EFFECT OF COMPONENT EFFICIENCIES ON THE PERFORMANCE OF REHEAT AND REGENERATIVE BRAYSSON CYCLE

A project report submitted in partial fulfilment of the requirement for the award of the degree of

BACHELOR OF TECHNOLOGY
IN
MECHANICAL ENGINEERING
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ABSTRACT

In the present scenario of Energy crisis, particularly in the power sector, it is important to use the available Conventional energy resources efficiently. Hybrid cycles are being developed and tested to maximize the gains. One such hybrid gas turbine cycle is proposed by Frost et.al in 1997, based on the conventional Brayton cycle for the high temperature heat addition process while adopting the Ericsson cycle for the low-temperature heat rejection process. It thus incorporates the thermodynamic advantages of a combined gas and steam turbine (CCGT) cycle without the irreversibilities of the boiler and the ancillaries of the steam turbine plant. This couldn't be implemented as attaining isothermal compression is very difficult.

To make its implementation a reality, the original cycle has been modified by Chandramouli et.al in 2015, by incorporating regenerator and a cooler before the final compression process. Reheating was included for augmenting the power output. It has been found that the energy and exergy efficiency of the cycle equals the efficiency of normal Braysson cycle at a much lower pressure ratio. The efficiency achieved through this modified cycle with 2 stages of compression is only 2.2% less than the efficiency through ideal isothermal compression. Hence the effect of number of stages of final compression is highly nullified in the proposed cycle and it may not be a necessity to attain isothermal compression. In the present work, we have focused on the effect of different component efficiencies on the performance of the reheat and regenerative Braysson cycle.

As TIT increased from 800K to 1200K an efficiency improvement of 13.31% was observed by taking the component efficiency values Eta_c=0.9, Eta_cs=0.9, Eta_t=0.9, Eta_reg=0.85. We observed that as TIT increases for any given component efficiency, the performance of the hybrid cycle improves except in the case of regenerator. At all TIT, as the regenerator effectiveness improved, we observed that the work output reduced by a negligible value.

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N = Number of stages of multi-stage intercooled compressor

T_0 = Dead state temperature (K)

C_V = Specific heat at constant pressure of both air and gases (kJ/kg K)

l.c.v = Lower calorific value of fuel (kJ/kg)

m_f = mass flow rate of fuel in the combustion chamber (kg/ s)

m_{f1} = mass flow rate of fuel in the reheater (kg/s)

m_g = mass flow rate of gases through the main gas turbine (kg/s)

m_{g6} = mass flow rate of gases through the reheat gas turbine (kg/s)

r_p = pressure ratio across the main compressor

r_{p1} = pressure ratio across the main turbine

r_{p2} = pressure ratio across the reheat turbine

r_{p_0} = overall pressure ratio across the multi-stage intercooled compressor

η_C = Isentropic efficiency of the main compressor

η_{CS} = Isentropic efficiency of a stage of multi-stage intercooled compressor

η_T = Isentropic efficiency of the turbine

η_{reg} = First law efficiency of regenerator

γ = Ratio of specific heats

$K = (\gamma-1)/\gamma$

ΔT = Temperature rise in a stage of multistage compression process (K)

T_1= inlet temperature of main compressor (K)

T_2= inlet temperature of regenerator (K)

T_3= outlet temperature of regenerator (K)

T_4= turbine inlet temperature (TIT) (K)

T_5= high pressure turbine outlet temperature (K)

T_6= low pressure turbine inlet temperature (K)

T_7= low pressure turbine outlet temperature (K)

T_8= regenerator outlet temperature (K)

T_9= cooler outlet temperature (K)

T_a= exit temperature after 1st stage of multi stage compressor (K)

η_{en}= reheat and regenerative Braysson cycle efficiency

CHAPTER 1
INTRODUCTION

1 INTRODUCTION

In the present scenario of Energy crisis, particularly in the power sector, it is important to use the available Conventional energy resources efficiently. Hybrid cycles are being developed and tested to maximize the gains. One such hybrid gas turbine cycle which is known as Braysson cycle is proposed by Frost et.al in 1997, based on the conventional Brayton cycle for the high temperature heat addition process while adopting the Ericsson cycle for the low-temperature heat rejection process. It thus incorporates the thermodynamic advantages of a combined gas and steam turbine (CCGT) cycle without the irreversibility's of the boiler and the ancillaries of the steam turbine plant.

1.1 BRAYTON CYCLE

A Brayton cycle is a thermodynamic cycle that explains the operation of a heat engine having constant pressure. An engineer George Brayton was originally designed the Brayton cycle to use it in a piston engine. It is known as the Brayton cycle due to the name of its inventor.

A Brayton cycle describes how a heat engine extracts energy from the flowing fuel and air to produce useful work, which is further uses to drive a vehicle by providing it thrust. This cycle is also referred to as the Joule cycle. The reverse joule cycle has an external heat source in combination with the regenerator.

1.1.1 WORKING OF BRAYTON CYCLE

Isentropic compression: Firstly, the gas is sucked from the atmosphere to the engine combustion chamber. As the working gas (such as helium) is sucked, the compressor compresses it in such a way that there is no change in heat. Due to the compression process, the temperature and pressure of the gas become very high but decrease its volume. This process is also known as a reversible adiabatic process because there is no heat transfer.

Isobaric heat addition (lines 2 to 3): As the isentropic compression process is completed, the compressed gas is transferred to the combustion chamber, where an external source uses to provide heat to the working gas. Due to heat addition, the compressed gas ignites, and this ignition process further adds heat to the gas. During this whole process, the enthalpy of the working gas increases, but its pressure remains constant. The net heat provide to the gas is:

$Q_{add} = H_3 – H_2$

Isentropic expansion (line 3 to 4): In this stage, the heated compressed gas enters the turbine area, where it expands. As the gas expands, its pressure and temperature reduce, but entropy remains the same. In this process, the gas works on the turbine and turns the turbine blades. These blades rotate the crankshaft, which further moves the vehicle wheels.

Isobaric heat rejection (line 4 to 1): In the isobaric heat rejection process, the unused heat of the gas expels via heat exchanger due to that temperature of the gas decreases, but pressure remains unchanged. The net heat rejected is:

$Q_{re} = H_4 – H_1$

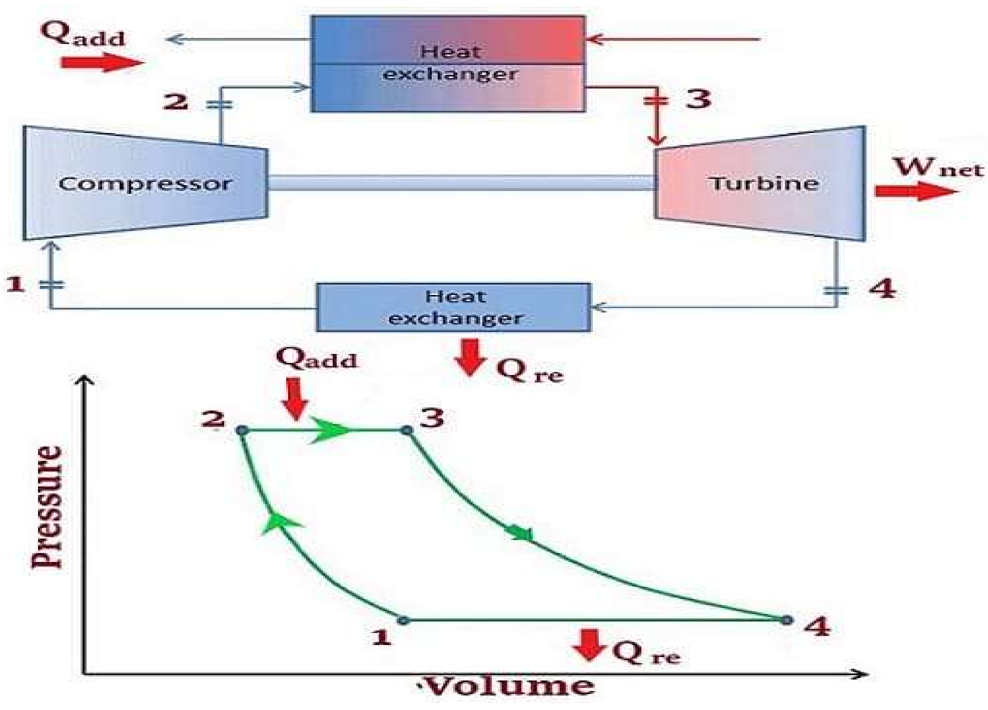

Figure 1: Working of Brayton Cycle

1.1.2 EFFICIENCY

The net output work (W) divided by the input heat supplied at high temperature (Q_H) is known as the thermal efficiency of the heat engine.

$$\eta_{th} = \frac{W}{Q_H}$$

For calculating the efficiency of the Brayton Cycle, first, we need to find how much work is done in the total internal energy.

So, Internal energy = U= −w+q1+q2 =0

According to the 1st law of thermodynamics, no heat is generated or demolished during a thermodynamic cycle. Therefore, in the Brayton cycle:

U = 0

Put U= 0 in the above equation, and the final equation is:

w=q1+q2

In the above equation:

q1 = Heat gained by the combustion process

q2 = Released heat after expansion

q1 can find by considering gas as a perfect gas along with constant specific heat (cp):

q1= cp(TI−TF)

and q2 is:

q2 =cp(TF−TI)

In the above equation:

TF = combustion chamber's final temperature

TI = Combustion chamber's initial temperature

According to the PV diagram, we can replace the TF with T4 and TI with T3, and the final equation for q1 will be

q1=cp(T4−T3)

1.1.3 ADVANTAGES

(1) High thermal efficiency in relatively low turbine inlet temperature.

(2) Compactness of the turbomachinery's and heat exchangers.

(3) Simpler cycle layout at an equivalent or superior thermal efficiency.

1.1.4 DISADVANTAGES

1) The part-load efficiency of this turbine is poor.

2) This gas turbine is very sensitive.

1.1.5 APPLICATION

The gas turbine is used in a wide range of applications. Common uses include stationary power generation plants (electric utilities) and mobile power generation engines (Ships and aircraft). In power plant applications, the power output of the turbine is used to provide shaft power to drive a generator, a helicopter rotor, etc. A jet engine powered aircraft is propelled by the reaction thrust of the exiting gas stream. The turbine provides just enough power to drive the compressor and produce the auxiliary power. The gas stream acquires more energy in the cycle than is needed to drive the compressor. The remaining available energy is used to propel the aircraft forward.

1.2 ERICSSON CYCLE

Ericsson cycle refers to a type of thermodynamic cycle which has constant pressure and constant temperature processes. This cycle also resembles the Carnot cycle where the Ericsson cycle contains two constant pressure processes instead of two isentropic processes in the Carnot cycle.

The Ericsson cycle is named after a Swedish-American inventor John Ericsson, who designed and built many unique heat engines based on various thermodynamic cycles. He is credited with inventing two unique heat engine cycles and developing practical engines based on these cycles. His first thermodynamic cycle “the first Ericsson cycle” is now called the “Brayton cycle “, in fact it is the closed Brayton cycle, which is commonly applied to modern closed cycle gas turbine engines. The second Ericsson cycle is what is now called the Ericsson cycle. The second Ericsson cycle is similar to the Brayton cycle but uses external heat and incorporates the multiple use of an inter-cooling and reheat.

1.2.1 WORKING

P-V and T-S Diagrams of the Ericsson Cycle.

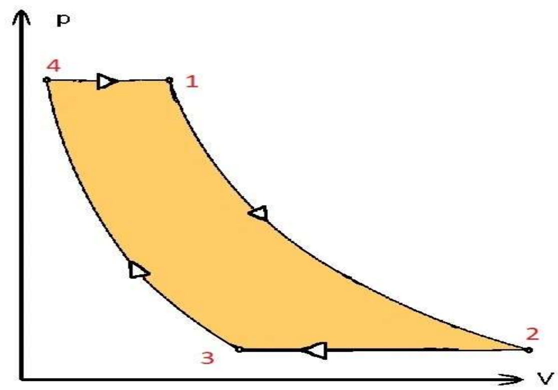

Figure 2: P-V Diagram of Ericsson Cycle

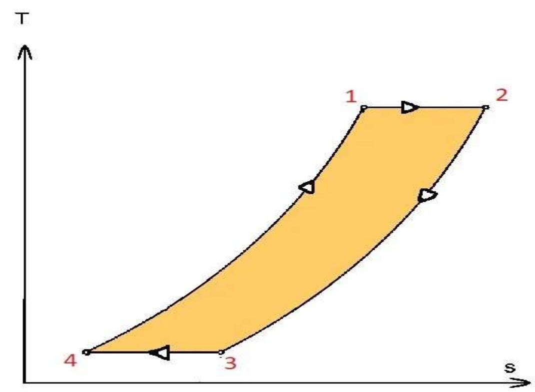

Figure 3: T-S Diagrams of the Ericsson Cycle.

Ericsson Cycle is comprised of four processes

Isothermal heat addition process (1-2): Volume of the system increases due to isothermal heat addition. A drop in pressure also happens in this process.

Isobaric heat removal process (2-3): Both temperature and volume of the system decreases due to isobaric heat removal.

Isothermal heat removal process (3-4): It is a compression process hence pressure of the system increases and volume decreases.

Isobaric heat addition process (4-1): Both temperature and volume of the system increases due to isobaric heat addition.

1.2.2 EFFICIENCY

Efficiency of the Ericsson Cycle is the ratio of work output to the heat input.

Work output = $[RT1 \ln(V2/V1) + CP(T1-T4)] – [CP(T2-T3) + RT3 \ln(V3/V4)]$

Heat Input = $RT1 \ln(V2/V1) + CP(T1-T4)$

Efficiency = Work Output/Heat Input

After putting values of heat input and work output in the above formula, we get

$\eta = 1 – [(CP(T2-T3) + RT3 \ln(V3/V4))/(RT1 \ln(V2/V1) + CP(T1-T4))]$ … (1)

Since Ericsson cycle is a regenerative cycle hence heat rejected in process 2-3 is used for heat addition in process 4-1. It means CP(T2-T3) gets cancelled by CP(T1-T4) hence, we can replace these values by zero in equation (1).

Also, $V3/V4 = V2/V1$

Hence, new thermal efficiency (after solving equation (1))

$\eta = 1 – (T3/T1)$

Which is equal to Carnot Cycle efficiency.

1.2.3 ADVANTAGES

The advantage of the Ericsson cycle over the Carnot and Stirling cycles is its smaller pressure ratio for a given ratio of maximum to minimum specific volume with higher mean effective pressure.

1.2.4 DISADVANTAGES

The drawback to the external combustion Ericsson engine is that fuel and air enter the external combustor at ambient environmental temperature. The energy required to heat the combustion gases to the high cycle temperature is not available to the working fluid and is lost to the cycle.

1.2.5 APPLICATION

Ericsson cycle is practically used in hot air engines and running a ship. Since the medium used is gas and having lower thermal conductivity, the application is restricted.

1.3 BRAYSSON CYCLE

Braysson cycle is a hybrid gas turbine cycle, a combination of Brayton cycle and Ericsson cycle. This hybrid cycle is proposed based on the Brayton cycle at high temperature heat addition and at low temperature heat rejection on Ericsson cycle.

The conventional Braysson cycle has not found practical use due to the difficulty in achieving isothermal compression. To make its implementation a reality, the original cycle has been modified by incorporating regenerator and a cooler before the final compression process. Reheating was included for augmenting the power output. Expressions for energy efficiency and energy destruction for all the components are derived along with the energy and energy efficiencies of the complete cycle. The effects of maximum temperature, pressure ratio and number of compression stages on the cycle efficiencies have been evaluated. It has been found that the energy destruction in the combustion chamber and re-heater put together accounts for more than 55% of the total energy destruction. The cycle efficiency is maximum at an optimum pressure ratio which itself is found to be a function of maximum temperature in the cycle. The energy and energy efficiency of the cycle equals the efficiency of normal Braysson cycle at a much lower pressure ratio. The efficiency achieved through the modified cycle with 2 stages of compression is only 2.2% less than the efficiency through ideal isothermal compression for a pressure ratio of 3 and turbine inlet temperature of 1200 K.

A detailed parametric and optimization studies of reheat and regenerative Braysson cycle has been carried out. The effect of compressor and turbine inlet temperatures, temperature rise in a stage of multistage compression, individual component efficiencies and exit pressure of reheat turbine on the performance has been studied.

The effect of perfect cooling a regeneration leads to a gain of 7.4% in maximum energy efficiency and 20% in maximum power output. A computer programme has been developed to evaluate the optimum pressure ratio for minimum specific fuel consumption and maximum power output. It is interesting to note that the optimum pressure ratio for maximum power output and minimum specific fuel consumption are different and they vary by a wide margin. It has been further seen that this optimum pressure ratio is a function of turbine inlet temperature. A thermodynamic system will have degeneracy in operational effectiveness with the decrease in component efficiencies due to aging. Hence the variations of optimum pressure ratio with component efficiencies are also studied and reported in this work. To make the system economically viable, it has been recommended to design the system for the operating condition of minimum specific fuel consumption rather than for maximum power output.

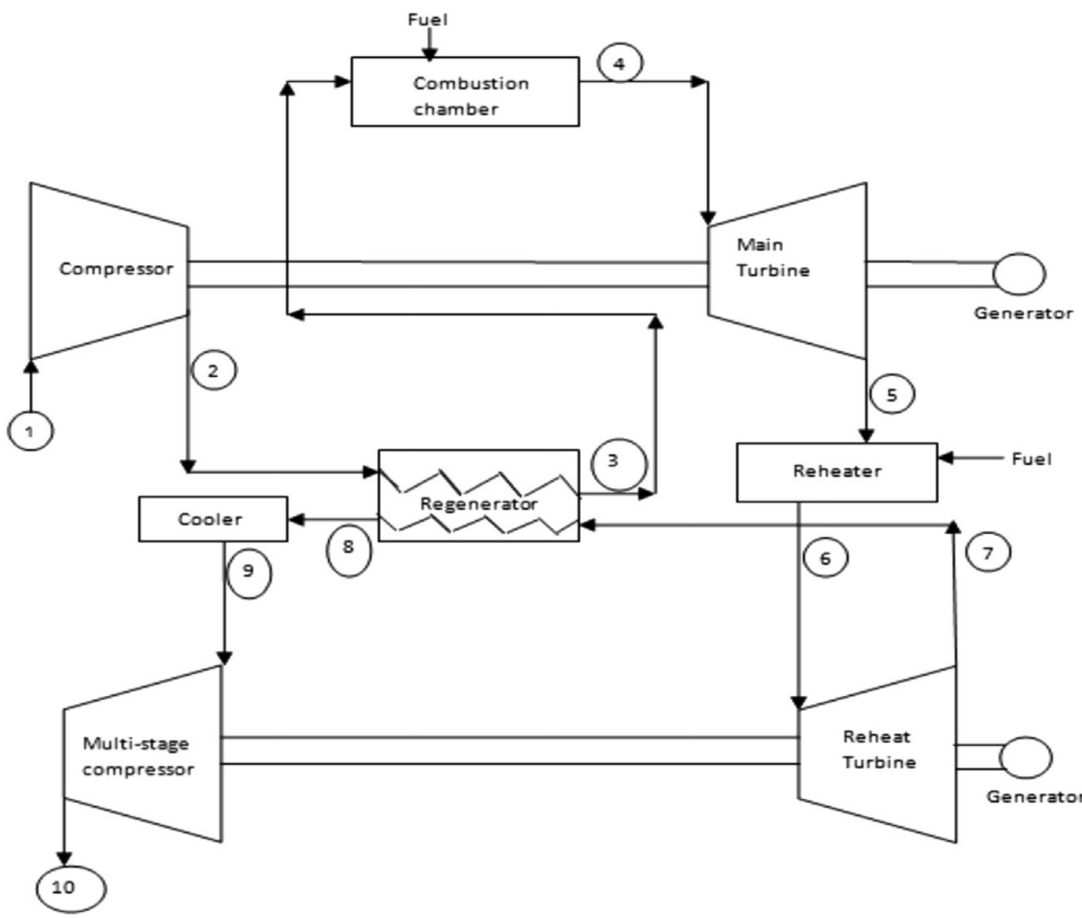

Figure 4: Schematic Diagram of Braysson Cycle with Reheat and Regeneration

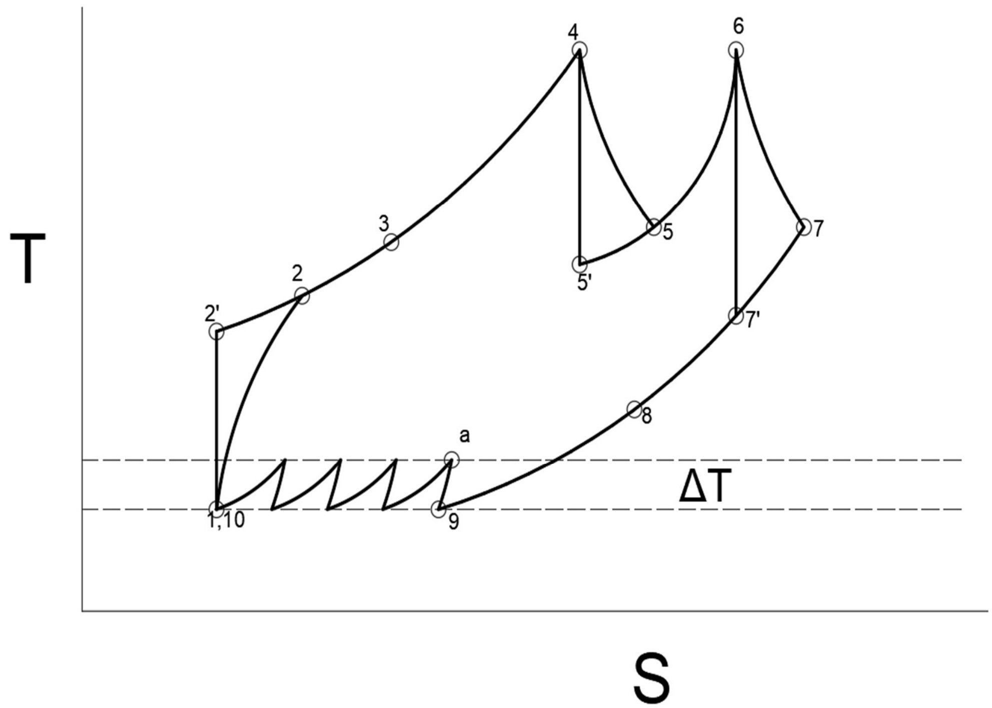

Figure 5: T-S Diagram of Braysson Cycle with Reheat and Regeneration

1.3.1 WORKING OF BRAYSSON CYCLE

Polytropic compression (1-2): The compression process has been accomplished using a rotodynamic compressor which is mechanically coupled to the main turbine.

Regeneration (2-3) & (7-8): Regeneration has been accomplished by using a counter flow heat exchanger. The heat of exhaust gases from the reheat turbine is transferred to the high-pressure air through regenerative heat transfer process. The process is assumed to be Isobaric.

Isobaric heat addition (3-4): The heat addition process occurs in the combustion chamber by the introduction of fuel through spray injectors.

Polytropic expansion (4-5) & (6-7): The expansion of working fluid takes place in the main and reheat gas turbines with intermediate reheat. The intermediate pressure at which reheating is carried out is optimized for maximum work output of the turbines. The downstream pressure of reheat turbine is considered as vacuum pressure.

Isobaric heat addition in reheater (5-6): The power developed during expansion by the gases operating between two pressure levels can be augmented by utilizing multistage expansion with reheating. It has been assumed that this process doesn't involve any pressure losses due to friction (or) turbulence.

Isobaric heat rejection (8-9): The gases from the regenerator are cooled in a heat exchanger before letting them into the multistage compressor to room temperature.

Isothermal heat rejection (9-10): The isothermal heat rejection has been achieved with the help of a multistage intercooled compressor, which is powered by the reheat turbine. The compression and heat rejection take place simultaneously and the pressure rises to atmospheric pressure.

The overall energy efficiency of reheat and regenerative Braysson cycle is given by the following formula

$$\eta_{en} = \frac{m_g c_p (T_4 - T_5) - m_a c_p T_1 \left[\left(r_p^K - 1\right)/\eta_c\right] + m_{g6} c_p (T_6 - T_7) - N m_{g6} c_p T_9 \left[\left(r_{p0}\right)^{K/N} - 1\right]/\eta_{cs}}{\text{l.c.v}\ (m_f + m_{f1})}$$

CHAPTER 2

LITERATURE SURVEY

2 LITERATURE SURVEY

A HYBRID GAS TURBINE CYCLE (BRAYTON/ERICSSON): AN ALTERNATIVE TO CONVENTIONAL COMBINED GAS AND STEAM TURBINE POWER PLANT

T H Frost, A Anderson and B Agnew

A hybrid gas turbine cycle is proposed based on the conventional Brayton cycle for the high- temperature heat addition process while adopting the Ericsson cycle for the low-temperature heat rejection process. It thus incorporates the thermodynamic advantages of a combined gas and steam turbine (CCGT) cycle without the irreversibility's of the boiler and the ancillaries of the steam turbine/condenser plant. Thermodynamic analysis shows that a similar overall thermal efficiency as current CCGT plant (i.e. 0.54) would be achieved at a maximum gas temperature of 1311 °C if polytropic efficiencies of 0.90 for com- pression and expansion could be realized and if a maximum temperature of 77 °C was obtained during iso- thermal compression in the bottoming Ericsson cycle.

RADIANTLY-HEATED BRAYTON ERICSSON CYCLE

Jon D. McWhirter

In addition to high temperatures of heat addition, the isothermal addition of heat to a heat engine tends to raise the efficiency. This is difficult to do with gases if conventional methods are used for adding heat and extracting work (i.e Combustion followed by work extraction). This paper proposes a way to accomplish the isothermal addition of heat by simultaneously extracting work. That is, adding the combustion heat to the working fluid while it is performing work in the turbine. To accomplish this task, the combustion heat is transferred radiantly from an external chamber through a radiantly transparent casing to the fixed and moving turbine blades. The moving working fluid then picks up this heat convectively from the turbine blades.

OPTIMIZATION OF AN AIR-COOLED HYBRID GAS TURBINE CYCLE (BRYATON/ERICSSON)

T H Frost, A Anderson, B Agnew and I Potts

The performance of a Braysson cycle, a hybrid gas turbine cycle, has been examined to establish the effect of air cooling and heat exchanger effectiveness on the cycle efficiency and specific power. The air-cooled heat exchanger was optimized to produce the maximum net efficiency for the specified minimum cycle temperature. The cycle performance was shown to be adversely influenced by the air cooling as it reduced both the specific power and efficiency. The heat exchanger effectiveness was shown to have a secondary impact on the performance parameters. An additional optimization of the heat exchanger at minimum volume is also presented to act as a benchmark against which the performance of the heat exchanger in the optimized cycle can be compared.

THERMODYNAMIC ANALYSIS OF A CLOSED BRAYTON/ERICSSON CYCLE ENGINE WITH SCROLL MACHINES

Young Min Kim

Stirling and Ericsson engines have great potential for many applications, including micro- cogeneration, solar power, and biomass. However, ideal cycles of both types of engines are difficult to achieve in practice because neither isothermal compression nor isothermal expansion is practical with reciprocating piston engines or with turbomachinery. On the other hand, scroll compressor and expander can be very suitable for effective cooling and heating because of the high area-to-volume ratio of scroll geometry or the application of two-phase °ow. To achieve quasi- isothermal compression, either a large amount of liquid is injected into the inlet of the compressor or the compressor is externally cooled by liquid. Similarly, for quasi-isothermal expansion, either hot liquid, such as thermal oil, is injected into the inlet of the expander or the expander is externally heated by a heat source. In this current study, we have undertaken a theoretical investigation of thermodynamic analyses of several kinds of scroll-type engines, in particular with regard to associated compression and expansion processes, adiabatic or quasi-isothermal pro- cesses, and the highest cycle temperature. We selected power density, or thermal efficiency, as an objective function, and then deduced optimal design parameters for the scroll-type engine.

SECOND LAW ANALYSIS OF AN IRREVERSIBLE BRAYSSON CYCLE

B. Sreenivas* R. Chandramouli I. Sudhakar, A.R. Thulasiram

This paper demonstrates the second-law analysis of an irreversible Braysson cycle. The overall second law efficiency for the process at variable operating conditions are being derived and plotted. In addition, for further pinpointing and thorough quantification of the losses, the component-wise second law efficiencies are also derived. It had been concluded that, the second law efficiency can be improved by changing the operating conditions of the cycle such as pressure, temperature, etc. In addition, a clear view of the thermodynamic losses occurring in each component is obtained and they too, can be minimised by tuning the operating conditions.

POWERS AND EFFICIENCY PERFORMANCE OF AN ENDO REVERSIBLE BRYASSON CYCLE

Junlin Zheng, Lingen Chen, Fengrui Sun, Chih Wu

Performance analysis of a Braysson cycle has been performed using entropy generation minimization or finite time thermodynamics. The analytical formula about power output and efficiency of an endo reversible Braysson cycle with heat resistance losses in the hot and cold-side heat exchangers are derived. The influences of the design parameters on the performance of the cycle are analyzed by detailed numerical examples.

MULTI-OBJECTIVE OPTIMIZATION AND EXERGETIC-SUSTAINABILITY OF AN IRREVERSIBLE NANO SCALE BRAYSSON CYCLE OPERATING WITH MAXWELL-BOLTZMAN GAS

Mohammad H.Ahmadi Mohammad AliJokar TingzhenMing MichelFeidt FathollahPourfayaz Fatemeh RaziAstaraei

Nano technology is developed in this decade and changes the way of life. Moreover, developing nano technology has effect on the performance of the materials and consequently improves the efficiency and robustness of them. So, nano scale thermal cycles will be probably engaged in the near future. In this paper, a nano scale irreversible Braysson cycle is studied thermo- dynamically for optimizing the performance of the Braysson cycle. In the aforementioned cycle an ideal Maxwell–Boltzmann gas is used as a working fluid. Furthermore, three different plans are used for optimizing with multi-objectives; though, the outputs of the abovementioned plans are assessed autonomously. Throughout the first plan, with the purpose of maximizing the ecological coefficient of performance and energy efficiency of the system, multi-objective optimization algorithms are used. Furthermore, in the second plan, two objective functions containing the ecological coefficient of performance and the dimensionless Maximum available work are maximized synchronously by utilizing multi-objective optimization approach. Finally, throughout the third plan, three objective functions involving the dimensionless Maximum available work, the ecological coefficient of performance and energy efficiency of the system are maximized synchronously by utilizing multi-objective optimization approach. The multi-objective evolutionary approach based on the non-dominated sorting genetic algorithm approach is used in this research. Making a decision is performed by three different decision makers comprising linear programming approaches for multidimensional analysis of preference and an approach for order of preference by comparison.

EFFECT OF TEMPERATURE AND GAS FLOW ON THE EFFICIENCY OF AN AIR BOTTOMING CYCLE

R. Teflissi *and* A. Ataei

The objective of this research is to analyze the performance of an Air Bottoming Cycle (ABC) by considering the effect of temperature and gas flow. Based on the energetic analysis, a thermodynamic calculation has been performed to investigate improving the thermal efficiency of the ABC. In this research, we consider the full cycle should consist of two sub-cycles: **Braysson** and Joule. The flue gas from the top cycle is used in Braysson cycle turbine to generate power and the rest of the energy of this flow is exchanged to the pressurized Joule cycle air flow. The results indicate that the higher inlet temperatures of the Braysson turbine and lower Braysson turbine outlet temperatures lead to higher thermal efficiencies. Moreover, the lower overall pressure ratios in Joule cycle are required to achieve the higher overall thermal efficiencies. The maximum thermal efficiency and also the optimal design point of this ABC have been found. The results of this ABC analysis show an increase in net output work and the thermal efficiency up to more than 55%, which in turn leads to have less specific fuel consumption. The proposed cycle has been found to have more energy recovery potential compared to some other gas turbine-based ABCs. Besides, in contrast to common steam bottom cycles, some water can be generated using the flue gas stream which is important because the lack of water resources is going to be a major problem in many regions worldwide.

OPTIMUM CRITERIA ON THE PERFORMANCE OF AN IRREVERSIBLE BRAYSSON HEAT ENGINE BASED ON THE NEW THERMOECONOMIC APPROACH

Sudhir Kumar Tyagi, Yinghui Zhou and Jincan Chen

An irreversible cycle model of a Braysson heat engine operating between two heat reservoirs is used to investigate the thermo economic performance of the cycle affected by the finite-rate heat transfer between the working fluid and the heat reservoirs, heat leak loss from the heat source to the ambient and the irreversibility within the cycle. The thermo economic objective function, defined as the total cost per unit power output, is minimized with respect to the cycle temperatures along with the isobaric temperature ratio for a given set of operating parameters. The objective function is found to be an increasing function of the internal irreversibility parameter, economic parameters and the isobaric temperature ratio. On the other hand, there exist the optimal values of the state point temperatures, power output and thermal efficiency at which the objective function attains its minimum for a typical set of operating parameters. Moreover, the objective function and the corresponding power output are also plotted against the state point temperature and thermal efficiency for a different set of operating parameters. The optimally operating regions of these important parameters in the cycle are also determined. The results obtained here may provide some useful criteria for the optimal design and performance improvements, from the point of view of economics as well as from the point of view of thermodynamics of an irreversible Braysson heat engine cycle and other similar cycles as well.

PERFORMANCE ANALYSIS AND OPTIMUM CRITERIA OF AN IRREVERSIBLE BRAYSSON HEAT ENGINE

YinghuiZhou S.K.Tyagi JincanChen

An irreversible cycle model of a Braysson heat engine operating between two heat reservoirs is used to investigate the performance of the cycle affected by the finite-rate heat transfer between the working fluid and the heat reservoirs, heat leak loss between the heat reservoirs and irreversibility inside the cycle. The specific power output is maximized with respect to the cycle temperatures along with the isobaric temperature ratio. The specific power output is found to be a decreasing function of the internal irreversibility parameter and isobaric temperature ratio while there exist the optimal values of the state point temperatures at which the specific power output attains its maximum value for a typical set of operating parameters. Moreover, the maximum specific power output and other cycle parameters are calculated for different sets of operating conditions. The optimally operating regions of the important parameters in the cycle are determined. The results obtained here may provide some useful criteria for the optimal design and performance improvement of a realistic Braysson heat engine.

THE PROCESS OF ISOTHERMAL COMPRESSION OF GASES AT SUB-ATMOSPHERIC PRESSURES THROUGH REGULATED WATER INJECTION IN BRAYSSON CYCLES

Demos P.Georgiou Triantafyllos Xenos

Although the Braysson cycle constitutes the ideal limit for the Combined Cycle Power Plants, its actual implementation has not been achieved due to the difficulty in building the required isothermal compressor. The present study proposes the incorporation of regulated water injection during the final compression, which could maintain the temperature constant due to the evaporation. The analysis for the thermodynamic implications of the injection on the ideal version of the Braysson cycle indicates that the (ideal cycle) efficiency reduction will be minimal. The study provides an analysis for the water injection rate that will permit such a process and shows that the additional work needed to drive the process will not be affected significantly by the injection. In addition, it shows that the minimum temperature of the Braysson cycle will be lower than the corresponding level of the conventional (Gas–Steam turbine Combined cycle plants), something that could improve the efficiency as well. Finally, it shows that the process may be expressed by a polytropic relationship of the type pv^{β} = constant, where $\beta \approx 1.06$.

THE EFFECTS OF A MULTI-STEP INTERCOOLED COMPRESSION PROCESS IMPLEMENTED ON A SOLAR-DRIVEN BRAYSSON HEAT ENGINE

D.P.Georgiou K.F.Milidonis E.N.Georgiou

The present study develops the thermodynamic analysis for the cycle of a solar-driven, Braysson cycle-based plant in the ideal limit and in the presence of process irreversibility's. The plant cycle differs from the conventional idealized Braysson cycle in that the implementation of the final isothermal compression process is substituted by a multistep intercooled compression. The cycle's efficiency is analytically formulated after taking into account several loss (irreversibility) sources such as the non-isentropic behaviour of the main compressor, the power turbine and the intercooled compressor stages as well as the actual heat transferred through counter-flow heat exchangers. All pressure losses associated with heat exchangers are related to the actual heat transfer load within each exchanger. The analysis develops a parametric evaluation for the effectiveness of the main cycle free variables on the thermal efficiency of the cycle. Such free variables include the working fluid maximum temperature, the compressor pressure ratio and the operating temperature limits of the intercooled compression stages, in addition to the polytropic coefficients of the compressor and power turbine (quasi-) isentropic processes. The results indicate that such a plant may reach efficiency levels above 30%, i.e. exceeding the efficiencies of the conventional Photovoltaic plants by a wide margin.

POWER, EFFICIENCY AND ECOLOGICAL OPTIMIZATION OF AN IRREVERSIBLE BRAYSSON HEAT ENGINE

Jizhou He; Xian He; Bei Yang

An irreversible Braysson heat engine cycle model, in which the heat transfer obeys a linear heat transfer law between the working fluid and the external heat reservoirs, with heat resistance, heat leak and internal irreversibility is established in this paper. The analytical expressions of power output, efficiency and ecological objective function are derived. These performance parameters are maximized with respect to the cycle temperatures along with the isobaric temperature ratio. The performance

characteristic curves are obtained by numerical calculation. The optimal operating regions of the important parameters are determined.

PARAMETRIC AND OPTIMIZATION STUDIES OF REHEAT AND REGENERATIVE BRAYSSON CYCLE

R.Chandramouli M.S.S.Srinivasa Rao K.Ramji

A detailed parametric and optimization studies of reheat and regenerative Braysson cycle has been carried out. The effect of compressor and turbine inlet temperatures, temperature rise in a stage of multi-stage compression, individual component efficiencies and exit pressure of reheat turbine on the performance has been studied. The effect of perfect cooling after regeneration leads to a gain of 7.4% in maximum exergy efficiency and 20% in maximum power output. A computer programme has been developed to evaluate the optimum pressure ratio for minimum specific fuel consumption and maximum power output. It is interesting to note that the optimum pressure ratio for maximum power output and minimum specific fuel consumption are different and they vary by a wide margin. It has been further seen that this optimum pressure ratio is a function of turbine inlet temperature. A thermodynamic system will have degeneracy in operational effectiveness with the decrease in component efficiencies due to aging. Hence the variations of optimum pressure ratio with component efficiencies are also studied and reported in this work. To make the system economically viable, it has been recommended to design the system for the operating condition of minimum specific fuel consumption rather than for maximum power output.

PARAMETRIC OPTIMIZATION OF A SOLAR-DRIVEN BRAYSSON HEAT ENGINE WITH VARIABLE HEAT CAPACITY OF THE WORKING FLUID AND RADIATION-CONVECTION HEAT LOSSES

LanmeiWu GuoxingLin JincanChen

An irreversible solar-driven Braysson heat engine system is presented, in which the temperature-dependent heat capacity of the working fluid, the radiation–convection heat losses of the solar collector and the irreversibility's resulting from heat transfer and non-isentropic compression and expansion processes are taken into account. Based on the thermodynamic analysis method and the optimal control theory, the mathematical expression of the overall efficiency of the system is derived and the maximum overall efficiency is calculated, and the operating temperatures of the solar collector and the cyclic working fluid and the ratio of heat-transfer areas of the heat engine are optimized. By using numerical optimization technology, the influences of the variable heat capacity of the working fluid, the radiation–convection heat losses of the solar collector and the multi-irreversibility's on the performance characteristics of the solar-driven heat engine system are investigated and evaluated in detail. Moreover, it is expounded that the optimal performance and important parametric bounds of the irreversible solar-driven Braysson heat engine with the constant heat capacity of the working fluid and the irreversible solar-driven Carnot heat engine can be deduced from the conclusions in the present paper.

ENERGY AND EXERGY BASED THERMODYNAMIC ANALYSIS OF REHEAT AND REGENERATIVE BRAYSSON CYCLE

R. Chandramouli, M.S.S. Srinivasa Rao, K. Ramji.

The conventional Braysson cycle has not found practical use due to the difficulty in achieving isothermal compression. To make its implementation a reality, the original cycle has been modified by incorporating regenerator and a cooler before the final compression process. Reheating was included for augmenting the power output. Expressions for exergy efficiency and exergy destruction for all the components are derived along with the energy and exergy efficiencies of the complete cycle. The effects of maximum temperature, pressure ratio and number of compression stages on the cycle efficiencies have been evaluated. It has been found that the exergy destruction in the combustion chamber and reheater put together accounts for more than 55% of the total exergy destruction. The cycle efficiency is maximum at an optimum pressure ratio which itself is found to be a function of maximum temperature in the cycle. The energy and exergy efficiency of the cycle equals the efficiency of normal Braysson cycle at a much lower pressure ratio. The efficiency achieved through the modified cycle with 2 stages of compression is only 2.2% less than the efficiency through ideal isothermal compression for a pressure ratio of 3 and turbine inlet temperature of 1200 K.

PERFORMANCE ANALYSIS OF AN ENDOREVERSSIBLE BRAYSSON CYCLE BASED ON ECOLOGICAL CRITERION

Yasin ust, Tamer Yilamz

A performance analysis based on the ecological criterion was carried out for an endoreversible Braysson cycle model that includes finite rate heat transfer irreversibility. The ecological objective function is defined as the power output minus the loss power, which is equal to the product of the environmental temperature and the entropy production rate. The maximization of the ecological function was achieved for various design parameters and the obtained results are compared with those obtained using the maximum power criterion.

EXERGY ANALYSIS FOR A BRAYSSON CYCLE

Zheng, J., Sun, F., Chen, L. and Wu, C

An exergy analysis has been carried out for an irreversible Braysson cycle. The analytical formulae of power output and exergy efficiency are derived. The influences of various parameters on the exergy performance are analysed by numerical calculation, and the results obtained have been compared with those of Brayton cycle under the same conditions. It is shown that the exergy loss in the combustion is the largest in the Braysson cycle, and both specific work and exergy efficiency of the cycle are larger than those of Brayton cycle.

THERMODYNAMIC ANALYSIS AND PARAMETRIC STUDY OF AN IRREVERSSIBLE REGENERATIVE INTERCOOLED REHEAT BRAYTON CYCLE

S. K. Tyagi, G. M. Chen, Q. Wang, S. C. Kaushik

An irreversible cycle model of a regenerative-intercooled-reheat Brayton heat engine along with a detailed parametric study. The power output and the efficiency are optimized with respect to the cycle temperatures for a typical set of operating conditions. It is found that there are optimal values of the turbine outlet temperature, inter- cooling, reheat and cycle pressure ratios at which the cycle attains the maximum power output and efficiency. But the optimal values of these parameters corresponding to the maximum power output are different from those corresponding to the maximum efficiency for the same set of operating condition. The maxima of the power output and efficiency again changes as any of the cycle parameters is changed. The maximum power point and the maximum efficiency point exist but the power output corresponding to the maximum efficiency is found to be lower than that can be attained. The optimum operating parameters, such as the turbine outlet temperature, inter-cooling, reheat and cycle pressure ratios etc. corresponding to the maximum power output and corresponding to the maximum efficiency are obtained and discussed in detail. This cycle model is general and some of the results obtained by earlier workers can be derived directly from the present cycle model as a special case.

CHAPTER 3

METHODOLOGY

3 METHODOLOGY

We have used MATLAB to see the effects of different component efficiencies on the overall energy efficiency of reheat and regenerative Braysson cycle.

All of the following equations were used and put in the form of a MATLAB code and some assumptions were also made for some values.

The number of stages of multi-stage compression used are 4 for our project, that is, N=4.

All the component efficiencies have been varied individually while keeping the other component efficiencies as a constant.

$$\eta_c=0.9,\ \eta_{cs}=0.9,\ \eta_T=0.9,\ \eta_{reg}=0.85$$

Assuming the mass flow rate of air (ma) ¼ 1 kg/sec, the actual temperature of air at the exit of a compressor is

$$T_2 = T_1 + \frac{T_1 \cdot \left[r_p^{(\gamma-1)/\gamma} - 1\right]}{\eta_c} ----------\rightarrow Eq.(1)$$

The first law efficiency of a regenerator is defined as the ratio of actual heat transfer to the maximum possible heat transfer.

$$\eta_{\text{reg}} = \frac{H_3 - H_2}{H_7 - H_2} ----------\rightarrow\ Eq.(2)$$

Assuming specific heat of air and gases as 1.005 kJ/kg K, the temperature of air at the outlet of regenerator after simplification of is given by

$$T_3 = \eta_{reg} m_{g6}(T_7 - T_2) + T_2 ----------\rightarrow Eq.(3)$$

where, mg6 is the mass flow rate of gases flowing through the reheat turbine.

$$m_{g6} = m_a + m_f + m_{f1} = 1 + m_f + m_{f1} ----------\rightarrow Eq.(4)$$

The temperature at the exit of reheater is equal to the maximum temperature in the cycle, i.e.,

$$T_4 = T_6 = T_{max} ----------\rightarrow Eq.(5)$$

The intermediate pressure P3 between the two stages of expansion has been evaluated based on the condition that the gross work output of the turbines is maximum, and is determined as

$$P_3 = \left\{ \left(\frac{m_g6}{m_g} \right)^{\frac{\gamma}{2(\gamma-1)}} \right\} \sqrt{P_2 P_4} ----------\rightarrow Eq.(6)$$

where mg is the mass flow rate of gases flowing through the main turbine,

$$m_g = m_a + m_f = 1 + m_f ----------\rightarrow Eq.(7)$$

P3 equation can be rewritten in terms of pressure ratios of the main and the reheat turbines as

$$r_{p2} = r_{p1} \left[m_{g6}/m_g \right]^{\gamma/2(\gamma-1)} ----------\rightarrow Eq.(8)$$

The temperature of gases at the exit of main turbine is given as

$$T_5 = T_4 \left[1 - \eta_T \left(1 - r_{p1}^{(1-\gamma)/\gamma} \right) \right] ----------\rightarrow Eq.(9)$$

Similarly, the temperature of gases at the exit of reheat turbine is given as

$$T_7 = T_6 \left[1 - \eta_T \left(1 - r_{p2}^{(1-\gamma)/\gamma} \right) \right] ----------\rightarrow Eq.(10)$$

Substituting for T3 from and ma = 1 kg/s and rearranging the terms, we have

$$\begin{aligned} &m_f \{ \eta_{reg} \cdot c_p (T_7 - T_2) - c_p T_4 + l \cdot c \cdot v \} + m_{f1} \cdot \eta_{reg} c_p (T_7 - T_2) \\ &= c_p (T_4 - T_2) - \eta_{reg} \cdot c_p (T_7 - T_2) \end{aligned} --\rightarrow Eq.(11)$$

On simplification, the temperature of gases at the exit of regenerator is given as

$$T_8 = T_7 - \left(\frac{1}{(1 + m_f + m_{f1})} \right) (T_3 - T_2) ----------\rightarrow Eq.(12)$$

Consider an intermediate stage of multi-stage compression in which the temperature at the end of actual compression is Ta and at the end of isentropic compression is Ta’.

$$\frac{T_{a'}}{T_9} = r_{pi}^{(\gamma-1)/\gamma} ----------\rightarrow Eq.(13)$$

The isentropic efficiency of any stage is given as:

$$\eta_c = \frac{T_{a'} - T_9}{T_a - T_9} ----------\rightarrow Eq.(14)$$

The rate of heat supplied to the cycle consists of heat input in the combustion chamber and reheater. It is given by the expression:

$$Q = 1.c.v\left(m_f + m_{f1}\right) ----------\rightarrow Eq.(15)$$

The overall energy efficiency of the reheat and regenerative Braysson cycle is given by:

$$\eta_{\text{en}} = \frac{\text{Net rate of work output of the cycle}}{\text{rate of heat supplied to the cycle}}$$

$$\eta_{\text{en}} = \frac{m_g c_p (T_4 - T_5) - m_a c_p T_1\left[\left(r_p^K - 1\right)/\eta_c\right] + m_{g6} c_p (T_6 - T_7) - N m_{g6} c_p T_9\left[\left(r_{p0}\right)^{K/N} - 1\right]/\eta_{cs}}{\text{l.c.v}\left(m_f + m_{f1}\right)} ----------\rightarrow Eq.(16)$$

CHAPTER 4
RESULTS

4 RESULTS

From Table 1, it can be seen that as the inlet temperature to the gas turbines is enhanced, there is an appreciable increase in the energy efficiency.

As the maximum temperature rises from 800K to 1200K the energy efficiency increases by about 13.31%. This efficiency represents the maximum value at the corresponding temperatures as they have been evaluated at optimum pressure ratio.

Turbine Inlet Temperature	Optimum pressure ratio	Energy Efficiency
800K	2	38.96%
900K	2.3	43.27%
1000K	2.6	46.79%
1100K	2.9	49.75%
1200K	3.2	52.27%

Table 1: Energy efficiency of the cycle at optimum pressure ratio for different maximum turbine inlet temperatures (N=4)

The effect of each component efficiency on the performance of the reheat and regenerative Braysson cycle is discussed in the following graphs.

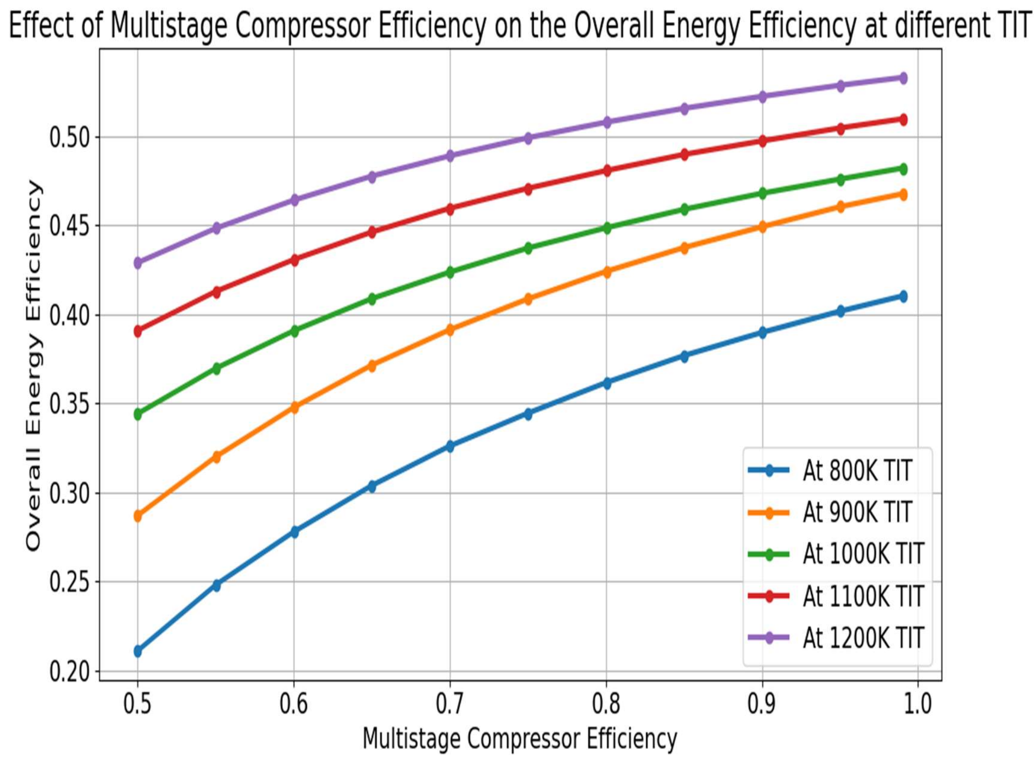

Figure 6: Effect of Multistage Compressor Efficiency on the Overall Efficiency at different TIT

We've observed that at a given multistage compressor efficiency, the overall energy efficiency increases with increase in TIT. At a given TIT, as the multistage compressor Efficiency increases, an increase in the overall energy efficiency is observed.

At a given TIT, initially the overall energy efficiency increases at a faster rate with respect to the multistage compressor efficiency and the curve tends to increase at a slower rate after 0.8 multistage compressor efficiency.

The minimum value of overall energy efficiency (0.21) is observed at a multistage compressor efficiency of 0.5 at 800K TIT.

The maximum value of overall energy efficiency (0.54) is observed at a multistage compressor efficiency of 0.99 at 1200K TIT.

All the curves in the graph follow an increasing trend and are of similar shape separated by a distance due to different TIT's and different pressure ratios. The distance between the curves reduces with increase in the multistage compressor efficiency.

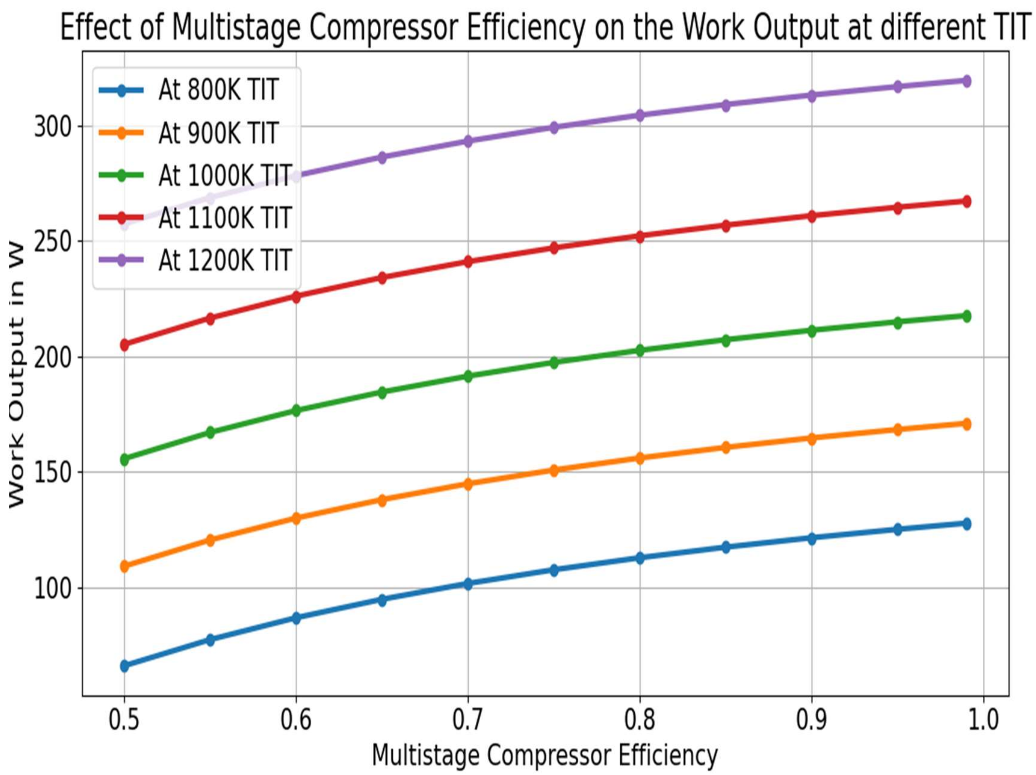

Figure 7: Effect of Multistage Compressor Efficiency on the Work Output at different TIT

We've observed that at a given multistage compressor efficiency, the work output increases with increase in TIT. At a given TIT, as the multistage compressor efficiency increases, an increase in the work output is observed.

At a given TIT, initially the work output increases at faster rate with respect to the multistage compressor efficiency and the curve increases at a slower rate from 0.75 multistage compressor efficiency.

The minimum value of work output (25W approx.) is observed at a multistage compressor efficiency of 0.5 at 800K TIT.

The maximum value of work output (315W approx.) is observed at a multistage compressor efficiency of 0.99 at 1200K TIT.

All the curves in the graph follow an increasing trend and are of similar shape separated by a distance due to different TIT's and different pressure ratios.

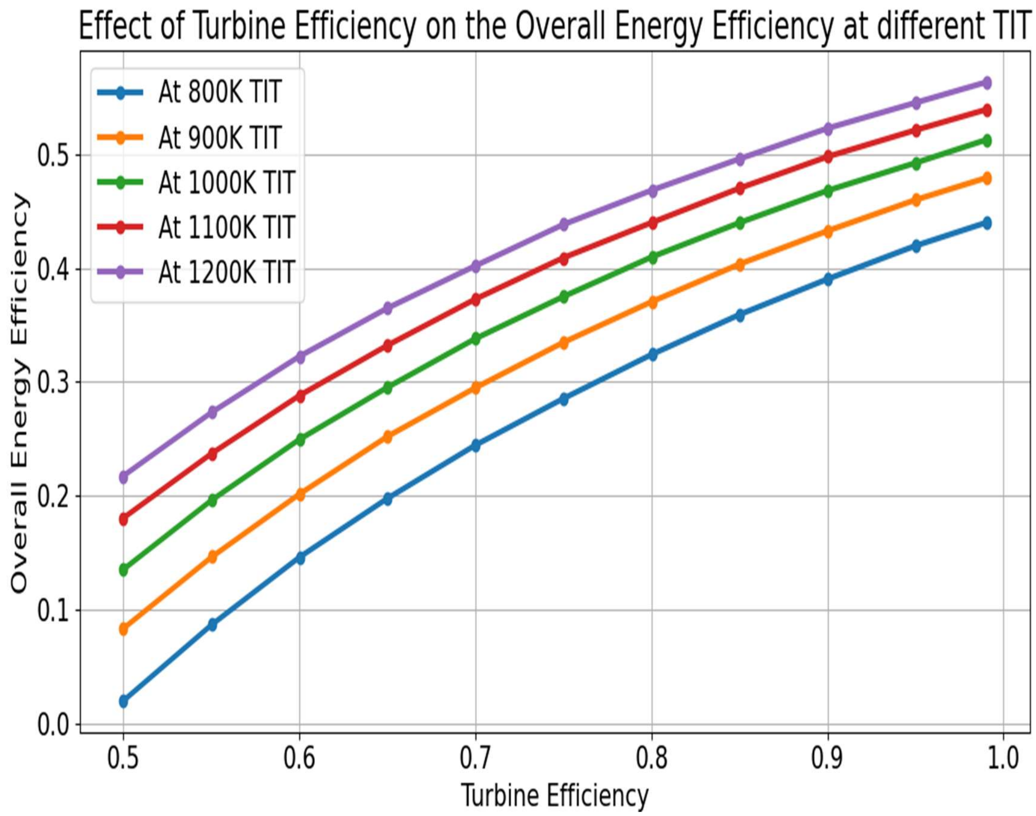

Figure 8: Effect of turbine efficiency on the overall energy efficiency at different TIT

We've observed that at a given turbine efficiency, the overall energy efficiency increases with increase in TIT. At a given TIT, as the turbine Efficiency increases, an increase in the overall energy efficiency is observed.

At a given TIT, initially the overall energy efficiency increases at a faster rate with respect to the turbine efficiency and the curve tends to increase at a slower rate after 0.8 turbine efficiency.

The minimum value of overall energy efficiency (0.02) is observed at a turbine efficiency of 0.5 at 800K TIT.

The maximum value of overall energy efficiency (0.56) is observed at a turbine efficiency of 0.99 at 1200K TIT.

All the curves in the graph follow an increasing trend and are of similar shape separated by a distance due to different TIT's and different pressure ratios.

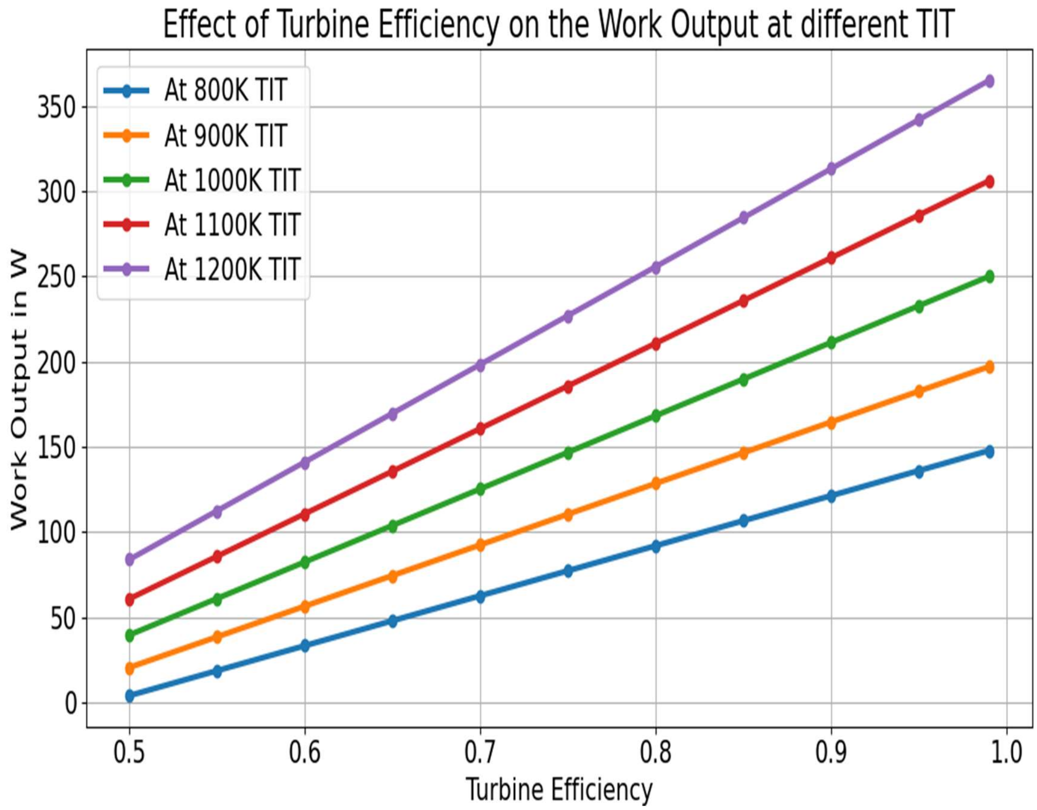

Figure 9: Effect of Turbine Efficiency on the Work Output at different TIT

We've observed that at a given turbine efficiency, the work output increases with increase in TIT. At a given TIT, as the turbine efficiency increases, an increase in the work output is observed.

The minimum value of work output (1W approx.) is observed at a turbine efficiency of 0.5 at 800K TIT.

The maximum value of work output (365W approx.) is observed at a turbine efficiency of 0.99 at 1200K TIT.

All the curves in the graph follow an increasing trend and the distance between them increases as the turbine efficiency increases for increasing TIT i.e. the slope of the lines increases with increase in TIT.

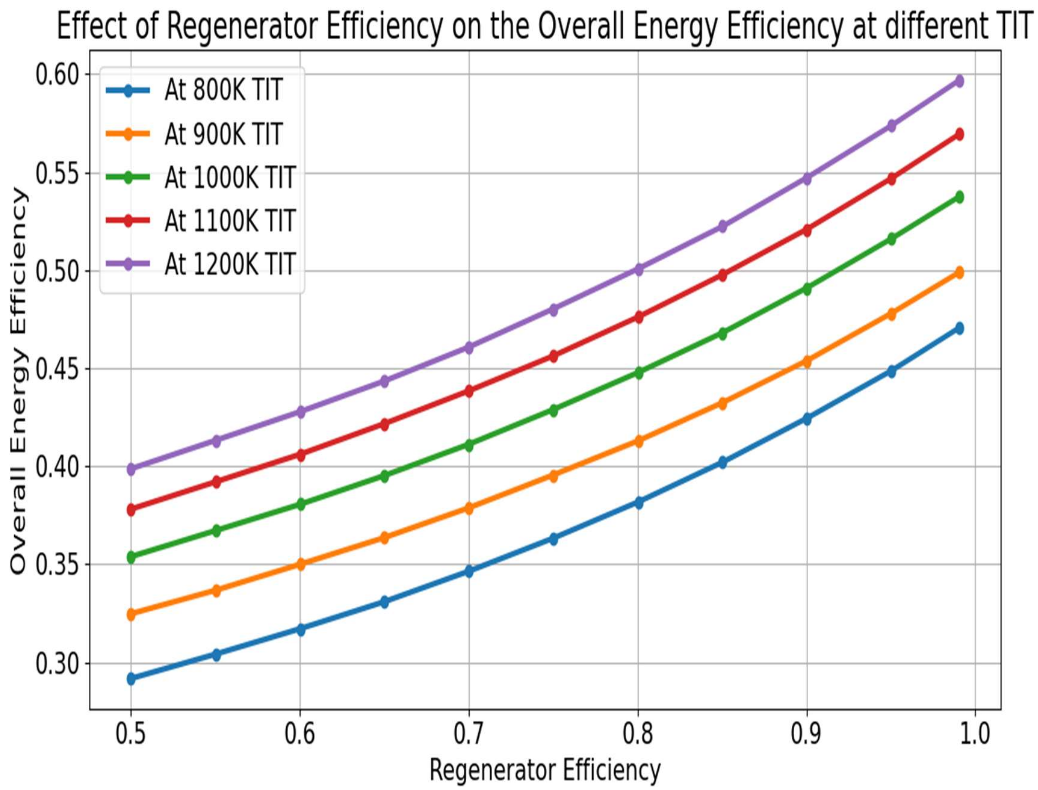

Figure 10: Effect of Regenerator Efficiency on the Overall Energy Efficiency at different TIT

We've observed that at a given regenerator efficiency, the overall energy efficiency increases with increase in TIT. At a given TIT, as the regenerator Efficiency increases, an increase in the overall energy efficiency is observed.

At a given TIT, initially the overall energy efficiency increases at a slower rate with respect to the regenerator efficiency and the curve tends to increase at a faster rate after 0.8 regenerator efficiency.

The minimum value of overall energy efficiency (0.28) is observed at a regenerator efficiency of 0.5 at 800K TIT.

The maximum value of overall energy efficiency (0.60) is observed at a regenerator efficiency of 0.99 at 1200K TIT.

All the curves in the graph follow an increasing trend and are of similar shape separated by a distance due to different TIT's and different pressure ratios.

Figure 11: Effect of Regenerator Efficiency on the Work Output at different TIT

We've observed that at a given regenerator efficiency, the work output increases with increase in TIT. At a given TIT, as the regenerator efficiency increases, a slight decrease in the work output is observed, the work output is a straight line with a slight decreasing slope i.e. having a negative slope.

The minimum value of work output (120W approx.) is observed at a compressor efficiency of 0.5 at 800K TIT.

The maximum value of work output (320W approx.) is observed at a compressor efficiency of 0.5 at 1200K TIT.

All the curves in the graph follow a decreasing trend and are of similar shape separated by a distance due to different TIT's and different pressure ratios.

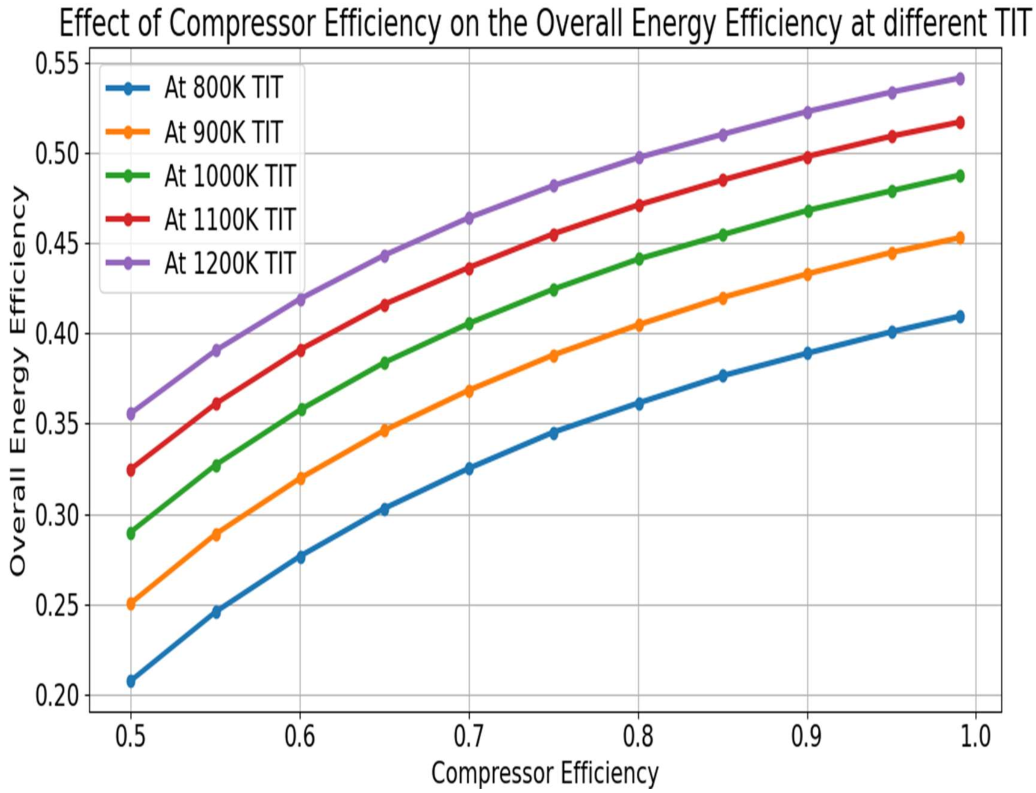

Figure 12: Effect of Compressor Efficiency on the Overall Energy Efficiency at different TIT

We've observed that at a given compressor efficiency, the overall energy efficiency increases with increase in TIT. At a given TIT, as the compressor Efficiency increases, an increase in the overall energy efficiency is observed.

At a given TIT, initially the overall energy efficiency increases at a faster rate with respect to the compressor efficiency and the curve tends to increase at a slower rate after 0.8 compressor efficiency.

The minimum value of overall energy efficiency (0.21) is observed at a compressor efficiency of 0.5 at 800K TIT.

The maximum value of overall energy efficiency (0.54) is observed at a compressor efficiency of 0.99 at 1200K TIT.

All the curves in the graph follow an increasing trend and are of similar shape separated by a distance due to different TIT's and different pressure ratios.

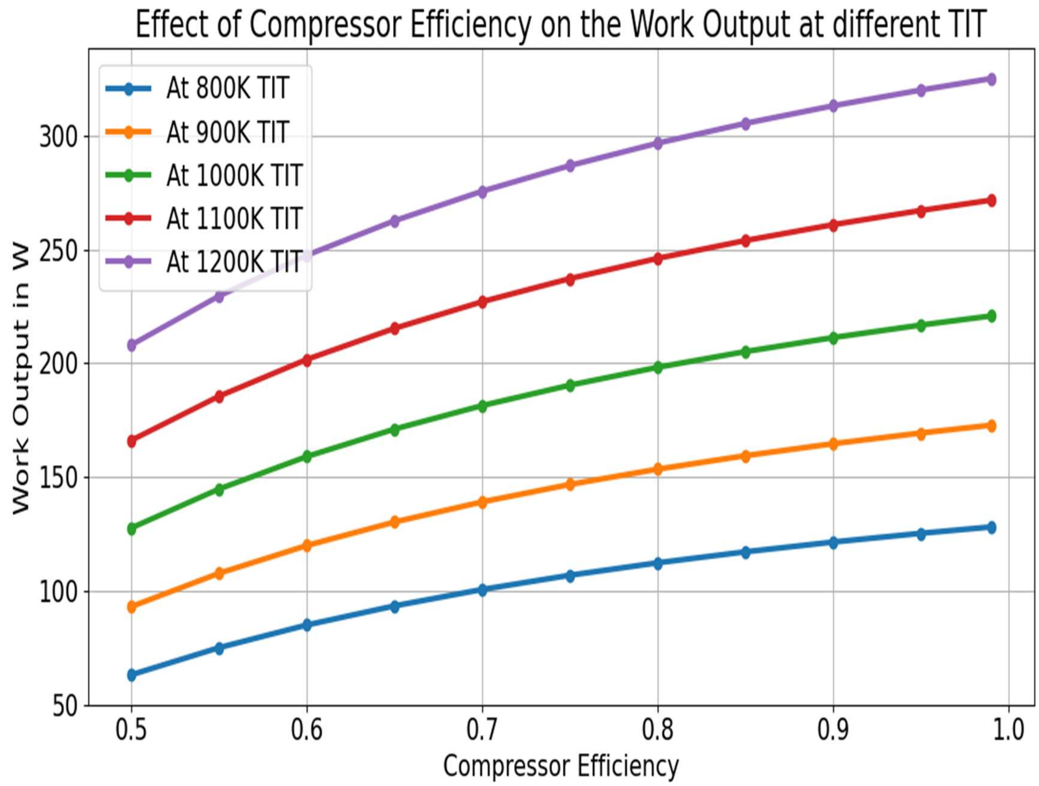

Figure 13: Effect of compressor efficiency on the work output at different TIT

We've observed that at a given compressor efficiency, the work output increases with increase in TIT. At a given TIT, as the compressor efficiency increases, an increase in the work output is observed.

At a given TIT, initially the work output increases at faster rate with respect to the compressor efficiency and the curve increases at a slower rate from 0.75 compressor efficiency.

The minimum value of work output (65W approx.) is observed at a compressor efficiency of 0.5 at 800K TIT.

The maximum value of work output (325W approx.) is observed at a compressor efficiency of 0.99 at 1200K TIT.

All the curves in the graph follow an increasing trend and are of similar shape separated by a distance due to different TIT's and different pressure ratios.

CHAPTER 5
CONCLUSIONS

5 CONCLUSIONS

Effect of component efficiencies on the performance of the reheat and regenerative Braysson cycle is analysed in this work. The following important conclusions are made:

- It can be observed that both the energy efficiency and the work output of the combined cycle increase with increase in pressure ratio, then reach an optimum value and then decreases.
- Both the energy efficiency and the work output of the combined cycle increases with increase in maximum cycle temperature value.
- The number of stages in the multistage compressor is fixed to 4.
- The performance of the combined cycle improves as the component efficiency increases except in the case of regenerator, as the regenerator effectiveness increases, there is a slight decrease in the work output.

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APPENDIX

MATLAB CODE

```
%for t6=1550:10:1600
%    t6
i=1;
rp=2;
for etareg=0.5:0.0001:0.99
%for rp=1.5:.1:20
   rp;
%rp=4.5;
eof=47346;
    cp=1.005;
    %cv=42000;
    cv=44427;
    p2=rp*1.013;
p4=0.5;
     rp1=sqrt(rp*1.013/p4);
     N=4;
t6=800;
t0=298;
t4=t6;
t9=t0;
t1=t0;
gamma=1.4;
a=(gamma-1)/gamma;
etacomp=0.9;
etamcomp=0.9;
etatur=0.9;
%etareg=0.85;
t2=((((rp^a-1)*t1)/etacomp)+t1);
ma=1;
t5=((1-(((rp1^a-1)/rp1^a)*etatur))*t4);
```

```
t7=t5;
t71=0;
t51=0;
err1=t7-t71;
err2=t5-t51;
if err1>0.0001 & err2>0.0001
   t71=t7;
   t51=t5;
k1=cp*(etareg*(t7-t2)-t4)+cv;
k2=etareg*cp*(t7-t2);

c1=cp*(t4-t2)-etareg*cp*(t7-t2);
k3=cp*(t6-t5);
k4=cv-cp*t6;
c2=cp*(t5-t6);
mf1=(k3*c1-k1*c2)/(k2*k3+k1*k4);
mf=(c2+mf1*k4)/k3;
mg=mf+ma;
mg6=mg+mf1;
p3=(mg6/mg)^(1/(2*a))*sqrt(p2*p4);
rp1=p2/p3;
t5=((1-(((rp1^a-1)/rp1^a)*etatur))*t4);
rp2=p3/p4;
t7=((1-(((rp2^a-1)/rp2^a)*etatur))*t4);
err1=t7-t71;
err2=t5-t51;

end
t3=((1+mf)*cp*t4-mf*cv)/cp;
t8=t7-(ma/(ma+mf+mf1))*(t3-t2);
rp0=(rp1*rp2)/rp;
num=((t4-(t0*log(t4/t0)))*cp*mg);
den=((cp*ma*(t3-(t0*log(t3/t0))))+(eof*mf));
eta2cc=num/den;
```

```
edcc=den-num;

num1=((cp*t6)-(t0*cp*log(t6/t0)))*(mg+mf1);
den1=cp*mg*(t5-(t0*log(t5/t0)))+(eof*mf1);
eta2reh=num1/den1;
edreh=den1-num1;

eta2comp=(1-((t0*(log(t2/t1)-(a*(log(rp)))))/(t1*(((rp^a)-1)/etacomp))));
edcom=ma*cp*t0*(log(t2/t1)-(a*log(rp)));

den2=(t4-t5)-(t0*(log(t4/t5)-(a*log(rp1))));
num2=(t4-t5);
eta2tur1=num2/den2;
edtur1=cp*mg*(den2-num2);

den21=(t6-t7)-(t0*(log(t6/t7)-(a*log(rp2))));
num21=(t6-t7);
eta2tur2=num21/den21;
edtur2=cp*mg6*(den21-num21);

edcooler=mg6*cp*((t8-t9)-t0*log(t8/t9));

eta2reg=(ma/mg6)*(((t3-t2)-(t0*log(t3/t2)))/((t7-t8)-(t0*log(t7/t8))));
exsupp=((t7-t8)-(t0*log(t7/t8)))*mg6;
exrecov=ma*((t3-t2)-(t0*log(t3/t2)));
edreg=cp*(exsupp-exrecov);

edmulc=cp*N*mg6*t0*(log((((rp0^(a/N))-1)/etamcomp)+1)-((a/N)*log(rp0)));
%edmulc=((N*cp*mg6)/etamcomp)*(((rp0^(a/N))-1)*t9)

ta=t9*(((rp0^(a/N)-1)/etamcomp)+1);
edintercooler=cp*(N-1)mg6((ta-t9)-t0*log(ta/t9));

toted=edmulc+edreg+edtur2+edtur1+edcom+edreh+edcc+edcooler+edintercooler;
```

```
%toted=edmulc+edreg+edtur2+edtur1+edcom+edreh+edcc+edcooler

A=ma*(t1*cp*(((rp^a)-1)/etacomp));
B=((N*cp*mg6)/etamcomp)*(((rp0^(a/N))-1)*t9);
C=mg*cp*(t4-t5);
D=mg6*cp*(t6-t7);
deno=eof*(mf+mf1);
numo=C-A+D-B;

overalleff=numo/deno
eta1cyc=numo/(cv*(mf+mf1));

edoverall=deno-numo;
%plot(rp,eta1cyc,'-');
kk1(i,1)=etareg;
kk2(i,1)=overalleff;

i=i+1
%plot(rp,eta2reg,'-m')

%plot(rp,eta2tur1,'-')
%hold on
%plot(rp,eta2tur2,'y-')
%hold on
%plot(rp,eta2comp,'g-')
%hold on
%plot(rp,eta2reh,'r-')
%hold on
%plot(rp,eta2cc,'k-')
%hold off
end
plot(kk1,kk2,'-k')

%end
```