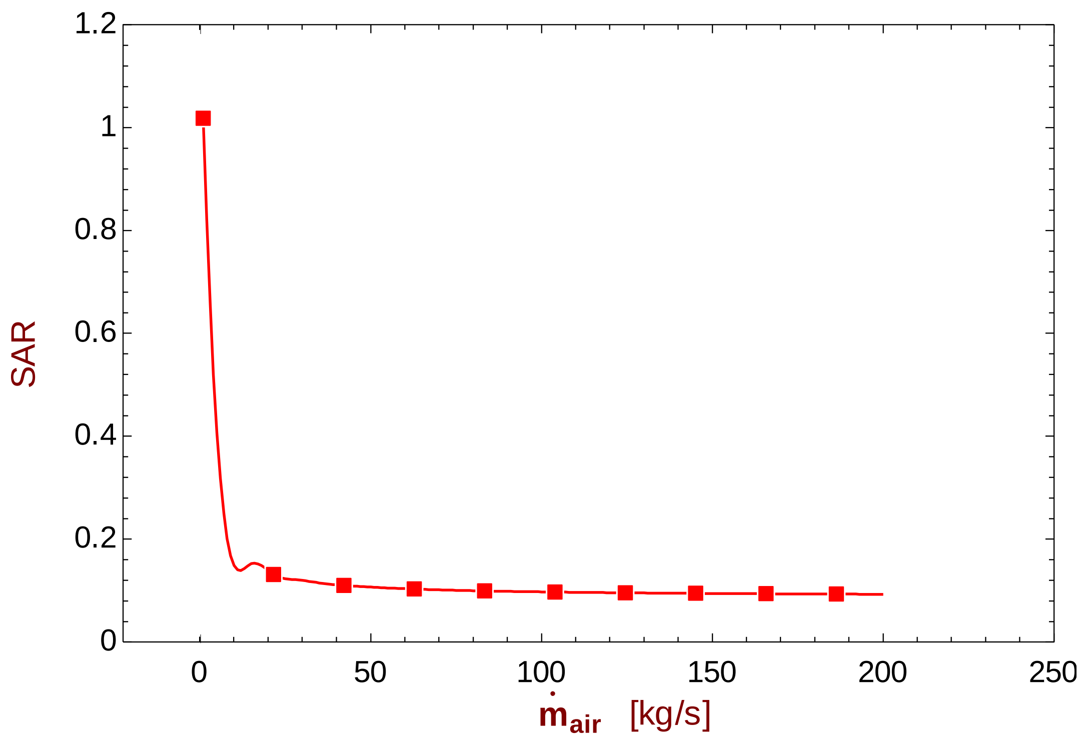

Figure 5.16 SAR variation of proposed cycle at constant solar input with flow rate of inlet air

Observation 5.16

Observation in figure 5.16 indicates that at a constant heat input of 2086 kW from sun the SAR vary significantly for lower ranges of inlet mass flow rate air but on keep varying it, the SAR attains a nearly constant value, this is because of the reason that the solar input available is quite low.

5.9 Variation in solar input with SAR

Practically speaking, the heat energy available from sun to convert the water after the economizer into saturated steam is quite low and the amount of steam generation depends upon the magnitude of heat input available. But, theoretically evaluation can be performed to obtain values of heat input required to obtain higher values of SAR as shown in figure 5.17.

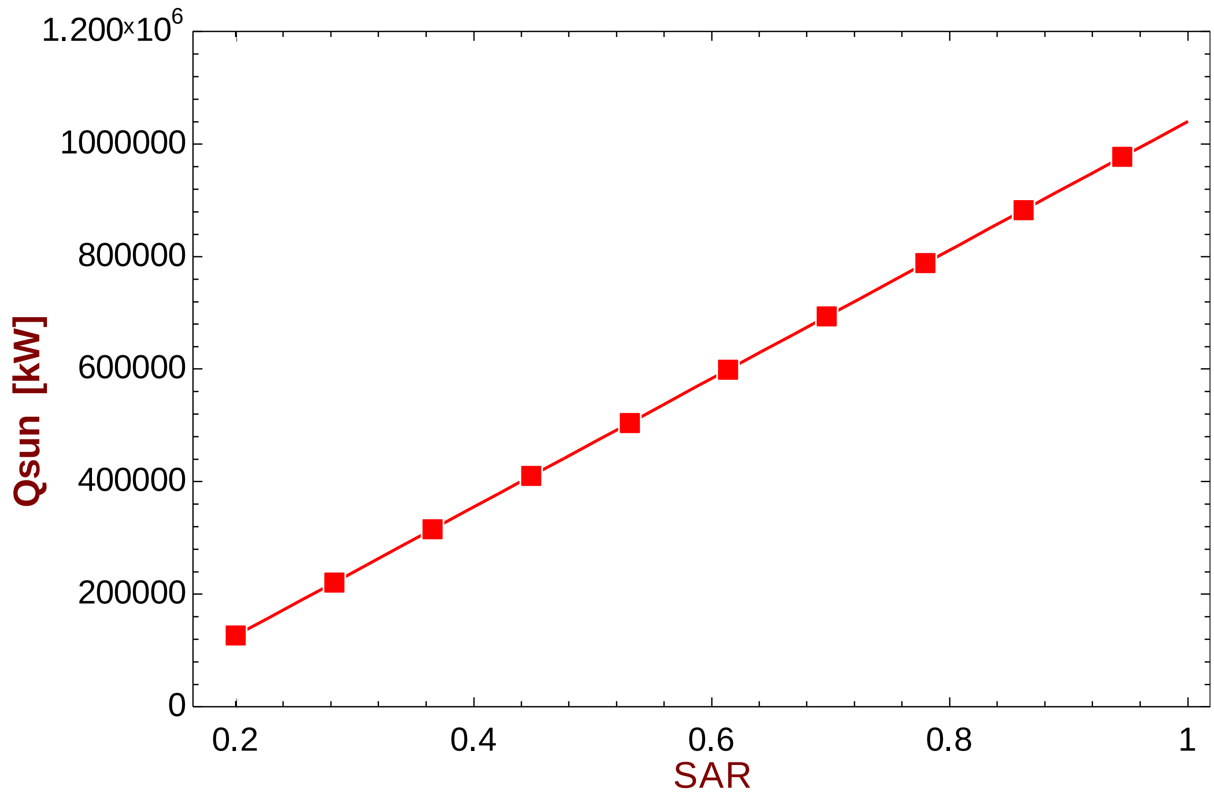

Figure 5.17 Solar input variation of proposed cycle with SAR

Observation 5.17

It can be clearly seen from figure 5.17 that even to obtain 20% SAR very high magnitude of heat input is required for the proposed cycle with solar heat input.

After carrying out the above evaluation and using the values for input variables as given in table 7 and 8, the following table shows the performance of the proposed configuration(without solar input).

Table 9: Results of the proposed cycle (without solar input)

Performance parameter	**Result**
Net work output(MW)	201.12
Sfc(kg/hr.kW)	0.1786
Fuel consumption(kg/s)	9.98
Thermal efficiency (%)	40.3
Inlet air flow rate(kg/s)	484.7
Steam injection rate(%)	5
Stack Temperature(°C)	130.8

It is pointless to calculate the efficiency for the proposed configuration with solar input because the adding the heat input would only decrease the efficiency as only a minute amount of heat is practically available from the sun so it would not add much to the amount of generated steam.

CHAPTER 6

CONCLUSION

6.1 Conclusion

With the rise in compressor inlet temperature or IAT, both the first law efficiency and the net work of a simple GT, simple STIG cycle varies inversely while the sfc varies directly.

With the maximum amount of steam that can be obtained in the HRSG supplied to the steam cycle (without any injected steam) combined with GT cycle, for the same amount of steam cycle efficiency of the simple GT cycle is lowest followed by gas turbine with steam cycle followed by partial steam injection and the highest efficiency is that of the conventional STIG system. For the partial injection (proposed cycle) especially while no new steam is generated for the steam turbine cycle then the cycle efficiency also varies inversely with IAT while the sfc varies directly.

Thus it can be concluded that the most efficient results are obtained when of all the generated steam is injected in the GT cycle but practically the design considerations limit the amount of steam injection so it would be effective to pass the steam remaining after injection to a steam cycle and the resulting cycle will have more efficiency then the simply combined cycle and the simple GT cycle.

Also, utilizing solar heat for increasing the amount of steam generation is a good option, provided that it is feasible economically. Moreover, the practical value of heat energy available from the sun is limited and can only be utilized for steam generation up to some extent and the collective amount of steam generation from exhaust heat recovery and solar energy can be used for increasing the amount of steam injection.

6.2 FUTURE SCOPE

In a simple combined steam and gas cycle, the steam is separately generated for the sole motive of generating more work output. The proposed system of partial steam injection generates more work output then the simple combined cycle, so if a part of steam generated for steam cycle is injected in the GT cycle, then the output will surely become better than that of the latter cycle. Moreover, the exhaust energy can be utilized to further super heat the generated steam. This project work doesn’t carry out the economic analysis of the proposed cycle, so there is a scope of further study economic feasibility of this cycle in the future.

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