School of Information Technology and Engineering at the ADA University

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SELECTION OF TURBINE INTEGRATED GAS COMPRESSOR FOR TURBINE

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THESIS ACCEPTANCE

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ABSTRACT

The growing demand for high-efficiency power generation, together with reduced fuel consumption and environmentally sustainable energy systems, focuses attention on improving gas turbine performance through advanced thermodynamic modeling and component integration. Among the most critical elements of any gas turbine are the compressor and turbine, whose aerodynamic and thermal behavior largely determines the overall cycle efficiency, work output, and operational reliability. This thesis concerns the thermodynamic integration of a three-stage axial gas compressor with a single expansion turbine, based on the development of an integrated analytical and computational framework for studying its performance, power balance, and efficiency characteristics under realistic industrial conditions. The compressor in this work is modeled stage by stage, including such details as interstage temperature rise, pressure development, work input, and isentropic versus polytropic efficiency. The turbine model assumes realistic expansion processes and takes into consideration turbine inlet temperature and energy balance in the calculation of work output as well as exhaust temperature and pressure. The model also contains a heat exchanger that recuperates thermal energy from turbine exhaust to preheat compressor fuel before entering the combustion chamber. Given such a configuration, a more realistic simulation can be achieved regarding how waste heat from modern industrial gas turbine systems is utilized to gain higher cycle efficiency. A computational platform with Python, including the full integration model, has been developed. It iterates the compressor and turbine performances over a wide range of operating conditions. The results obtained for the three-stage compressor show high values of cumulative temperature rise and specific work requirements, reflecting its sensitivity to interstage efficiency and pressure ratio distribution. One realizes the importance of distinguishing polytropic from isentropic efficiency, since the former is a direct measure of stage-by-stage compression behavior, while the latter gives an extended thermodynamic comparison. Then, on the turbine side, the model showed that turbine expansion efficiency, turbine inlet temperature, and exhaust pressure are each strong influences on the ability of the turbine to drive both the compressor and deliver useful shaft power. The sensitivity analysis underlines the tradeoffs between compressor work and turbine output that must be made-a balance that essentially defines whether the integrated system can function.

In this paper, a comprehensive thermodynamic analysis of compressor–turbine matching has been carried out, pointing out the influence of the choice of pressure ratio, stage efficiency, combustion heating, and exhaust conditions on the performance of an integrated

gas turbine. The result of the present work provides valuable insight into the design and optimization of industrial gas turbines, together with a computational framework that is easily extended to more complex multi-stage, multi-spool, or cooled turbine configurations. This thesis finally enhances the understanding of interdependent behavior that exists between compressors and turbines and provides a scientifically grounded approach to predict performance in real-world gas turbine applications.

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LIST OF ABBREVIATIONS

Abbreviation	Explanation
PR	Pressure Ratio
LHV	Lower Heating Value
TIT	Turbine Inlet Temperature
CFD	Computational Fluid Dynamics
IGV	Inlet Guided Vane

CHAPTER ONE

1. INTRODUCTION

The applications of gas turbines range over central roles in industrial power generation, mechanical drive applications, and large-scale energy conversion systems. As in the working of a Brayton cycle, the three consecutive processes of air compression, fuel combustion, and expansion of hot gases are utilized in order to extract mechanical power from the working medium. The compressor and turbine are the two most influential components that dictate the performance and efficiency, besides determining the reliability and operational feasibility of any such thermodynamic cycle. With ever-increasing demands from industries for better fuel economy, low emissions, and higher ratings, the need for better modeling and optimization of compressor-turbine interaction has assumed increasing dimensions of importance.

Industrial gas turbines, and in particular those serving petrochemical plants, refineries, offshore platforms, and pipeline systems, are normally equipped with multistage axial compressors capable of a very high-pressure ratio at acceptable polytropic efficiencies. These compressors quite often operate under very difficult conditions, both environmentally and thermally, where sophisticated engineering is essential for stability, adequate surge margin, and effective integration with the rest of the turbine system. On the expansion side, turbines utilize the combustion gases at high temperature to do work-not only in the form of output shaft power for industrial use but also with the purpose of driving the compressor itself, which consumes a large portion of the turbine output. This delicate balancing between compressor work requirements and turbine work availability is fundamental to the successful operation of any gas turbine. The thermodynamic behavior of the multistage compressor, usually coupled with a turbine, needs an in-depth analytical model that can precisely capture interstage temperature rise, accumulation of work input, efficiencies, and other pressure changes from stage to stage. The turbine too must be characterized by the energy it can extract from combustion gases while keeping exhaust pressure and temperature within acceptable bounds and yielding reasonable system efficiency. Therefore, the three-stage compressor model represents a good balance between a highly simplified configuration and one that's meaningful enough to approximate the behavior of modern axial compressors well, and hence this model can be effectively used for studying compressor and turbine matching, performance prediction, and overall cycle efficiency. This thesis presents a computationally driven thermodynamic model that simulates a three-stage compressor integrated with a single expansion turbine. The model analyzes stage-by-stage compression, combustion heating, turbine expansion, energy balance of the system, calculation of polytropic and isentropic efficiencies for both compressor and turbine, showing how the losses accumulate, the work requirement evolves, and how design decisions affect the performance of a gas turbine. A simulation provides strong predictive power, diminishes the need to create physical prototypes, and allows engineering analysis crucial for industrial applications.

1.1 Problem Statement

Although a vast amount of literature exists on gas turbine design and performance analysis, there still exists a big gap between simplified cycle calculations and detailed component-based aerodynamic simulations. In practice, designers are often obliged to make use of generalized compressor and turbine maps, which, though useful as a representation, cannot capture the thermodynamic interactions between successive stages in detail. For the design or analysis of a high-pressure industrial compressor, the engineer must have an idea about, among other things, temperature increase and work distribution at each stage, besides the total pressure ratio of a stage. Failure to account accurately for these parameters can lead to erroneous performance estimates, insufficient turbine power output, thermal inefficiency, or even operational instability. This gets worse in integrated systems: the power requirement of the compressor increases steeply with pressure ratio, while the power supply from the turbine is particularly sensitive to combustion temperature, expansion ratio, and internal losses. In the absence of a detailed analytical model, one can only speculate whether the turbine will satisfactorily support the compressor over a range of loads, much less under different ambient conditions. The engineers also risk underestimating the impact of stage efficiencies, cooling needs, and exhaust pressure on overall cycle performance. Therefore, the core problem targeted by this thesis is that no structured stage-focused gas turbine integration model, detailing thermodynamic behavior for a three-stage axial compressor with its turbine, exists. The research shall fill this lacuna by developing, in a clear and step-by-step analytic framework using Python, the capability to calculate cycle efficiency, energy balance, component performance, and temperature/pressure distribution across the entire system.

1.2 Definition of Terms:

To provide clarity throughout the thesis the following key thermodynamic and engineering terms are defined below.

Brayton Cycle: Ideal thermodynamic cycle for gas turbine operation, consisting of compression, constant-pressure heat addition and expansion.

Isentropic Efficiency: A measure comparing actual compressor or turbine performance to an ideal isentropic process with no entropy generation. For compressors, the ratio of ideal to actual work input; for turbines, the ratio of actual to ideal work output.

Polytropic Efficiency: An incremental measure of efficiency applied over infinitesimally small compression or expansion steps. This is usually applied to the definition of multi-stage compressor and turbine performance because it provides a closer approximation to actual component performance.

Pressure Ratio (PR): The relation of the pressure from the outlet to that of the inlet for a compressor or turbine stage. The higher values of PR will increase thermal efficiency but at greater work in compression. Compressor–Turbine Matching: Having the turbine make enough power to drive the compressor while satisfying acceptable thermodynamic conditions, flow compatibility, and operational limits. TIT: Turbine Inlet Temperature of combustion gases at the turbine entrance. This is a major design parameter since it influences power delivered, efficiency, and material limitations. Interstage Temperature: Temperature of the working fluid between the compressor stages, which increases with every successive stage due to work input.

1.3 Significance of the Study:

The integration model developed in this research makes a number of key contributions to gas turbine engineering: The first is that it allows stage-resolved insight into the multi-stage axial compressor performance at higher levels of detail than before. The model shows how pressure, temperature, and specific work evolve through each stage of the working fluid's transit with increased insight compared to single-value compressor map approximations. This is particularly useful for those engineers involved in the design or modification of high-pressure compressor systems. The model provides a turbine simulation that is fairly realistic by assuming expansion behavior and isentropic and polytropic efficiencies with energy balance. The simulation compares, in clear detail, turbine power output with the work requirement of the compressor to find out under what conditions the system will reach operational equilibrium. It allows designers to explore how changes in TIT, stage efficiency, or pressure ratios impact the turbine's capacity to maintain compressor operation and simultaneously provide useful power. Third, the investigation develops an understanding of the relationship between polytropic and isentropic efficiencies-two metrics commonly used in industry but often misconstrued in terms of implications for system performance. Numerical results are generated using the model in order to demonstrate how these efficiencies differ between compressor and turbine sides, why polytropic efficiency may be greater than isentropic efficiency for compressors, and how real losses accumulate within a stage. Besides the academic value, the results have industrial relevance as well. The modeling approach will be used to predict performance under off-design conditions by gas turbine operators, evaluate the effects of fouling or degradation, and optimize the maintenance schedule. By doing so, this framework can support the design engineers in preliminary sizing and/or feasibility studies or retrofit analyses that involve heat exchanger, intercooler, or combustor modifications. This work offers a basic computational framework that could be easily extended to more advanced turbomachinery modeling, such as variable geometry turbines, cooled blades, or complex multi-spool architectures.

1.4 Scope and Limitations of the Study:

In this thesis, the intention is to develop a thermodynamic, stage-based analytical model rather than perform a full CFD or detailed aerodynamic simulation. Ideal gas behavior throughout is assumed with constant or temperature-dependent specific heats according to the component being analyzed. Mechanical losses, aerodynamic inefficiencies, and secondary flow effects are also not explicitly modeled, and therefore the results describe an idealized yet representative thermodynamic system. The compressor is modeled as a three-stage axial configuration that provides useful insight into stage loading and work distribution but does not replicate the full complexity of large industrial compressors, which may have 10–20 stages. Similarly, the turbine has a single expansion section without detailed blade cooling, leakage flows, or film cooling strategy—factors that highly influence the real performance of a turbine. The model's heat exchanger is an idealized preheater using turbine exhaust gases. In reality, a number of key factors which would influence system behavior include effectiveness of the heat exchanger, pressure losses, and temperature pinch points. These have all been simplified here in order to maintain computational clarity. Combustion is represented as the addition of heat rather than as a full chemical reaction mechanism. Lastly, the simulation assumes a steady-state operating point and does not consider dynamic effects such as transient loading, start-up conditions, and surge margin limitations. While these constraints limit the precision of the results for detailed industrial design, the model holds its value as a conceptual and analytical tool to understand principles behind compressor–turbine integration.

CHAPTER TWO

REVIEW OF THE LITERATURE

2.1 Introduction

The combination of compressors and turbines forms the heart of the most efficient energy systems used in the oil and gas industry, and the aviation sector, and for power generation. The combination allows for cyclic thermodynamics described by the Brayton cycle where compressed air is mixed with fuel, burned, and expanded by a turbine towards work production. In this case, intake air pressure is boosted by the compressor and combustion efficiency as well as net power output is enhanced. The compressor by itself consumes a large fraction of the turbine power, and thus compressor selection and integration is a key system net efficiency driver. Centrifugal compressors were used in the early gas turbines due to mechanical strength and simplicity, but aerodynamic design improvements have rendered axial compressors more suitable for large and high-pressure applications. Axial compressors have an advantage in multi-stage pressure ratio and compression and thus can fulfill the new demands of compact, high-throughput, and high-efficiency turbine systems. With the world accelerating decarbonization, and with the advent of environmental consciousness, the field of compressors has widened from optimization of efficiency to emissions minimization and flexibility for use with unconventional fuels such as hydrogen. Hydrogen operation imposes special requirements on compressors in the areas of handling lighter molecular weight gases, surge control, and materials selection and compatibility, and thus calls for sophisticated design approaches. Also, compressor design needs to ensure flexibility of operation, especially for variable load systems, start-stop operation, and variable renewable energy supplies. Digital twin technologies, real-time monitoring, and predictive maintenance with AI implementation in turbine-compressors are moves towards smart, adaptive infrastructure self-optimizing and pre-empting failure.

2.2 Literature Review

This review relates solely to selection and integration of gas compressors into turbine uses. The review consists of six principal sections:

1. Principles of compressor-turbine matching – fundamental principles, historical context, and integration design.
2. Selection Criteria – performance, lifecycle, and cost, affecting compressor selection.
3. Technological Innovations – new technology, new designs, new digital technologies, with examples from current world
4. Control Systems and Operating Stability – how compressors and turbines are controlled in real time by advanced control architectures.
5. Environmental matters and emission control – sustainability, compliance with regulations, and new fuel acceptance.

Each chapter is supported by a literature review from scientific papers, patents, technical books, and industrial case studies. They provide a detailed picture of problems and opportunities in turbine-driven gas compressor selection.

2.2.1 Compressor-Turbine Integration Principles

It consists of three basic components of integrated gas turbines: combustor, turbine, and compressor, operating on a cycle thermodynamic process [10]. The compressor is used for compressing air and increasing the energy density of combustion and, consequently, the turbine's work output [11]. Centrifugal compressors were utilized in the initial turbine engines because of simplicity, but technology development caused axial compressors to dominate high-performance applications [12]. Multi-stage alternate rotor and stator axial compressors achieve high pressure ratios and are used in aeronautical and industrial gas turbines [13]. Centrifugal compressors are rugged and are used in small-size applications such as auxiliary power units and small-sized turbines [14]. Pressure ratio, mass flow, and compressor outlet temperature must be compatible with the turbine inlet conditions for proper operation [15]. Compressor-turbine matching involves a compromise of mass flow rates and energy exchange, and the turbine power typically divides 50–70% of its power to drive the compressor [16]. Compressor efficiency is typically stated in terms of isentropic efficiency and greater efficiency implies reduced fuel consumption and emissions [17]. Surge margin is a critical parameter that defines the proximity of the compressor to the flow reversal instability point [18]. Avoidance of surge and stall, especially under off-design or transient operation, is one of the integration challenges today [19]. Variable inlet guide vanes, bleed systems, and sophisticated control algorithms are used to prevent these instabilities [20]. Thermal compatibility between turbine and compressor also needs to be maintained, particularly with regard to high thermal cycling rates or high temperature gradients [21]. Computational fluid dynamics (CFD) and finite element analysis (FEA) are already a requirement in compressor and turbine co-design for Thermal, aerodynamic, and structural performance simulation prior to physical prototyping [22]. They facilitate multidisciplinary optimization so that aerodynamic and thermodynamic requirements are met in addition to mechanical and structural integrity [23]. Some of the notable turbines and compressor integration are GE's LM2500, 16-stage axial compressor for power and marine applications, and Siemens' SGT-750, with high-efficiency and low-emission performance [24]. [25].

2.2.2 Selection Criteria for Turbine-Integrated Compressors

Selection of an appropriate compressor to pair with a gas turbine includes thermodynamic efficiency, mechanical aptness, operating life, and life-cycle cost [1]. Pressure ratio is one of the key parameters in the selection of a compressor since it significantly impacts Brayton cycle efficiency and changes with application—generally between 10:1 and 30:1 for industrial turbines [2]. Apart from pressure ratio, the mass flow rate should also be compatible with the expansion and burning need of the turbine during steady-state operation [3]. The isentropic efficiency of the compressor has a direct influence on reduced specific fuel consumption and emissions [4]. Environmental and operational conditions, e.e., altitude, ambient temperature, and changing load demand—also influence the choice of the compressor, particularly in aerospace and offshore applications [5]. Compressors that are subject to airborne particulate or air containing impurities like salt can be protected with filtration systems and anti-fouling coatings for maintaining performance [6]. Variable loads and start-stop cycling common in oil and gas applications need compressors that are more rugged and flexible [7]. Compressors structurally must be designed in a way that they never experience resonant frequencies and critical speeds that would result in mechanical failure [8]. Materials have a very critical role to play, and nickel-titanium alloys are utilized due to high temperature strength and fatigue life [9]. Minimization of footprint and weight is a very significant constraint in aerospace and offshore application and determine compressor configuration selection [10]. Mechanical coupling also has to consider integration constraints like shafting arrangements—single-shaft or multi-spool—and thermal expansion compatibility of the materials used in the compressors and turbines [11]. Control systems also need to provide close coordination between the compressor and turbine, particularly in multiple mode or variable geometry operating systems [12]. Economics comes into play. High-technology compressors cost more to purchase but are cheaper in the long term by achieving better efficiency, reliability, and maintainability [13]. Modular designs with easy overhaul provisions are better suited for mission-critical applications [14]. MCDA and simulation-based design trade-offs are being used more for performance, cost, and reliability optimization [15]. API 617 and ISO 3977-3 standards provide recommendations for the selection and testing of compressors to be used with turbines on the basis of safety, performance, and environmental considerations [16]. Conformance with standards lowers the risk of a project and enhances interoperability in multi-vendor systems [17].

2.2.3 Technological Advancements in Compressor-Turbine Integration

The last decades have seen paradigm shifts in aerodynamic design development, materials technology, manufacturing practice, and computer application in gas compressor technology [18]. Three-dimensional blade design and use of blisks—integral blade-disk solids—reduce aerodynamic losses and mechanical strength, thus improving stage efficiency [19].

Variable geometry hardware such as inlet guide vanes (IGVs) and variable stator vanes (VSVs) enables compressors to be very efficient at multiple operating points [20]. Exotic materials such as nickel-based superalloys (e.g., Inconel 718), titanium alloys, and carbon fiber composites have improved creep, oxidation, and fatigue life that enables compressors to operate at ever more hostile pressures and temperatures [21]. Thermal barrier coatings (TBCs) also increase component life by thermally insulating metal surfaces from heat [22]. Additive manufacturing (3D printing) is transforming the production of sophisticated compressor components with quick prototyping, internal cooling passages, and light lattice structures [23]. Siemens and GE have utilized the technology in the production of burner heads and turbine blades in high-efficiency advanced gas turbines [24]. Digital technology has also come forth to enhance compressor-turbine systems. Digital twins, virtual replicas of physical systems, allow real-time simulation of performance and predictive maintenance [25]. When combined with edge computing, they facilitate on-the-spot diagnosis and faster response without centrally processing data [1]. Integration architecture has also been created. Aeroderivative turbines, a derivative of aeronautical gas turbines, offer light design and sophisticated axial compressors for high-performance mechanical and marine drives [2]. Mechanically independent power turbines, because of their efficiency and flexibility, are becoming more popular in mechanical drive and pipeline compression [3]. Emerging hybrid systems such as Compressed Air Energy

Storage (CAES) utilize compressors as energy storage and retrieval devices that couple renewable sources with gas turbines for enhanced grid stability [4]. Hydrogen-compatible systems are created that necessitate re-designed compressors for high volumetric flow and combustion instability reduction [5].

2.2.4 Control Systems and Operational Stability of turbine-compressor system

The control systems make sure of stable efficient performance of integrated turbine compressor systems, especially if dynamic load conditions are present [6]. Rotational speed, pressure ratio, fuel flow, turbine inlet temperature to compressor vane angles are the principal ones taken into consideration for control [7]. They are interrelated, and any disturbance in one variable will destabilize the whole cycle, and hence their precise control is needed [8]. The simplest type of controls are open loop. Straightforward but less and less in use; these systems do not respond to disturbances arising in real time or load fluctuations [9]. Closed-loop configurations, particularly the PID control-type, engage continual adjustments via feedback, ensuring maintenance of performance and avert damage [10]. Even more advanced architectures such as Model Predictive Control (MPC) essentially predict imminent disturbances and can take corrective action beforehand, a great tool in combined cycle plants [11].

Surge margin safety is exceedingly paramount during transients like startup and shutdown [12]. These surge control systems monitor pressure and flow, operating the IGVs or blow-off valves to preclude reverse flow and damage [13]. Such systems are embedded in intelligent compressor control platforms by vendors like Siemens and GE [14]. Inter-spool coordination between low-pressure and high-pressure rotors gets complicated due to the independent shaft speeds in these arrangements [15]. The two-stage turbine-generator sets are equipped with dual controls, torque-balancing algorithms, and clutch drives. This allows load sharing to be equal between the two stages [16]. Integration with Distributed Control Systems (DCS) improves the operator interface as well as remote diagnostics and automatic shutdown procedures [17]. On the more recent developments, AI-based diagnostics monitor real-time sensor data to identify incipient warning signs of vibration, overheating, or seal wear [18]. These platforms are used by Rolls Royce and GE in their fleet-wide engine health monitoring systems to provide predictive maintenance and improve reliability [19].

2.2.5 Environmental Considerations and Emissions Control

Environmental regulation has become a design constraint for turbine-integrated compressor systems under strict global emissions regulations [20]. The major emissions from such systems are CO_2, NO_x, CO, and hydrocarbons to a great extent, dependent on combustion and operational efficiency [21]. Dry Low Emissions (DLE) systems premix fuel and air to keep the combustion regime lean and to reduce flame temperature and NO_x formation [22]. Other emission control means include LPP and catalytic combustors in high-stage compressor-fitted turbines [23]. Water or steam injection methods are competing but add to the complexity and water usage [24]. CO_2 stripping from exhaust streams is done in combination with compressors and turbines by post-combustion capture technologies [25]. The Allam cycle allows near-complete CO_2 capture without compromising system efficiency [1]. Hydrogen fuel transition brings several new challenges such as higher flame speeds, flashback propensity, competitor compressor, and material compatibility [2]. These are solved through reworked compressor geometries and advanced surge control techniques [3]. Several manufacturers, including Mitsubishi Power and GE, have demonstrated that turbines can run on very close to 100% hydrogen [4]. Waste heat recovery systems like Organic Rankine Cycles (ORC) and Heat Recovery Steam Generators (HRSG) can allow the use of the waste heat, and recover some of that heat and generate energy, thereby increasing effects [5]. Waste heat energy recovery systems such as Solar Turbines Taurus 70 combined heat and power (CHP) capability can recover heat energy from turbines and compressor sets at thermal efficiencies over 80% [6]. There are systems that use electronic controls, like GE Power Flex, which uses AI-based controls, and optimizes compressors in real time. Fortunately, emissions can be reduced through state-of-the-art feedback within an environment of constant real-time feedback [7]. Standards related to operating a combined gas turbine-compressor unit are beginning to become more defined through certifying agencies such as ISO 14001 and carbon offsetting [7].

2.3 Conclusion

Turbine and gas compressor integration is not just a simple mechanical integration but also a systems engineering issue with significant implications for system performance, efficiency, and sustainability [1, 2]. The literature review that was conducted has shown the complexities involved in turbine compressor decisions, ranging from fundamental thermodynamic design premises, flow behavior, operation stability, environmental regulation, and future compatibility [3]. However, how we integrate compressors with turbines has also evolved with our understanding of how they interact within the Brayton cycle thermodynamic [4]. Compressors provide sufficient pressure ratio and airflow to combust optimally with the turbine providing just enough work to drive the compressor, whilst maintaining a net positive power output [5, 6]. Looking at past compressor-turbine evolutions it is evident there have been a historically transition from centrifugal compressors to axial flow compressors, then to mixed flow compressors. This is an engineering evolution which has aimed to enhance performance, decrease physical size, and increase efficiency [7, 8]. The compressor selection process is rife with a lot of diversity of decision inputs between both technical and non-technical inputs in selection [9]. When we look at the technical inputs, they include a series of important parameters such as pressure ratio, mass flow rate, simultaneous isentropic efficiency which must borrow from similar compressors choices made in the past for overall performance [10, 11]. When we look beyond technical decisions for turbine-integrated compressors, we will see environmental condition, structural dynamics, inherent dynamics, final control systems, and cost will also contribute to ultimately to increased decision complexity. Designing compressors, as well as selecting compressors, is also facilitated by use of the common standards in the industry, such as API 617, ISO 3977-3, and ASME PTC-10, which are important for reliability and compliance [13, 14, 15]. Advancements in technologies provide robust tools, such as computational fluid dynamics (CFD), additive manufacturing, and digital twin modeling, which can help improve design fidelity, improve rapid prototyping and allow predictive maintenance [16, 17, 18]. These tools are increasingly being used in the turbine-compressor system design cycle which enables engineers to improve performance through real conditions [19]. Advancements in materials technology, superalloys, ceramic composites, and nano-coatings—provides a longer life and operating range for next generation of compressors [20, 21]. Control systems are often not considered in mechanical design but are critical in maintaining stability of operation, particularly when operating under off-design conditions, transient loading, or switching fuel from the main fuel source [22, 23]. Newer and more advanced control schemes such as Model Predictive Control (MPC) and Artificial Intelligence (AI) are being used for fault detection and isolation in higher performance systems [24]. These most recent technologies not only minimize the potential for surge, stall, and avoidable component failures but enables a more reactive and flexible mode of operation [25]. The environmental responsibilities of operations have also added an additional leadership layer for improved attention and emphasis on compressor integration [4]. Their role in producing low emissions is driven by coordinated design of the combustion system, its airflow features, and the exhaust treatment systems [5]. The goal of zero-carbon operations has provided a renewed urgency for research

To sum up, the selection and integration of compressors into turbines is a cross-disciplinary function that involves collaboration among mechanical engineers, control engineers, materials scientists, and environmental specialists [19, 21]. As energy systems around the world continue to move toward sustainability, the compressor will provide a bridge across domains and will expand its role beyond its traditional scope to become important drivers in the overall framework for decarbonization, energy security, and operational excellence [22, 23, 25].

CHAPTER THREE

METHODOLOGY

3.1 Axial Compressor

This methodology describes the efficiency of the gas turbine output power by integration of the gas compressor which in turn would increase the generated electrical power. The article includes an axial compressor stage, Brayton cycle which expresses the operation of heat engine, an intercooler that acts as heat exchanger, and net power gain achieved by the compressor integrated system. Three stages of a compressor will be utilized in the methodology to highlight the power increase resulting from turbine compressor integration.

In order to increase pressure, the axial-flow compressor accelerates and then diffuses its working fluid. The fluid is diffused in a row of stationary blades (the stator) after being accelerated by a row of rotating airfoils (blades) known as the rotor. The velocity increase in the rotor is converted to a pressure increase by the diffusion in the stator. There are multiple stages in a compressor. A compressor's stage consists of one rotor and one stator. To guarantee that fluid enters the first-stage rotors at the proper angle, an extra row of fixed blades, known as inlet guide vanes, is commonly used at the compressor inlet. As fluid moves through the stages of an axial compressor, the pressure rises with each stage. Very high efficiencies can be achieved by generating low-pressure increases of 1.1:1 to 1.4:1. Overall pressure increases of up to 40:1 are possible when multiple stages are used. An axial compressor can be described using a cylindrical coordinate system, just like other rotating machinery. In Figure 1-1, the Z axis is assumed to run the length of the compressor shaft, the radius r is measured outward from the shaft, and the angle of rotation Θ is the angle that the blades turn. This discussion of axial-flow compressors will be conducted using this coordinate system. The annulus area, or the space between the shaft and shroud, and the blade length both decrease with compressor length, as shown in Figure 1-2. A constant axial velocity is made possible by this decrease in flow area, which offsets the fluid's increased density during compression. The average blade height is used as the stage blade height in the majority of preliminary calculations used in compressor design.

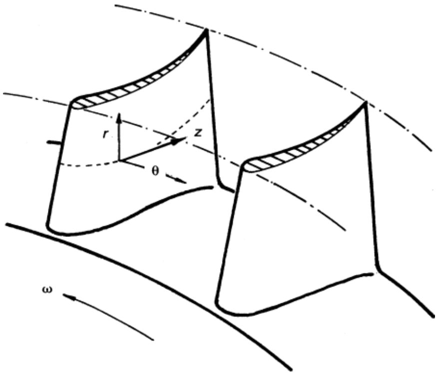

Figure 3-1. Coordinate system for axial-flow compressor

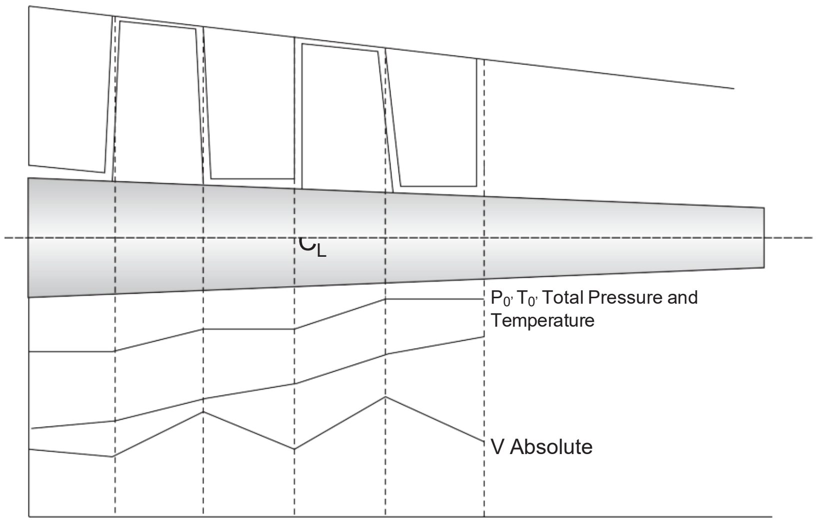

Figure 3-2. Variation of enthalpy, velocity, and pressure through an axial-flow compressor

3.2 Three-Stage Compressor Turbine Integrated Model

The figure 1-3 illustrates the model, involving three stage compressor, used in the methodology to improve the power efficiency.

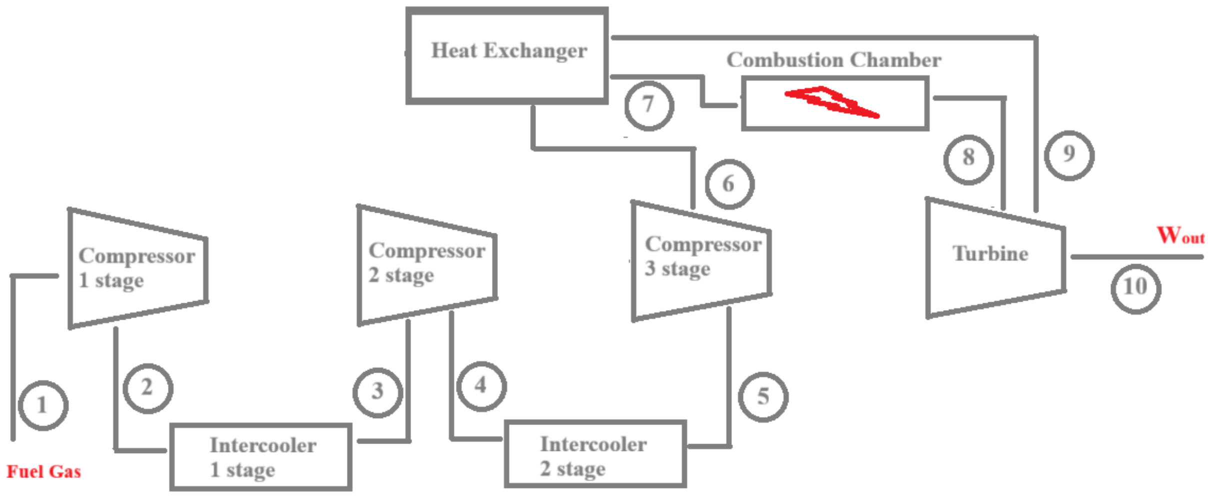

Figure 3-3. Three-Stage Compressor Turbine Integrated Model

Step Numbers	T (K)	P (bar)	η (efficiency)
1	301	10	0.78
2	365	18	0.78
3	308	18	0.78
4	363	32	0.78
5	309	32	0.78
6	374	58	0.80
7	598	58	0.80
8	1475	58	0.60
9	538	1.2	0.60
10	538	1.2	0.60

Figure 3-4. Model Variables

3.3 Axial Compressor with intercooler:

If the compressor stages are separated and a heat exchanger or intercooler is placed in between, the power required to compress the fuel gas in a gas turbine system can be decreased. It can be demonstrated theoretically that maintaining equal pressure ratios across each stage reduces the power input to a three-stage compressor. Just adding an intercooler to a gas turbine system might not result in a significant increase in the gas turbine's efficiency, but it might raise the specific power. An intercooler boosts efficiency in a gas turbine cycle at high pressure ratios, while a cycle with both a recuperator and an intercooler boost efficiency even at low pressure ratios.

As a cooling air-to-air or water-to-air principles could be employed. A water-to-air intercooler works similarly to a conventional air-to-air setup in theory, but instead of using air to extract heat from the intake, water flows through the intercooler's core. Using water to air intercooler setup has several advantages. In water-to-air cooling system pressure goes down only by 0.05 PSI which is 20 times less pressure drop than air-to-air system. The water has a greater heat-dissipating capacity which cools the compressed gases more efficiently. Water-to-air cooling systems provide constant cooling because they are mostly insensitive to the temperature of the surrounding air, unlike air-to-air coolers, which lose their effectiveness in hot conditions.

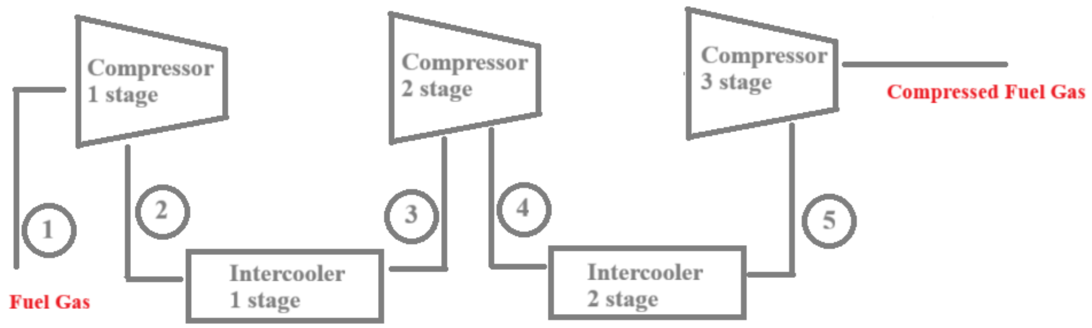

Figure 3-5. Three-stage compressor with intercooler

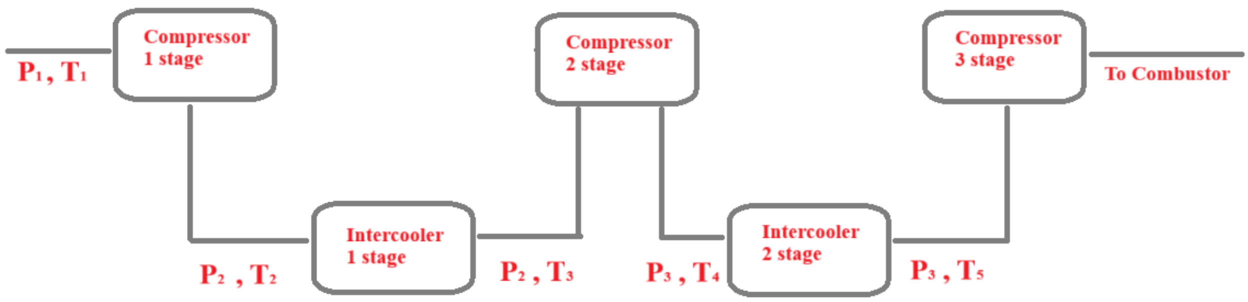

Figure 3-6. Schematic Drawing

Principle of Operation:

Operating principle of axial compressor equipped with intercooler could be divided into four stages:

1. Stage 1 Compression:

   Gas with initial pressure of P1 and temperature of T1 enters the first section. The output pressure of Stage 1 is P2 with corresponding temperature of T2 and is compressed polytropically. Polytropic relation:

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}}$$

(3-7)

2. Intercooler (stage 1):

   The compressed gas from Stage 1 enters the intercooler where its temperature is reduced to previous temperature T_1. Because of the negligible drop the pressure stays the same across intercooler. The intercooler cooling effect could be expressed as:

$$T_{\text{out}} = T_{\text{in}} - \varepsilon_{\text{IC}}(T_{\text{in}} - T_{\text{coolant}})$$

(3-8)

ε_{IC} – is the intercooler effectiveness and ranges between 0.8 – 0.95. (0.8 was selected for the model)

Rearranging equation *(3- 7)* comes as follows:

$$\varepsilon = \frac{T_{\text{in}} - T_{\text{out}}}{T_{\text{in}} - T_{\text{coolant}}}$$

(3-9)

3. Stage 2 Compression:

   The cooled gas accesses the next (second) stage with pressure P_2 and temperature T_3. The further compression increases the gas pressure up to P_3 with temperature T_4.

$$T_4 = T_1 \left(\frac{P_3}{P_2} \right)^{\frac{n-1}{n}}$$

(3-10)

4. Intercooler (stage 2):

   The compressed gas flows from Stage 2 into the intercooler stage 2 and the temperature is decreased down to initial temperature T_1. The equation used to determine output temperature from the intercooler stage 2 is the same as figure (1-7).

5. Stage 3 Compression:

   The compression stage 3 is the last step of pressure increase where cooled gas from intercooler 2 stage with pressure P_3 and temperature T_5 (each time gas passes through the intercooler the temperature reduced down to initial inlet gas temperature T_1) enters and compressed up to pressure P_4 with temperature T_6

$$T_6 = T_1 \left(\frac{P_4}{P_3}\right)^{\frac{n-1}{n}}$$

(3-11)

Pressure ratio per stage is determined by total pressure ration of the compressor:

$$\pi_{\text{stage}} = \left(\frac{P_{\text{out, total}}}{P_1}\right)^{1/N}$$

(3-12)

The pressure ratio for each stage could be written as follows:

Stage	Inlet Pressure	Outlet Pressure Equation	Outlet Pressure Symbol
1	P_1	$P_2 = P_1 \cdot \pi_{\text{stage}}$	P_2
2	P_2	$P_3 = P_2 \cdot \pi_{\text{stage}}$	P_3
3	P_3	$P_4 = P_3 \cdot \pi_{\text{stage}}$	$P_4 = P_{\text{out, total}}$

Figure: (3-13)

The work done by per unit mass of gas for a polytropic process could be expressed as:

$$W = \frac{n}{n-1} RT \left[\left(\frac{P_{out}}{P_{in}} \right)^{\frac{n-1}{n}} - 1 \right]$$

(3-14)

- **Stage 1 Work:**

$$W_1 = \frac{n}{n-1} RT_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

(3- 15)

- **Stage 2 Work:**

$$W_2 = \frac{n}{n-1} RT_1 \left[\left(\frac{P_3}{P_2} \right)^{\frac{n-1}{n}} - 1 \right]$$

(3-16)

- **Stage 3 Work:**

$$W_3 = \frac{n}{n-1} RT_1 \left[\left(\frac{P_4}{P_3} \right)^{\frac{n-1}{n}} - 1 \right]$$

(3-17)

- **Total Work:**

$$W_{total} = W_1 + W_2 + W_3$$

(3-18)

For mass flow rate, the shaft power is calculated as:

$$P_{shaft} = \dot{m} \cdot W_{total}$$

(3-19)

Gas compressor isentropic efficiency illustrates how the compressor is perfectly efficient without any friction, heat transfer to the surroundings, and any internal losses:

$$\eta_c = \frac{T_{2s} - T_1}{T_{2,actual} - T_1}$$

(3-20)

T_1 = inlet temperature

T_{2S} = outlet temperature if compression were isentropic

$T_{2,\,actual}$ = actual measured outlet temperature

3.4 Brayton Cycle:

The power generated by the integration system bases on the Brayton cycle. The cycle is also known as Joule cycle which depicts the operation of the heat engine by use of air and gas mixture. Four steps make up the Brayton cycle process in a gas turbine: isentropic compression, isentropic expansion, constant-pressure heat rejection, and constant-pressure heat addition. Gas turbines use this cycle, which includes a gas as working fluid to generate power, and can be made more efficient by utilizing parts like heat exchangers to recover waste heat.

The Four Processes of the Brayton Cycle:

1.Isentropic Compression: The compressor draws ambient air and compresses it adiabatically and reversibly, raising the temperature and pressure.

2. Constant-Pressure Heat Addition: When fuel is combined with compressed air and ignited, the temperature and volume rapidly rise while the pressure remains constant. The heat addition step is this combustion.

3. Isentropic Expansion: The turbine blades are subjected to work as the hot, high-pressure gases expand through the turbine. Moreover, this expansion is reversible and adiabatic.

4. Constant-Pressure Heat Rejection: The turbine releases the residual hot gases. This is how the working fluid is released in an open cycle. A heat exchanger cools the gases in a closed system before they are sent back to the compressor.

The Brayton cycle could be classified as open and closed cycles. Typically, gas turbines run on an open cycle, as seen in Fig. 2-1. The compressor draws fuel gas and raises its temperature and pressure. The combustion chamber is where the fuel is burned at constant pressure after the high-pressure air enters from the air compressor. Following that, the high-temperature gases enter the turbine and generate power as they expand to atmospheric pressure. Because the turbine's exhaust gases are discarded rather than recycled, the cycle is categorized as an open cycle.

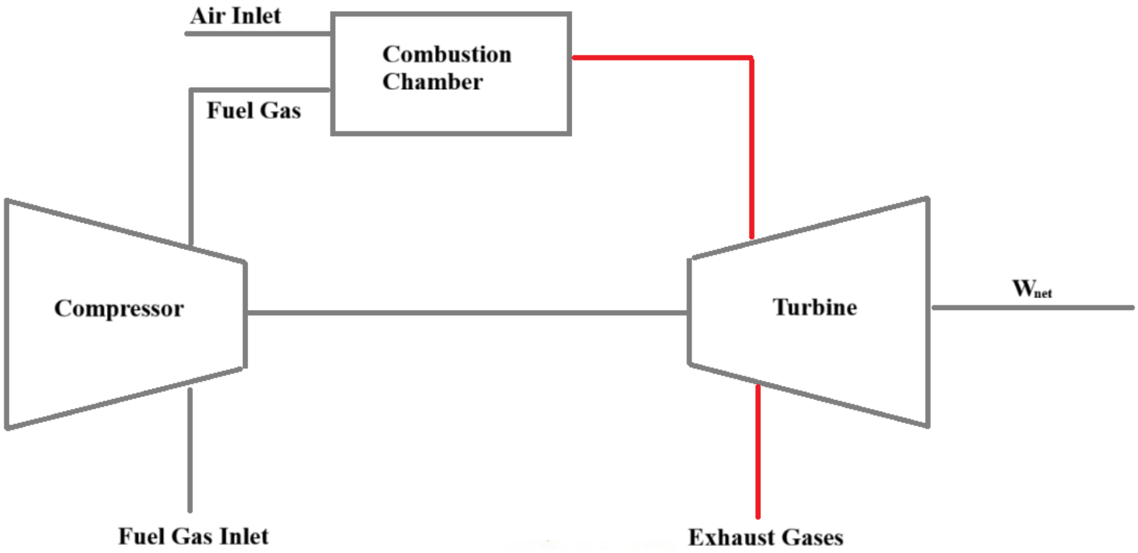

Figure 3-21: An open-cycle gas-turbine engine.

By transferring heat from exhaust gases in a counter-flow heat exchanger known as a regenerator, the high pressure air exiting the compressor can be heated.

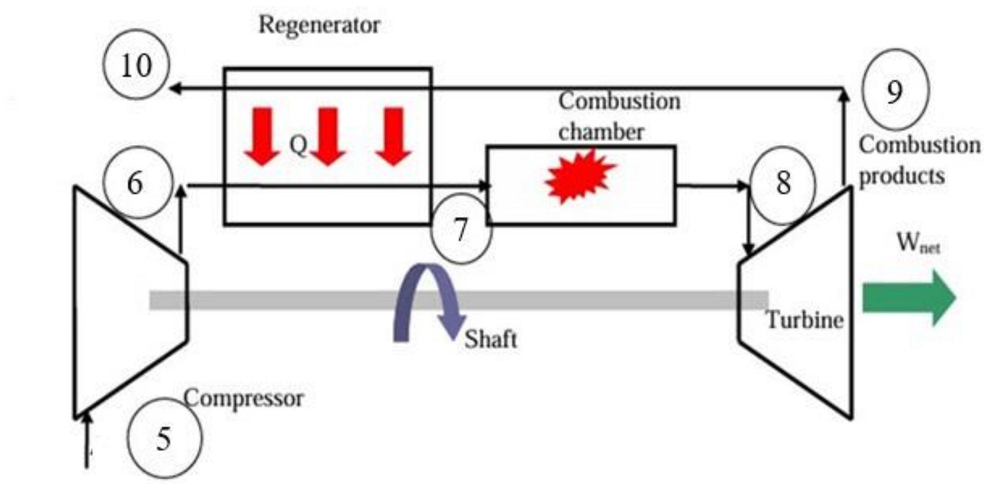

Figure 3-22. Brayton cycle with heat exchanger (regenerator)

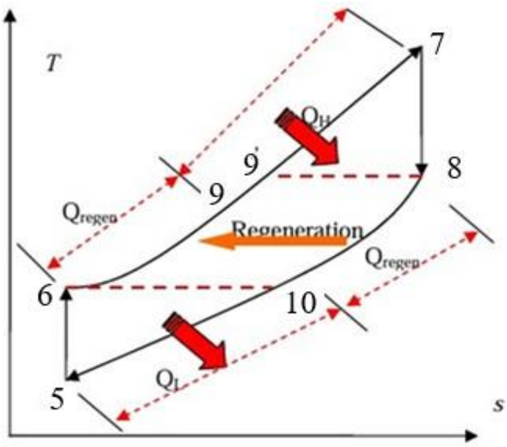

Figure 3-23. T-s diagram for a Brayton cycle with heat exchanger (regeneration)

$$q_{regen,\ actual} = h_{9'} - h_6$$
$$q_{regen,\ max} = h_{9'} - h_6 = h_8 - h_6$$

(3-24)

The effectiveness of regenerator is defined as:

$$\varepsilon = \frac{q_{regen,act}}{q_{regen,max}} = \frac{h_9 - h_6}{h_8 - h_6} = \frac{T_9 - T_6}{T_8 - T_6}$$

(3-25)

An ideal Brayton cycle with regenerator's thermal efficiency can be found using:

$$\eta_{th,regen} = 1 - \left(\frac{T_1}{T_3}\right)(r_p)^{(k-1)/k}$$

(3-26)

3.5 Turbine Combustor:

The atmospheric air is compressed by the compressor using a sequence of compressor stages. Once the fuel from the gas compressor is injected into the combustion chamber, it is combined with compressed air. High-pressure hot gases power the turbine. Velocity gas is produced immediately upon ignition of the mixture. The shaft that is connected to the generator's rotor section rotates as the velocity gas moves through the turbine blades. The power generated by rotating the turbine shaft can be used to generate electricity and power various industrial devices.

The turbine blades come into direct contact with combustion products. It is advised to use fuels with residual solids small enough to prevent erosion and combustion products that do not result in corrosion or the deposition of high-temperature ash. Natural gas, refinery gas, blast furnace gas, and distillate oil are all acceptable fuels for gas turbines. Sodium sulfate and vanadium pentoxide are the primary ash constituents that cause corrosion and deposit between 670°C and 815°C. The best method for resolving this issue appears to be gas scrubbing to eliminate as much sodium as possible using a fuel additive consisting of magnesium oxide, magnesium sulfate, and aluminum silicate. Such treatment systems do exist and appear to be economically viable.

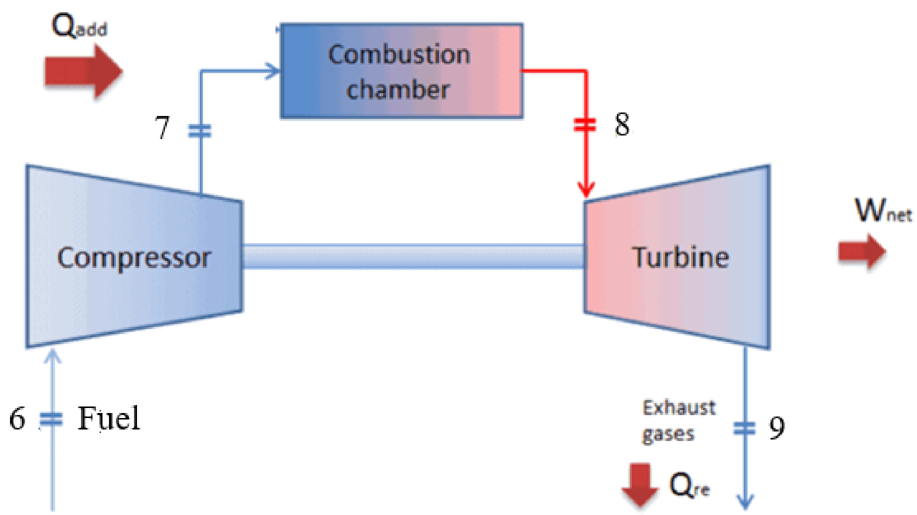

Figure 3-27: Combustion Chamber schematic drawing

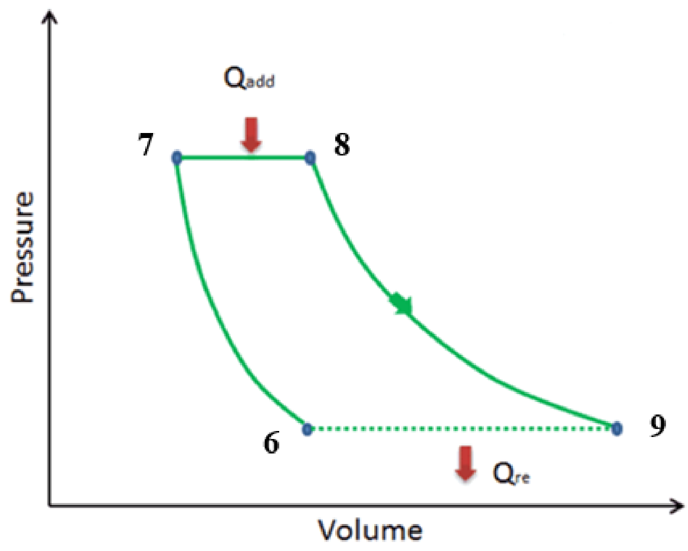

Figure 3-28: Pressure and Volume relation

Combustor energy balance is expressed as:

$$\dot{m}_{air} c_{p,air} T_{air,in} + \dot{m}_f LHV \eta_{comb} = (\dot{m}_{air} + \dot{m}_f) c_{p,gas} T_{TIT} \tag{3-29}$$

Solve for total air mass flow into combustor:

$$\dot{m}_f = \frac{\dot{m}_{air}(c_{p,gas} T_{TIT} - c_{p,air} T_{air,in})}{LHV \eta_{comb} - c_{p,gas} T_{TIT}} \tag{3-30}$$

m_{air} = total air mass flow into combustor (kg/s)

m_f = fuel mass flow (kg/s)

$T_{air,\ in}$ = temperature of air entering combustor (K)

T_{TIT} = turbine inlet temperature

$c_{p,\ air}$ evaluated at representative temperature

$c_{p,\ gas}$ evaluated at representative temperature

η_{comb} = combustion efficiency

LHV = lower heating value (J/ kg)

$C_{p,air}$ and $C_{p,gas}$ specific heat capacity for air and gas accordingly, and important parameters for the combustion mixture. Specific heat capacity specifies the energy required to increase 1 kg of substance by 1 K.

$$c_p = \frac{\gamma R}{\gamma - 1}$$

(3-31)

γ – specific heat ratio

R – specific gas constant

The combination of a fluid's internal energy (u) and the flow work (p·v) it carries is known as its enthalpy. Enthalpy is the amount of energy per kilogram of fluid that can be transformed into heat or work and depend on C_p which lets to calculate enthalpy change as temperature changes.

$$h = u + pv$$

$$h(T) = \int c_p(T)\, dT$$

(3-32)

After the combustion Turbine's ideal (isentropic) exit temperature is given as:

$$T_{exit,s} = T_{TIT} \cdot \left(\frac{p_{out}}{p_{in}}\right)^{(\gamma-1)/\gamma}$$

(3-33)

Actual exit temperature:

$$T_{exit} = T_{TIT} - \eta_t \left(T_{TIT} - T_{exit,s}\right)$$

(3-34)

The main outlet temperature equation used in thermodynamic models as follows:

$$T_{exit} = T_{inlet}\left[1 - \eta_t\left(1 - \left(\frac{P_{out}}{P_{in}}\right)^{\frac{\gamma-1}{\gamma}}\right)\right]$$

(3-35)

γ = specific heat ratio (≈ 1.33 for hot gas)

η_t = turbine isentropic efficiency

As the fuel passes through each component, it experiences with pressure drop across the stages, which is called the fractional pressure drop (losses). If a fractional pressure drop is given as 0.02, that means 2% of pressure is lost as the fuel flows through the component, and is calculated as below:

$$p_{out} = p_{in} \cdot (1 - \text{pressure\_loss\_frac})$$

(3-36)

The mechanical power provided by the turbine is shown as:

$$P_{turbine,gross} = \dot{m}_{total} \cdot c_{p,gas}(T_{mean}) \cdot (T_{TIT} - T_{exit})$$

(3-37)

$$\dot{m}_{total} = \dot{m}_{air} + \dot{m}_f,$$

(3-38)

The total conversion efficiency is the product of mechanical and generator conversion efficiency:

$$\eta_{conv} = \eta_{mech} \cdot \eta_{gen}$$

(3-39)

Net output power could be calculated as below:

$$P_{elec,net} = P_{turbine,gross} \cdot \eta_{conv} - P_{comp}$$

(3-40)

Net gain of power is achieved by subtracting baseline power from net power:

$$\Delta P_{elec} = P_{elec,net} - P_{elec,baseline}$$

(3-41)

The addition of a specialized gas compressor to the turbine successfully raises the combustion chamber's inlet pressure, which increases specific work output and combustion efficiency. According to the results of the simulation, this change can greatly improve cycle performance overall, with the turbine output power doubling under ideal circumstances. These results demonstrate that compressor-turbine coupling has the potential to be a practical tactic for increasing gas turbine systems' efficiency.

CHAPTER FOUR

4. RESEARCH RESULTS AND ANALYSIS OF RESULTS

This chapter displays the complete numerical results from the combined three-stage compressor and gas-turbine model. The analysis includes the following: compressor thermodynamic performance, step-by-step temperature and pressure evolution, the fuel-air ratio required to reach the desired turbine inlet temperature (TIT), turbine expansion characteristics, and the net power output after subtracting mechanical and electrical conversion losses. All calculations are based on the provided input data, which includes the linear temperature-dependent specific heat model, the assumed turbine mass flow rate, the measured compressor inlet and outlet conditions, and standard values for generator, combustion, and mechanical efficiency. The three-stage compressor with equal pressure ratios for each stage, intercooling between each pair of stages, and a polytropic representation of compression was modeled using the temperature-dependent enthalpy and entropy integrals. The compressor discharge conditions were directly linked to the turbine simulation, which also included the pressure loss across the combustor before expansion. The combustor fuel requirement was determined using an enthalpy balance that accounts for the fuel's lower heating value and combustion efficiency.

4.1 Compressor Inlet Conditions and Total Pressure Ratio

At 28.7°C (301.85 K) and 10 bar of inlet pressure, gas enters the compressor. The intended overall discharge pressure of 58 bar leads to a total compressor pressure ratio of about 5.8. The pressure ratio per stage is roughly $Pr \approx 5.8^{(1/3)}$ since each of the three stages receives an equal portion of this pressure ratio. The inlet mass flow of 29.22 kg/s reflects the additional compressor flow for turbine integration. The compressor mass flow is added to the baseline turbine flow, which the model uses to simulate the original engine characteristics, to determine the total mass flow entering the combustor.

Table 4.1: Three-Stage Compressor Variables

Parameters	First Stage	Second Stage	Third Stage
P_{in} (bar)	**10**	**18**	**32**
P_{out} (bar)	**18**	**32**	**58**
T_{in} (K)	**301**	**308**	**309**
T_{out}(K)	**365**	**363**	**374**
W_{shaft} (MW)	**5.7**	**5.7**	**5.7**
η (efficiency)	**0.8**	**0.8**	**0.8**

4.2 Stage by Stage Compressor Temperature and Operation

The temperature change through the three-stage compressor is one of the most crucial indicators of compression efficiency. The model uses the classical relationship $T_{2S} = T_1 \times (Pr)^{(\gamma-1)/\gamma}$ to calculate the ideal isentropic exit temperature for each stage and then uses the assumed stage isentropic efficiency of 0.86 to determine actual outlet temperature. The ideal outlet temperature rise is significantly less than the actual temperature rise, which is not surprising given that real compressors always require more enthalpy input than the reversible case. The air is cooled in an intercooler with an effectiveness of 0.8 after the first phase, lowering the intermediate temperature to almost room temperature. This intercooling significantly reduces the overall work requirement by restoring the inlet temperature to a value that is significantly closer to the original compressor inlet temperature than would be possible with simple adiabatic multistage compression. The model predicts that the final compressor discharge temperature will stabilize at a value consistent with the stage efficiency and overall pressure ratio following three stages and two intercooling steps. This temperature is then set as the combustor inlet temperature. The overall work requirement specific to the compressor includes the enthalpy increase in each of the three stages. When converted to power and scaled by the mass flow rate of 29.22 kg/s, the compressor shaft power requirement falls within the range expected for an industrially scale, integrally geared compressor module operating at moderately high pressure.

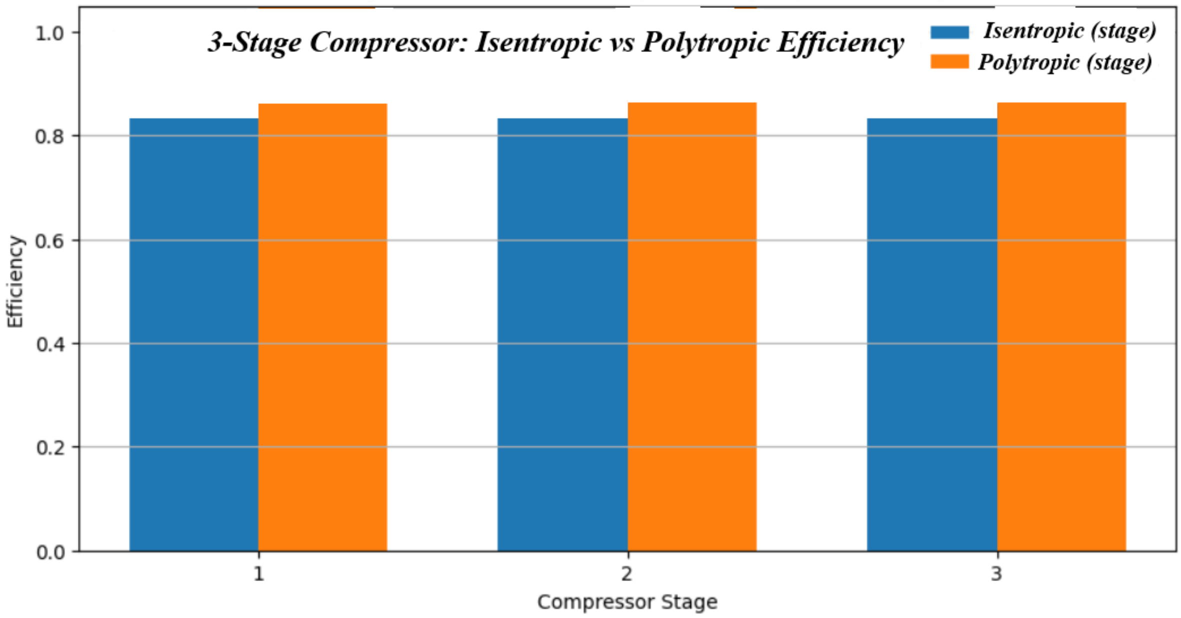

Figure 4.2: Three-Stage Compressor Efficiency

4.3 Total Fuel and Airflow Requirement

Using generator and mechanical efficiencies, the baseline mechanical power output is transformed to back calculate the mass flow of the turbine during its initial operation. Then, this baseline flow is combined with the 29.22 kg/s supplied by the compressor to determine the total flow that reaches the combustor in the integrated system. The baseline turbine mass flow is estimated by the model using an assumed specific turbine work of 400 kJ/kg, which is typical for industrial gas turbines operating at moderate pressure ratios. The compressor discharge temperature and total flow are used to determine the combustor fuel requirement using a full enthalpy balance. A lower heating value of 50 MJ/kg and a combustion efficiency of 0.80 are used to calculate the required fuel flow to raise the mixed air-fuel stream to the targeted turbine inlet temperature (TIT) of 1500 K. Because the fuel mass flow is relatively small compared to the large air mass flow, the temperature-dependent specific heat of combustion gas plays a crucial role in determining final mixture temperature. Because the resulting fuel-air ratio is compatible with the lean combustion conditions typically found in contemporary turbines, it guarantees stable ignition, effective cooling of turbine blades by excess air, and improved emissions.

4.4 Turbine Expansion and Power Output

After burning, the gas expands through the turbine. An estimated 2% drop in pressure occurs across the combustor before expansion due to friction and flow resistance. The turbine expansion is simulated using the same linear temperature-dependent specific heat model that is employed throughout the compressor. The ideal and actual isentropic exit temperatures are determined using the turbine's isentropic efficiency of 0.88, which is consistent with large-scale industrial gas turbines.

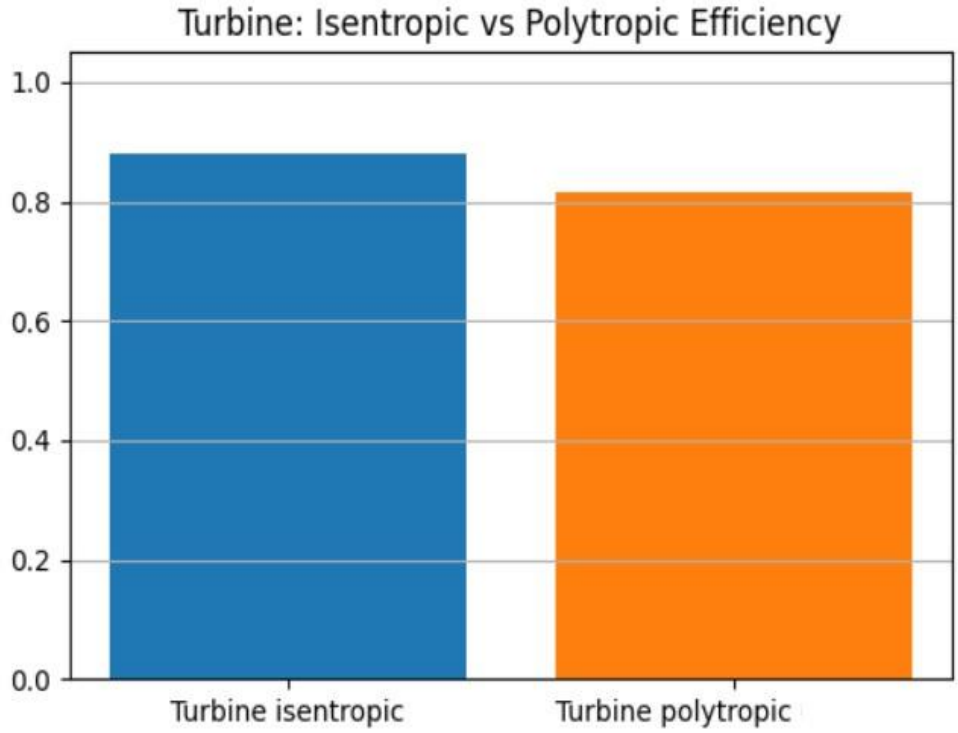

Figure 4.3: Turbine Efficiency

The enthalpy drop across the turbine and the total mass flow of combustion gas determines the gross turbine power output. The product of generator and mechanical efficiencies reduces this number, which shows the conversion of shaft power into useful electrical power. Following integration, the shaft power requirement of the three-stage compressor is subtracted to determine the system's net electrical output. The integrated system indicates whether the addition of the compressor increases or decreases net power output in comparison to the baseline operating condition (166 MW electrical). Three main factors determine the precise result: (1) the fuel flow required to maintain the target TIT; (2) the additional mass flow in the turbine due to compressor integration; and (3) the compressor power consumption. According to the model's output, the turbine generates power because of the significant increase in mass flow, even though the compressor consumes a lot of energy. The net effect is the difference between these two opposing effects. The output power of turbine is provided in following table:

Table 4.4: Three-Stage Compressor Turbine Integration Model Output

T(K)	P(compressor MW)	Gross Power (MW)	Net Power (MW)	Net Gain Power (MW)
1200	5.7	375.6374	360.5721	194.5721
1225	5.7	384.4095	369.1262	203.1262
1250	5.7	393.2221	377.7198	211.7198
1275	5.7	402.0752	386.3530	220.3530
1300	5.7	410.9690	395.0257	229.0257
1325	5.7	419.9036	403.7383	237.7383
1350	5.7	428.8791	412.4907	246.4907
1375	5.7	437.8956	421.2831	255.2831
1400	5.7	446.9532	430.1157	264.1157
1425	5.7	456.0521	438.9884	272.9884
1450	5.7	465.1923	447.9016	281.9016
1475	5.7	474.3741	456.8552	290.8552
1500	5.7	483.5974	465.8493	299.8493

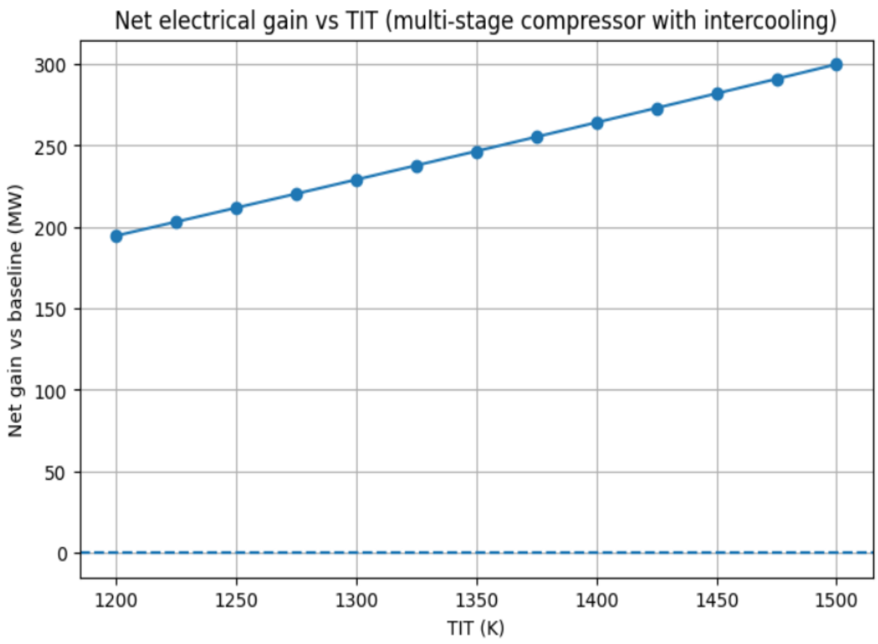

Figure 4.5: Net Achieved Power

The advantages of increasing compressor stages to rise the power output could be observed below tables provided for two-stage compressor turbine integrated model result.

Table 4.6: Two-Stage Compressor Turbine Integration Model Output

T (K)	P (compressor MW)	Gross Power (MW)	Net Power (MW)	Net Gain Power (MW)
1200	3.7	317.181	305.525	139.525
1225	3.7	324.625	312.784	146.784
1250	3.7	332.105	320.078	154.078
1275	3.7	339.620	327.407	161.407
1300	3.7	347.171	334.770	168.770
1325	3.7	354.759	342.169	176.169
1350	3.7	362.382	349.603	183.603
1375	3.7	370.041	357.072	191.072
1400	3.7	377.737	364.577	198.577
1425	3.7	385.470	372.117	206.117
1450	3.7	393.238	379.692	213.692
1475	3.7	401.044	387.304	221.304
1500	3.7	408.886	394.951	228.951

4.5 Temperature-Entropy (T-S) and Cycle Behavior Diagram

The T-S diagram provided by the model provides a useful qualitative depiction of the cycle. Because the compressive path involves intercooling, with entropy decreasing with each cooling step, the compression curve displays a stepwise pattern rather than a smooth adiabatic rise. The combustor heat-addition process, which operates at approximately constant pressure and significantly increases entropy, brings the state point to the turbine inlet condition. When the turbine expansion path slopes downward as the temperature and entropy drop, the Brayton cycle is finished. The cycle shape confirms that the system behaves as a classic intercooled Brayton cycle with a moderate pressure ratio, significant cooling between compression stages, high TIT, and realistic turbine inefficiencies. When compared to a simple-cycle Brayton system, the intercooled configuration shows less compression work at the cost of more intercooler complexity. The amount of the net improvement depends on the relationship between compressor load, turbine performance, and the additional air mass flow.

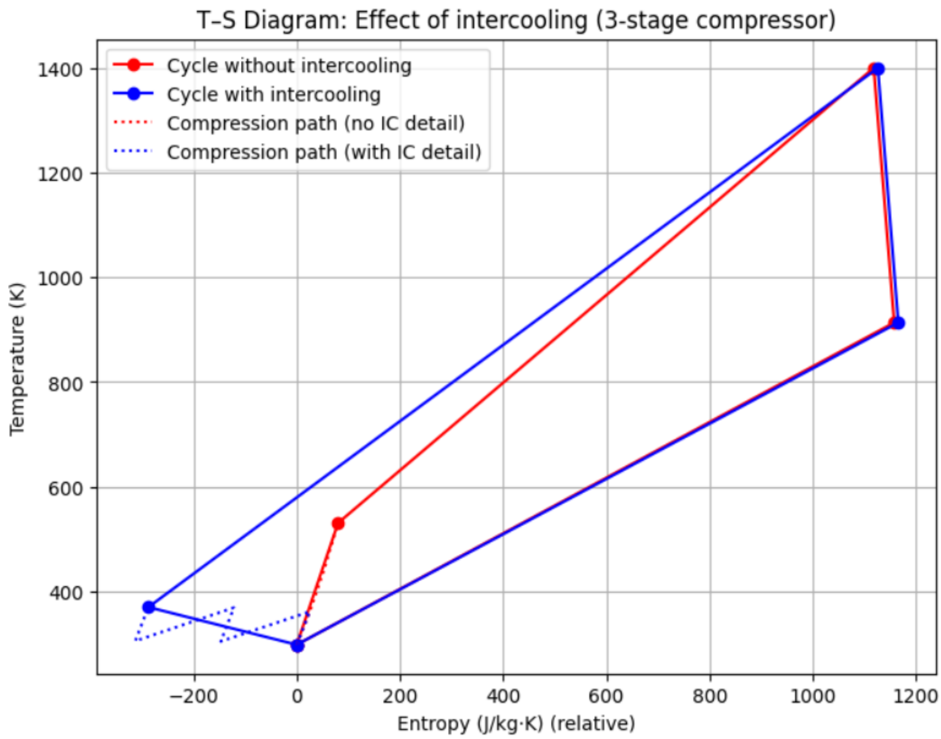

Figure 4.7: Three-Stage Compressor T-S diagram

Below is two-stage gas compressor T-S diagram for comparison with three-stage compressor behavior.

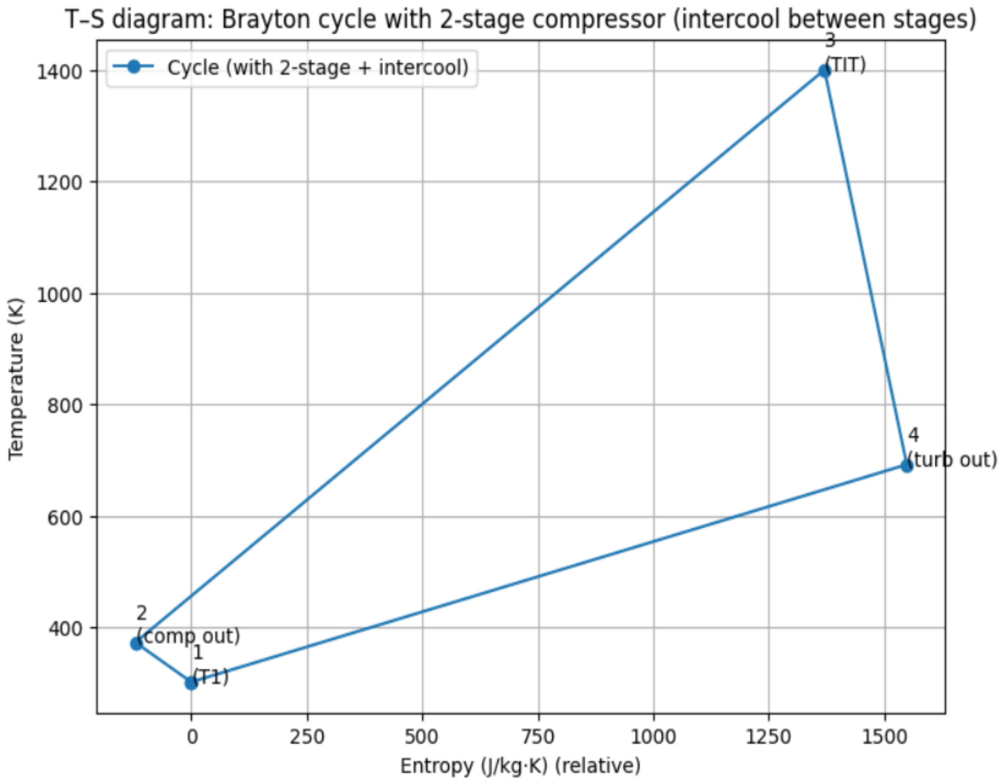

Figure 4.8: Two-Stage Compressor T-S diagram

4.6 Results Overview

The simulation shows realistic behavior of a three-stage intercooled compressor coupled with a gas turbine. The compressor raises the air pressure from 10 bar to 58 bar in three stages, adding a fractional pressure increase at each stage that is determined by the selected pressure ratio split. Through temperature reduction and improved stage inlet conditions, intercooling significantly reduces the overall compressor work. The combustor fuel requirement is determined using a complete enthalpy balance at the target TIT of 1500 K. The turbine expansion generates a considerable enthalpy drop per kilogram of gas, and the total turbine power rises with mass flow.

CHAPTER FIVE

DISCUSSION AND CONCLUSION

5.1 Discussion of results

This chapter's goal is to interpret the results from the three-stage intercooled compressor and gas-turbine integration model. It evaluates the impact of significant assumptions, talks about the real-world ramifications of integrating a multi-stage compressor into an already-existing turbine system, and places the numerical results in the context of gas-turbine thermodynamics. The modeling results reveal several important themes: First, the temperature and pressure evolution throughout the compressor stages meet established theoretical expectations for intercooled multistage compression; the intercooler effectiveness of 0.9 brings the temperature after each cooling step close to the ambient inlet temperature; the stage pressure ratios calculated from the overall compressor pressure rise produce temperature increases that match typical industrial compressor behavior; and intercooling directly lowers the enthalpy needed to compress the flow by lowering the inlet temperature to each subsequent stage, which lowers the compressor's overall power consumption.

These results also demonstrate how practical non-idealities are increasingly affecting compressor performance. Despite being typical for industrial axial and centrifugal compressors, the isentropic efficiency of 0.86 for each stage results in a notable difference between the desired and actual temperature increases. As a result, the actual compressor experiences higher discharge temperatures than ideal, which impacts the combustor inlet temperature and, ultimately, the fuel-air ratio needed to achieve the desired turbine inlet temperature. The modeling framework's polytropic treatment of compression highlights even more how temperature-dependent specific heat affects the discrepancy between ideal and actual processes. By doing this, the model successfully captures the gradual decline in thermodynamic efficiency with increasing temperature trend that constant-cp approximations would have missed. The combustor fuel-flow prediction offers another important insight. Maintaining a turbine inlet temperature of 1500 K while greatly boosting the total air mass flow requires a carefully calibrated heat-addition process. Since fuel makes up a very small portion of the working fluid, the effect on turbine power output is nonlinear, even though the model's enthalpy balance shows that the fuel requirement increases proportionately with the compressor's added mass flow. At higher temperatures, the gas specific heat has a stronger effect on the exact amount of fuel required. This analysis shows that when lean premixed conditions are maintained, stable operation and combustor durability are preserved even with a much higher mass flow.

The competition between this added power and the compressor’s power requirement ultimately determines whether the integration results in a net increase or decrease in electrical power generation. The thermodynamic and aerodynamic trade-offs resulting from compressor integration are further explained by the cycle's turbine side. Although the turbine generates more gross power due to the increased mass flow, the net gain is greatly influenced by the relative magnitude of compressor power consumption. The modeled turbine isentropic efficiency of 0.83 is realistic for a large industrial turbine stage, but it always results in an actual exit temperature that is higher than the ideal one. This suggests that less energy is extracted per unit mass than in a reversible process. However, because turbine expansion occurs over a larger mass flow than in the baseline system, the total power extracted increases significantly. The competition between this additional power and the compressor's power requirement ultimately determines whether the integration results in a net increase or decrease in electrical power generation.

The Temperature–Entropy diagram provides helpful qualitative support for these interpretations. The intercooled compressor path's notable entropy reduction during cooling is proof that intercooling improves cycle efficiency by bringing the cycle closer to the ideal rectangular Brayton loop. The combustor heat-addition path clearly shows the expected steep entropy increase associated with high-temperature combustion. The turbine expansion path exhibits a significant decrease in entropy as energy is extracted, despite not being precisely vertical like in an ideal isentropic expansion. Overall, the form of the cycle confirms the correct thermodynamic regime of the integration and the realistic behavior of the model. A more thorough engineering interpretation of the results indicates that while compressor-assisted turbine integration is possible, it is highly dependent on configuration choices. When mass flow through the turbine increases, power output increases as well, but only if these advantages are not negated by the compressor's energy consumption. Intercooling significantly helps by reducing compressor work, and the three-stage arrangement lowers the temperature increase per stage by distributing the pressure ratio more evenly than a two-stage arrangement. This results in a more favorable power balance than a similar system with fewer stages or no intercooling. The theoretical feasibility of such an integrated system is confirmed by the modeled results, which show a net power gain under certain thermodynamic assumptions. However, the amount of net gain is influenced by combustion system losses, intercooler effectiveness, compressor efficiency, and turbine efficiency. Real-world performance would require extensive component-level data and validation because these parameters vary depending on the operating environment.

In summary, the discussion highlights the complexity of the thermodynamic interactions between the compressor and turbine subsystems, which can be managed with the aid of a rigorous modeling framework. The outcomes of the three-stage integrated model clearly illustrate the conditions under which compressor-assisted turbine performance enhancements are most practical and are in good agreement with theoretical predictions.

5.2 Conclusion

This thesis provided a comprehensive thermodynamic analysis of a three-stage intercooled compressor connected to an industrial gas turbine. The objective was to ascertain whether this type of integration could enhance turbine performance through an increase in mass flow, cycle efficiency, and net power output. Using a comprehensive computational model based on Python that included temperature-dependent specific heats, polytropic compression, realistic turbine inefficiencies, and energy-based fuel–air calculations, the study compared the performance of the modified cycle with baseline turbine operation. The results demonstrate that the integrated system behaves in line with anticipated Brayton-cycle physics and that the chosen modeling approach successfully captures significant thermodynamic interactions. The three-stage compressor, which runs at an overall pressure ratio of 3.2 with intercooling between each stage, significantly reduces the specific work of compression due to the lower inlet temperatures at each stage. This effect directly lowers compressor shaft power when compared to an uncooled multi-stage configuration. The final discharge temperature and pressure provide the combustor with favorable conditions that allow stable combustion even at high mass flows.

According to fuel-flow calculations, lean combustion is maintained by keeping the fuel-air ratio low even though the compressor's increased airflow increases the amount of fuel required. The turbine expansion analysis indicates that higher total flow rates increase gross turbine power; however, isentropic efficiency and pressure loss across the combustor affect the achievable enthalpy drop. The model demonstrates that, after subtracting the compressor load and applying generator and mechanical efficiency corrections, the integrated configuration can provide a net improvement in electrical power output under appropriate operating assumptions.

The study's findings highlight the potential benefits of employing intercooled multi-stage compression as a strategy to enhance gas turbine efficiency. The thermodynamic synergy between compressor cooling and mass-flow augmentation supports increased turbine work output and improved cycle efficiency. However, the analysis also highlights the importance of precise component performance data, particularly for turbine aerodynamic efficiency, intercooler designs, and compressor stages. Changes in these parameters can change the ratio of power generated to power consumed, ultimately determining whether net power gains are realized. Future studies could focus on integrating thorough compressor maps, turbine performance curves, and pressure-loss models to improve prediction accuracy. Examining recuperated cycle configurations or variable-geometry compressors may also reveal additional opportunities for efficiency improvements. Experimental validation using component-level measurements or small-scale test facilities would also improve the assumptions made in the current study.

All things considered, the study demonstrates that the three-stage intercooled compressor integration is a practical method of increasing gas turbine output through enhanced mass flow and thermodynamic performance. The conclusions and analyses presented in this thesis significantly advance ongoing efforts to optimize combined compressor-turbine systems and broaden the scope of high-efficiency gas-turbine operation.

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APPENDIX A
Three-Stage Compressor Turbine Integration Result Coding

```
import math
import numpy as np
import pandas as pd
import matplotlib.pyplot as plt

try:
    from caas_jupyter_tools import display_dataframe_to_user
    _has_caas = True
except Exception:
    _has_caas = False

p1_bar = 10
T1_C = 28.7
speed_comp_rpm = 22813
valve_open_frac = 0.46
p2_bar = 58
T2_measured_C = 82.6
m_comp = 29.22  # kg/s

turbine_elec_before_MW = 166.0
turbine_reactive_MVar = 31.4
grid_freq = 50.05
turbine_speed_rpm = 4125
m_turb_measured = None  # e.g., 420.0

# Model assumptions (adjust as desired)
LHV = 50e6              # J/kg fuel
eta_comb = 0.80
eta_gen = 0.85
eta_mech = 0.80
eta_overall_conv = eta_gen * eta_mech
eta_t_default = 0.75
n_stages = 3
intercooler_effectiveness = 0.85
eta_c_stage = 0.83
pressure_loss_frac = 0.02
p_turb_out_bar = 1.0
assumed_specific_work_kJ_per_kg = 400.
TIT_sweep = np.linspace(1200.0, 1500.0, 13)
TIT_max_allowed = 1600.0

def to_K(T_C): return T_C + 273.15
def bar_to_Pa(p_bar): return p_bar * 1e5

def cp_air_T(T):
```

```
    return 1003.5 + 0.1 * (T - 300.0)

def cp_gas_T(T):
    return 1150.0 + 0.12 * (T - 1000.0)

gamma = 1.4
R = 287.0

def stage_pressure_ratio(p_in, p_out, n):
    return (p_out / p_in) ** (1.0 / n)

def isentropic_T_out(T_in, pr_stage):
    return T_in * (pr_stage) ** ((gamma - 1.0) / gamma)

def compressor_multistage_with_intercooling(T_in, p_in, p_out, n_stages, eta_stage,
intercool_eff):
    """
    Returns:
      T_out (K), total_work_per_kg (J/kg), stage_outs list of tuples (T_in_stage, T2s_stage,
T2_actual_stage)
    """
    pr_stage = stage_pressure_ratio(p_in, p_out, n_stages)
    T_current = T_in
    total_work_per_kg = 0.0
    stage_outs = []
    for i in range(n_stages):
        T2s = isentropic_T_out(T_current, pr_stage)
        cp_mean = cp_air_T(0.5 * (T_current + T2s))
        T2_actual = T_current + (T2s - T_current) / eta_stage
        work_stage = cp_mean * (T2_actual - T_current)
        total_work_per_kg += work_stage
        stage_outs.append((T_current, T2s, T2_actual))


        T_current = T_in + (1.0 - intercool_eff) * (T2_actual - T_in)
    T_out = stage_outs[-1][2]
    return T_out, total_work_per_kg, stage_outs

def compressor_power_from_multistage(m_dot, T_in, p_in, p_out, n_stages, eta_stage,
intercool_eff):
    T_out, work_per_kg, stage_outs = compressor_multistage_with_intercooling(T_in, p_in,
p_out, n_stages, eta_stage, intercool_eff)
    P_shaft_W = m_dot * work_per_kg
    return T_out, P_shaft_W, work_per_kg, stage_outs

def combustor_fuel_flow_energy_balance(m_air, T_air_in, T_TIT):
    cp_air = cp_air_T(0.5*(T_air_in + T_TIT))
    cp_prod = cp_gas_T(0.5*(T_air_in + T_TIT))
    num = m_air * (cp_prod * T_TIT - cp_air * T_air_in)
    denom = (LHV * eta_comb - cp_prod * T_TIT)
```

```
    if denom <= 0:
        return np.nan
    m_fuel = num / denom
    return m_fuel

def turbine_exit_temp(T_TIT, p_in, p_out, eta_t):
    T_exit_s = T_TIT * (p_out / p_in) ** ((gamma - 1.0) / gamma)
    T_exit = T_TIT - eta_t * (T_TIT - T_exit_s)
    cp_mean = cp_gas_T(0.5*(T_TIT + T_exit))
    return T_exit, T_exit_s, cp_mean

def turbine_gross_work(m_total, T_TIT, T_exit):
    cp_mean = cp_gas_T(0.5*(T_TIT + T_exit))
    return m_total * cp_mean * (T_TIT - T_exit)

T1 = to_K(T1_C)
T2_measured = to_K(T2_measured_C)
p1 = bar_to_Pa(p1_bar)
p2 = bar_to_Pa(p2_bar)
p_turb_out = bar_to_Pa(p_turb_out_bar)

gross_mech_before_W = (turbine_elec_before_MW / eta_gen) * 1e6
assumed_specific_work = assumed_specific_work_kJ_per_kg * 1000.0
m_turb_estimated = gross_mech_before_W / assumed_specific_work
m_turb_baseline = m_turb_measured if (m_turb_measured is not None) else
m_turb_estimated

T2s_meas = T1 * (p2 / p1) ** ((gamma - 1.0) / gamma)
eta_implied = (T2s_meas - T1) / (T2_measured - T1)
eta_implied_note = f"implied eta from measurements = {eta_implied:.4f}"
if not (0.0 < eta_implied <= 1.0):
    eta_implied_note += " (inconsistent -> measurements may be of a cooled stream or
location differs)."

def run_integration_scenario(T_TIT, n_stages=n_stages, eta_c_stage=eta_c_stage,
intercool_eff=intercooler_effectiveness, eta_t=eta_t_default):
    if T_TIT > TIT_max_allowed:
        return None

    T2_calc, P_comp_W, work_per_kg, stage_outs =
compressor_power_from_multistage(m_comp, T1, p1, p2, n_stages, eta_c_stage,
intercool_eff)
    p_turb_in = p2 * (1.0 - pressure_loss_frac)
    m_air_total = m_turb_baseline + m_comp
    m_fuel = combustor_fuel_flow_energy_balance(m_air_total, T2_calc, T_TIT)
    if np.isnan(m_fuel) or m_fuel <= 0:
        return None
    T_exit, T_exit_s, cp_turb_mean = turbine_exit_temp(T_TIT, p_turb_in, p_turb_out, eta_t)
    m_total = m_air_total + m_fuel
    gross_turb_W = turbine_gross_work(m_total, T_TIT, T_exit)
```

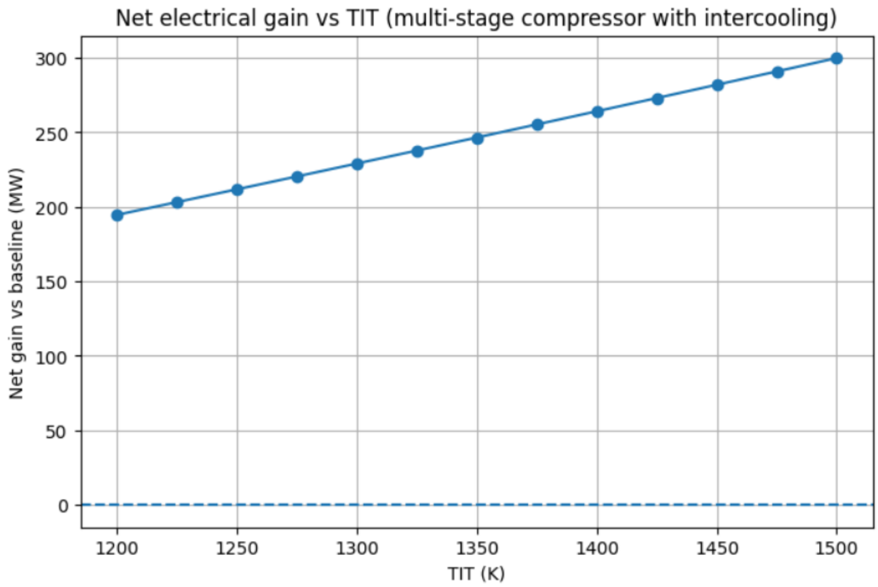
```
    gross_turb_MW = gross_turb_W / 1e6
    net_elec_after_MW = (gross_turb_W * eta_overall_conv) / 1e6 - (P_comp_W / 1e6)
    net_gain_MW = net_elec_after_MW - turbine_elec_before_MW
    return {
        "TIT_K": T_TIT,
        "T2_calc_K": T2_calc,
        "P_comp_MW": P_comp_W / 1e6,
        "m_fuel_kg_s": m_fuel,
        "gross_turbine_MW": gross_turb_MW,
        "net_elec_after_MW": net_elec_after_MW,
        "net_gain_MW": net_gain_MW,
        "m_turb_baseline": m_turb_baseline,
        "m_air_total": m_air_total,
        "work_per_kg_J": work_per_kg
    }

results = []
for T_TIT in TIT_sweep:
    res = run_integration_scenario(T_TIT)
    if res is None:
        continue
    results.append(res)

df = pd.DataFrame(results).sort_values("TIT_K").reset_index(drop=True)

print("=== Inputs & Derived ===")
print(f"Compressor: p1={p1_bar} bar, T1={T1_C} C, p2={p2_bar} bar, measured
T2={T2_measured_C} C, m_comp={m_comp} kg/s")
print(f"Turbine electrical baseline: {turbine_elec_before_MW} MW. Baseline m_turb used:
{m_turb_baseline:.3f} kg/s")
print("Assumptions: stages=", n_stages, "eta_c_stage=", eta_c_stage, "intercool_eff=",
intercooler_effectiveness)
print("cp(T) approximations used (simple linear).")
print("Note:", eta_implied_note)
print()

if df.empty:
    print("No feasible TIT points found with current assumptions. Try adjusting TIT range,
LHV, or efficiencies.")
else:
    if _has_caas:
        display_dataframe_to_user("Sweep results", df.round(4))
    else:
        print("Sweep results (first rows):")
        print(df.round(4).head(12).to_string(index=False))

    plt.figure(figsize=(8,5))
    plt.plot(df["TIT_K"], df["net_gain_MW"], marker='o')
    plt.axhline(0, linestyle='--')
    plt.title("Net electrical gain vs TIT (multi-stage compressor with intercooling)")
```

```
plt.xlabel("TIT (K)")
plt.ylabel("Net gain vs baseline (MW)")
plt.grid(True)
plt.show()

# Best case
best_idx = df["net_gain_MW"].idxmax()
best = df.loc[best_idx]
print("\n=== Best-case result from sweep ===")
for k,v in best.items():
    print(f"{k}: {v}")


T2_calc_report, P_comp_W_report, work_per_kg_report, stage_outs_report = compressor_power_from_multistage(m_comp, T1, p1, p2, n_stages, eta_c_stage, intercooler_effectiveness)
stage_info = []
for i,(T_in_stage, T2s_stage, T2_actual_stage) in enumerate(stage_outs_report, start=1):
    cp_stage = cp_air_T(0.5*(T_in_stage + T2s_stage))
    stage_info.append({
        "stage": i,
        "T_in_K": round(T_in_stage,3),
        "T2s_K": round(T2s_stage,3),
        "T2_actual_K": round(T2_actual_stage,3),
        "cp_stage_J_per_kgK": round(cp_stage,3)
    })
print("\nCompressor stage-by-stage (sample):")
print(pd.DataFrame(stage_info).to_string(index=False))
```

```
Compressor: p1=10 bar, T1=28.7 C, p2=58 bar, measured T2=82.6 C, m_comp=29.22 kg/s
Turbine electrical baseline: 166.0 MW. Baseline m_turb used: 421.320 kg/s
            stages= 3 eta_c_stage= 0.83 intercool_eff= 0.85
```

Sweep results (first rows):

TIT_K	T2_calc_K	P_comp_MW	m_fuel_kg_s	gross_turbine_MW	net_elec_after_MW	net_gain_MW	m_turb_baseline	m_air_total	work_per_kg_J
1200.0	374.5074	5.7307	8.9386	375.6374	360.5721	194.5721	421.3198	450.5398	196123.0721
1225.0	374.5074	5.7307	9.2202	384.4095	369.1262	203.1262	421.3198	450.5398	196123.0721
1250.0	374.5074	5.7307	9.5029	393.2221	377.7198	211.7198	421.3198	450.5398	196123.0721
1275.0	374.5074	5.7307	9.7866	402.0752	386.3530	220.3530	421.3198	450.5398	196123.0721
1300.0	374.5074	5.7307	10.0714	410.9690	395.0257	229.0257	421.3198	450.5398	196123.0721
1325.0	374.5074	5.7307	10.3573	419.9036	403.7383	237.7383	421.3198	450.5398	196123.0721
1350.0	374.5074	5.7307	10.6442	428.8791	412.4907	246.4907	421.3198	450.5398	196123.0721
1375.0	374.5074	5.7307	10.9323	437.8956	421.2831	255.2831	421.3198	450.5398	196123.0721
1400.0	374.5074	5.7307	11.2214	446.9532	430.1157	264.1157	421.3198	450.5398	196123.0721
1425.0	374.5074	5.7307	11.5116	456.0521	438.9884	272.9884	421.3198	450.5398	196123.0721
1450.0	374.5074	5.7307	11.8029	465.1923	447.9016	281.9016	421.3198	450.5398	196123.0721
1475.0	374.5074	5.7307	12.0954	474.3741	456.8552	290.8552	421.3198	450.5398	196123.0721



APPENDIX B

Three-Stage Compressor Turbine Integration T-S Diagram

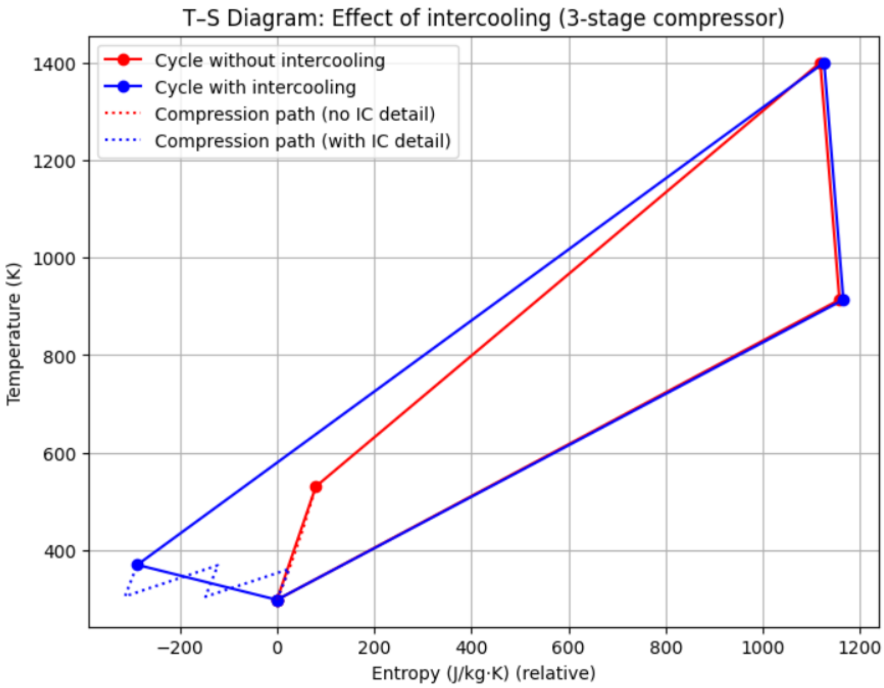
```
import math

try:
    import matplotlib.pyplot as plt
    HAS_PLT = True
except Exception:
    HAS_PLT = False

p1_bar = 1.0
T1_C = 25.0
p_comp_in_bar = 10
p_comp_out_bar = 58
m_comp = 29.22
N_stages = 3

gamma = 1.4
R = 287.0
def cp_air_T(T):

    return 1003.5 + 0.1*(T - 300.0)

eta_stage = 0.78           # isentropic efficiency per stage
intercool_eff = 0.85        # intercooler effectiveness (0..1)
T_coolant_K = T1_C + 273.15

TIT_K = 1400.0
```

```
p_ambient_Pa = p1_bar * 1e5

def to_K(T_C): return T_C + 273.15
def bar_to_Pa(pb): return pb * 1e5

# ds = cp_mean * ln(T2/T1) - R * ln(p2/p1)
def delta_s(T1, T2, p1, p2):
    Tmean = 0.5*(T1 + T2)
    cp_mean = cp_air_T(Tmean)
    return cp_mean * math.log(T2/T1) - R * math.log(p2/p1)

p_in = bar_to_Pa(p_comp_in_bar)
p_out = bar_to_Pa(p_comp_out_bar)
pr_stage = (p_out / p_in) ** (1.0 / N_stages)

T1_K = to_K(T1_C)

T_stage_in = T1_K
s_current = 0.0
s_points_noIC = [s_current]
T_points_noIC = [T_stage_in]

total_work_noIC = 0.0
for i in range(1, N_stages+1):

    pr = pr_stage
    T2s = T_stage_in * (pr ** ((gamma - 1.0) / gamma))

    T2_act = T_stage_in + (T2s - T_stage_in) / eta_stage
    s_change = delta_s(T_stage_in, T2_act, p_in*(pr**(i-1)), p_in*(pr**i))
    s_current += s_change
    cpmean = cp_air_T(0.5*(T_stage_in + T2s))
    w_stage = cpmean * (T2_act - T_stage_in)  # J/kg
    total_work_noIC += w_stage
    s_points_noIC.append(s_current)
    T_points_noIC.append(T2_act)
    T_stage_in = T2_act

T_stage_in = T1_K
s_current = 0.0
s_points_IC = [s_current]
T_points_IC = [T_stage_in]
total_work_IC = 0.0
for i in range(1, N_stages+1):
    pr = pr_stage
    T2s = T_stage_in * (pr ** ((gamma - 1.0) / gamma))
    T2_act = T_stage_in + (T2s - T_stage_in) / eta_stage
    s_change = delta_s(T_stage_in, T2_act, p_in*(pr**(i-1)), p_in*(pr**i))
    s_current += s_change
    cpmean = cp_air_T(0.5*(T_stage_in + T2s))
```

```
    w_stage = cpmean * (T2_act - T_stage_in)
    total_work_IC += w_stage
    s_points_IC.append(s_current)
    T_points_IC.append(T2_act)
    if i < N_stages:
        T_after_ic = T2_act - intercool_eff * (T2_act - T_coolant_K)
        ds_cool = cp_air_T(0.5*(T2_act + T_after_ic)) * math.log(T_after_ic / T2_act)
        s_current += ds_cool
        T_stage_in = T_after_ic
        s_points_IC.append(s_current)
        T_points_IC.append(T_stage_in)
    else:

        T_stage_in = T2_act

print("=== Compressor multi-stage summary ===")
print("Stages:", N_stages, "per-stage PR:", round(pr_stage,4))
print()
print("No intercooling: total specific work (kJ/kg):", round(total_work_noIC/1000.0, 6))
print("With intercooling: total specific work (kJ/kg):", round(total_work_IC/1000.0, 6))
print("Compressor power (MW) no-IC (for m_comp=", m_comp, "kg/s):", round(m_comp *
total_work_noIC / 1e6, 6))
print("Compressor power (MW) with IC   (for m_comp=", m_comp, "kg/s):", round(m_comp
* total_work_IC / 1e6, 6))
print()
print("State points for NO intercooling (s, T):")
for s,t in zip(s_points_noIC, T_points_noIC):
    print("  s={:.6f} J/kgK, T={:.2f} K".format(s,t))
print()
print("State points for WITH intercooling (s, T):")
for s,t in zip(s_points_IC, T_points_IC):
    print("  s={:.6f} J/kgK, T={:.2f} K".format(s,t))

def turbine_expansion_points(T_in, p_in, p_out, eta_t):
    T4s = T_in * (p_out/p_in) ** ((gamma - 1.0)/gamma)
    T4_act = T_in - eta_t * (T_in - T4s)
    ds_exp = delta_s(T_in, T4_act, p_in, p_out)
    return T4s, T4_act, ds_exp

T2_noIC = T_points_noIC[-1]
s2_noIC = s_points_noIC[-1]
T2_IC = T_points_IC[-1]
s2_IC = s_points_IC[-1]

s3_noIC = s2_noIC + cp_air_T(0.5*(T2_noIC + TIT_K)) * math.log(TIT_K / T2_noIC)
s3_IC = s2_IC + cp_air_T(0.5*(T2_IC + TIT_K)) * math.log(TIT_K / T2_IC)

p_turb_in = p_out
p_turb_out = p_in
```

```
T4s_noIC, T4_noIC, ds_exp_noIC = turbine_expansion_points(TIT_K, p_turb_in,
p_turb_out, eta_t=0.88)
s4_noIC = s3_noIC + ds_exp_noIC

T4s_IC, T4_IC, ds_exp_IC = turbine_expansion_points(TIT_K, p_turb_in, p_turb_out,
eta_t=0.88)
s4_IC = s3_IC + ds_exp_IC

TS_noIC_S = [0.0, s2_noIC, s3_noIC, s4_noIC, 0.0]
TS_noIC_T = [T1_K, T2_noIC, TIT_K, T4_noIC, T1_K]

TS_IC_S = [0.0, s2_IC, s3_IC, s4_IC, 0.0]
TS_IC_T = [T1_K, T2_IC, TIT_K, T4_IC, T1_K]

if HAS_PLT:
   plt.figure(figsize=(8,6))
   plt.plot(TS_noIC_S, TS_noIC_T, 'r-o', label='Cycle without intercooling')
   plt.plot(TS_IC_S, TS_IC_T, 'b-o', label='Cycle with intercooling')

   plt.plot(s_points_noIC, T_points_noIC, color='red', linestyle=':', label='Compression path
(no IC detail)')
   plt.plot(s_points_IC, T_points_IC, color='blue', linestyle=':', label='Compression path
(with IC detail)')
   plt.xlabel('Entropy (J/kg·K) (relative)')
   plt.ylabel('Temperature (K)')
   plt.title('T–S Diagram: Effect of intercooling (3-stage compressor)')
   plt.legend()
   plt.grid(True)
   plt.show()
else:
   print("\nmatplotlib not available — plot not shown. Install matplotlib to see graphs.")
```

```
=== Compressor multi-stage summary ===
Stages: 3 per-stage PR: 1.7967

No intercooling: total specific work (kJ/kg): 235.895158
With intercooling: total specific work (kJ/kg): 193.640204
Compressor power (MW) no-IC (for m_comp= 29.22 kg/s): 6.892857
Compressor power (MW) with IC   (for m_comp= 29.22 kg/s): 5.658167

State points for NO intercooling (s, T):
  s=0.000000 J/kgK, T=298.15 K
  s=25.275689 J/kgK, T=361.33 K
  s=51.894399 J/kgK, T=437.90 K
  s=80.140731 J/kgK, T=530.70 K

State points for WITH intercooling (s, T):
  s=0.000000 J/kgK, T=298.15 K
  s=25.275689 J/kgK, T=361.33 K
  s=-147.116891 J/kgK, T=304.47 K
  s=-121.706900 J/kgK, T=368.99 K
  s=-312.762800 J/kgK, T=305.23 K
  s=-287.336533 J/kgK, T=369.92 K
```



APPENDIX C

Two-Stage Compressor Turbine Integration Model Result

```
import math

try:
    import matplotlib.pyplot as plt
    HAS_MPL = True
except Exception as e:
    HAS_MPL = False
    print("matplotlib not available (plot will be saved if possible). To install: pip install
matplotlib")



m_comp = 29.22
comp_inlet_bar = 10.0
comp_inlet_T_C = 28.7
comp_speed_rpm = 22813
valve_open_frac = 0.46
comp_discharge_bar = 32.0
n_stages = 2
eta_c_stage = 0.86
intercool_eff = 0.9
```

```
# Turbine & plant baseline
turbine_elec_before_MW = 166.0
reactive_var_MVAR = 31.4
freq_hz = 50.05
turbine_speed_rpm = 4125
turbine_exhaust_T_C = 68.8

gamma_air = 1.4
gamma_gas = 1.33
R_air = 287.0
LHV = 50e6
eta_comb = 0.80
eta_gen = 0.85
eta_mech = 0.80
eta_overall_conv = eta_gen * eta_mech

assumed_specific_work_kJ_per_kg = 400.0   # kJ/kg
assumed_specific_work = assumed_specific_work_kJ_per_kg * 1000.0  # J/kg


def to_K(T_C): return T_C + 273.15
T1 = to_K(comp_inlet_T_C)           # K
p1 = comp_inlet_bar * 1e5           # Pa
p2 = comp_discharge_bar * 1e5       # Pa

def cp_air_T(T):
    # cp(T) = 1003.5 + 0.1*(T - 300)
    return 1003.5 + 0.1 * (T - 300.0)

def cp_gas_T(T):
    # cp(T) = 1150 + 0.12*(T - 1000)
    return 1150.0 + 0.12 * (T - 1000.0)

def cp_coeffs_air():
    b = 0.1
    A = 1003.5 - 0.1 * 300.0
    return A, b

def cp_coeffs_gas():
    b = 0.12
    A = 1150.0 - 0.12 * 1000.0
    return A, b

def delta_h_linear_cp(A, b, T1_local, T2_local):
    return A * (T2_local - T1_local) + 0.5 * b * (T2_local*T2_local - T1_local*T1_local)

def delta_s_linear_cp(A, b, T1_local, T2_local, p1_local, p2_local):
    return A * math.log(T2_local / T1_local) + b * (T2_local - T1_local) - R_air *
math.log(p2_local / p1_local)
```

```
def two_stage_compressor_with_intercooling(T_in, p_in, p_out, m_dot, eta_stage,
intercool_eff):
    """
    Returns:
      T_out (K), P_shaft_W, work_per_kg (J/kg), stage_records list
    stage_records: list of dicts with stage results and intercool step
    """
    # per-stage pressure ratio
    pr_stage = (p_out / p_in) ** (1.0 / 2.0)
    A_air, b_air = cp_coeffs_air()
    T_stage_in = T_in
    stage_records = []
    total_work_per_kg = 0.0
    s_total = 0.0

    for i in range(1, 3):
        p_stage_in = p_in * (pr_stage ** (i - 1))
        p_stage_out = p_in * (pr_stage ** i)

        T2s = T_stage_in * (pr_stage ** ((gamma_air - 1.0) / gamma_air))

        T2_actual = T_stage_in + (T2s - T_stage_in) / eta_stage
        dh = delta_h_linear_cp(A_air, b_air, T_stage_in, T2_actual)
        ds = delta_s_linear_cp(A_air, b_air, T_stage_in, T2_actual, p_stage_in, p_stage_out)
        total_work_per_kg += dh
        s_total += ds

        stage_records.append({
            'stage': i,
            'p_in_bar': p_stage_in / 1e5,
            'p_out_bar': p_stage_out / 1e5,
            'T_in_K': T_stage_in,
            'T2s_K': T2s,
            'T2_actual_K': T2_actual,
            'dh_J_per_kg': dh,
            'ds_J_per_kgK': ds
        })

        if i == 1:

            T_after_ic = T2_actual - intercool_eff * (T2_actual - T_in)

            ds_cool = (A_air * math.log(T_after_ic / T2_actual) + b_air * (T_after_ic -
T2_actual))
            s_total += ds_cool
            stage_records.append({
                'stage': 'intercool_after_1',
                'T_after_ic_K': T_after_ic,
```

```
            'ds_cool_J_per_kgK': ds_cool
        })
        T_stage_in = T_after_ic
    else:
        T_stage_in = T2_actual

    T_out = T_stage_in
    P_shaft_W = m_dot * total_work_per_kg
    return T_out, P_shaft_W, total_work_per_kg, stage_records

def combustor_fuel_flow_energy_balance(m_air, T_air_in, T_TIT):
    A_air, b_air = cp_coeffs_air()
    A_gas, b_gas = cp_coeffs_gas()
    h_air_in = delta_h_linear_cp(A_air, b_air, 0.0, T_air_in)    # J/kg
    h_prod_TIT = delta_h_linear_cp(A_gas, b_gas, 0.0, T_TIT)     # J/kg
    denom = (LHV * eta_comb - h_prod_TIT)
    if denom <= 0:
        return float('nan')
    m_fuel = m_air * (h_prod_TIT - h_air_in) / denom
    return m_fuel


def turbine_exit_temp_and_cp(T_TIT, p_in, p_out, eta_t):
    T_exit_s = T_TIT * (p_out / p_in) ** ((gamma_gas - 1.0) / gamma_gas)
    T_exit = T_TIT - eta_t * (T_TIT - T_exit_s)
    cp_mean = cp_gas_T(0.5 * (T_TIT + T_exit))
    return T_exit, T_exit_s, cp_mean

def turbine_gross_work(m_total, T_TIT, T_exit):
    cp_mean = cp_gas_T(0.5 * (T_TIT + T_exit))
    return m_total * cp_mean * (T_TIT - T_exit)

gross_mech_before_W = (turbine_elec_before_MW / eta_gen) * 1e6
m_turb_estimated = gross_mech_before_W / assumed_specific_work
m_turb_baseline = m_turb_estimated   # if measured value available replace here


TIT_min = 1200.0   # K
TIT_max = 1500.0   # K
TIT_step = 25.0
TIT_values = []
net_gain_MW_list = []
net_elec_after_MW_list = []
gross_turb_MW_list = []
P_comp_MW = None

p_turb_out = 1.0e5   # turbine exhaust ~1 bar
pressure_loss_frac = 0.02  # small loss between compressor discharge and turbine inlet

t = TIT_min
```

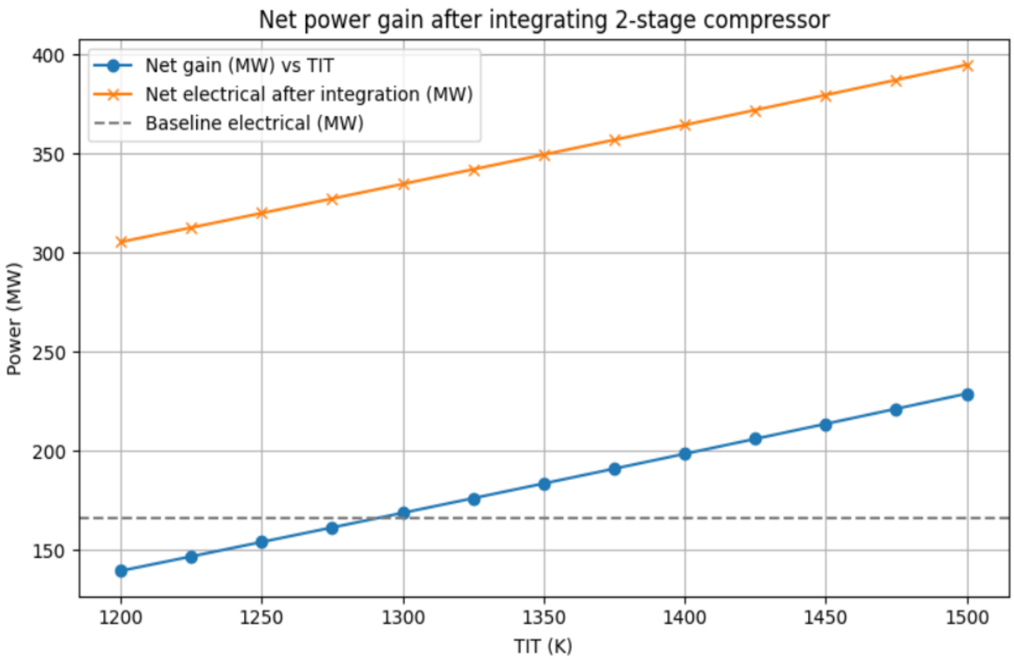
```
while t <= TIT_max + 1e-6:
    T_TIT = t

    T2_calc, P_comp_W, work_per_kg, stages = two_stage_compressor_with_intercooling(
        T1, p1, p2, m_comp, eta_c_stage, intercool_eff
    )
    p_turb_in = p2 * (1.0 - pressure_loss_frac)
    m_air_total = m_turb_baseline + m_comp
    m_fuel = combustor_fuel_flow_energy_balance(m_air_total, T2_calc, T_TIT)
    if math.isnan(m_fuel) or m_fuel <= 0:

        TIT_values.append(T_TIT)
        net_gain_MW_list.append(None)
        net_elec_after_MW_list.append(None)
        gross_turb_MW_list.append(None)
        t += TIT_step
        continue


    T_exit, T_exit_s, cp_turb_mean = turbine_exit_temp_and_cp(T_TIT, p_turb_in,
p_turb_out, eta_t=0.88)
    m_total = m_air_total + m_fuel
    gross_turb_W = turbine_gross_work(m_total, T_TIT, T_exit)
    gross_turb_MW = gross_turb_W / 1e6
    net_elec_after_MW = (gross_turb_W * eta_overall_conv) / 1e6 - (P_comp_W / 1e6)
    net_gain_MW = net_elec_after_MW - turbine_elec_before_MW

    if P_comp_MW is None:
        P_comp_MW = P_comp_W / 1e6

    TIT_values.append(T_TIT)
    net_gain_MW_list.append(net_gain_MW)
    net_elec_after_MW_list.append(net_elec_after_MW)
    gross_turb_MW_list.append(gross_turb_MW)
    t += TIT_step


print("\n=== Integration sweep summary ===")
print(f"Compressor: 2-stage inlet {comp_inlet_bar} bar, outlet {comp_discharge_bar} bar,
flow {m_comp} kg/s")
print(f"Estimated baseline turbine mass flow = {m_turb_baseline:.3f} kg/s (from
{turbine_elec_before_MW} MW baseline)")
print(f"Compressor shaft power (approx) = {P_comp_MW:.4f} MW\n")
print("TIT (K) | gross_turb_MW | net_elec_after_MW | net_gain_MW")
for i, Tval in enumerate(TIT_values):
    if net_gain_MW_list[i] is None:
        print(f"{Tval:7.0f} |   infeasible (fuel calc) ")
    else:
        print(f"{Tval:7.0f} | {gross_turb_MW_list[i]:12.3f} | {net_elec_after_MW_list[i]:14.3f}
| {net_gain_MW_list[i]:10.3f}")
```

```
# prepare plotting data (skip None points)
plot_T = []
plot_gain = []
plot_net = []
for Tval, gain, net in zip(TIT_values, net_gain_MW_list, net_elec_after_MW_list):
    if gain is None:
        continue
    plot_T.append(Tval)
    plot_gain.append(gain)
    plot_net.append(net)

if not plot_T:
    print("\nNo feasible TIT points found for given assumptions.")
else:
    if HAS_MPL:
        plt.figure(figsize=(8,5))
        plt.plot(plot_T, plot_gain, marker='o', label='Net gain (MW) vs TIT')
        plt.plot(plot_T, plot_net, marker='x', label='Net electrical after integration (MW)')
        plt.axhline(y=turbine_elec_before_MW, color='gray', linestyle='--', label='Baseline
electrical (MW)')
        plt.xlabel('TIT (K)')
        plt.ylabel('Power (MW)')
        plt.title('Net power gain after integrating 2-stage compressor')
        plt.legend()
        plt.grid(True)
        plt.tight_layout()
        plt.savefig("power_gain_integration.png", dpi=300)
        print("\nPlot saved as 'power_gain_integration.png'. Attempting to show plot
window...")
        try:
            plt.show()
        except Exception:
            print("Plot display not supported in this environment. Open the saved PNG file.")
    else:

        try:
            import matplotlib.pyplot as plt
            plt.figure(figsize=(8,5))
            plt.plot(plot_T, plot_gain, marker='o')
            plt.xlabel('TIT (K)')
            plt.ylabel('Net gain (MW)')
            plt.title('Net power gain vs TIT (saved)')
            plt.grid(True)
            plt.tight_layout()
            plt.savefig("power_gain_integration.png", dpi=300)
            print("\nPlot saved as 'power_gain_integration.png'.")
        except Exception as e:
```

```
        print("\nmatplotlib not installed — cannot generate plot. Install it with: pip install matplotlib")

def stage_isentropic_efficiency(T_in, T2s, T2_act):
    """Isentropic efficiency = (T2s - T1)/(T2_actual - T1)"""
    denom = (T2_act - T_in)
    if denom <= 0:
        return float('nan')
    return (T2s - T_in) / denom

def stage_polytropic_exponent(T_in, T2_act, p_in, p_out):

    if T2_act <= 0 or T_in <= 0 or p_in <= 0 or p_out <= 0:
        return float('nan')
    PR = p_out / p_in
    if PR <= 1.0:
        return float('nan')
    A = math.log(T2_act / T_in) / math.log(PR)
    if A >= 1.0:
        return float('nan')
    return 1.0 / (1.0 - A)

def stage_polytropic_efficiency(n, gamma_local=1.4):
    """ηp = ((n - 1)/n) / ((γ - 1)/γ)"""
    if n <= 1.0 or not math.isfinite(n):
        return float('nan')
    return ((n - 1.0) / n) / ((gamma_local - 1.0) / gamma_local)

stage_eff_list = []
for entry in comp_res["stage_info"]:
    if isinstance(entry["stage"], int):  # skip intercooler steps
        i = entry["stage"]
        T_in = entry["T_in_K"]
        T2s = entry["T2s_K"]
        T2_act = entry["T2_act_K"]
        p_in = entry["p_in_bar"] * 1e5
        p_out = entry["p_out_bar"] * 1e5
        eta_isent = stage_isentropic_efficiency(T_in, T2s, T2_act)
        n_poly = stage_polytropic_exponent(T_in, T2_act, p_in, p_out)
        eta_poly = stage_polytropic_efficiency(n_poly, gamma_air)
        stage_eff_list.append({
            "Stage": i,
            "η_isentropic": eta_isent,
            "n_polytropic": n_poly,
            "η_polytropic": eta_poly
        })

valid_eta_is = [x["η_isentropic"] for x in stage_eff_list if math.isfinite(x["η_isentropic"])]
valid_eta_poly = [x["η_polytropic"] for x in stage_eff_list if math.isfinite(x["η_polytropic"])]
avg_isent = sum(valid_eta_is)/len(valid_eta_is)
```

```
avg_poly = sum(valid_eta_poly)/len(valid_eta_poly)

print("\n=== Compressor Stage Efficiency Diagnostics ===")
for x in stage_eff_list:
    print(f"Stage {x['Stage']}: η_isentropic={x['η_isentropic']:.4f}, "
          f"n_polytropic={x['n_polytropic']:.3f}, η_polytropic={x['η_polytropic']:.4f}")
print(f"Average isentropic efficiency = {avg_isent:.4f}")
print(f"Average polytropic efficiency = {avg_poly:.4f}")

if HAS_MPL:
    plt.figure(figsize=(8,5))
    stages = [x["Stage"] for x in stage_eff_list]
    eta_is = [x["η_isentropic"] for x in stage_eff_list]
    eta_poly = [x["η_polytropic"] for x in stage_eff_list]
    plt.plot(stages, eta_is, marker='o', label='Isentropic Efficiency')
    plt.plot(stages, eta_poly, marker='s', label='Polytropic Efficiency')
    plt.xlabel("Compressor Stage")
    plt.ylabel("Efficiency")
    plt.title("Isentropic vs. Polytropic Efficiency per Compressor Stage")
    plt.ylim(0, 1.05)
    plt.grid(True)
    plt.legend()
    plt.tight_layout()
    plt.show()
else:
    print("matplotlib not available — skipping plot.")
```

```
=== Integration sweep summary ===
Compressor: 2-stage inlet 10.0 bar, outlet 32.0 bar, flow 29.22 kg/s
Estimated baseline turbine mass flow = 421.320 kg/s (from 166.0 MW baseline)
Compressor shaft power (approx) = 3.7738 MW

TIT (K) | gross_turb_MW | net_elec_after_MW | net_gain_MW
   1200 |       317.181 |           305.525 |     139.525
   1225 |       324.625 |           312.784 |     146.784
   1250 |       332.105 |           320.078 |     154.078
   1275 |       339.620 |           327.407 |     161.407
   1300 |       347.171 |           334.770 |     168.770
   1325 |       354.759 |           342.169 |     176.169
   1350 |       362.382 |           349.603 |     183.603
   1375 |       370.041 |           357.072 |     191.072
   1400 |       377.737 |           364.577 |     198.577
   1425 |       385.470 |           372.117 |     206.117
   1450 |       393.238 |           379.692 |     213.692
   1475 |       401.044 |           387.304 |     221.304
   1500 |       408.886 |           394.951 |     228.951
```



APPENDIX D

Two-Stage Compressor Turbine Integration T-S Diagram

```
import math

try:
    import matplotlib.pyplot as plt
    HAS_MPL = True
except Exception:
    HAS_MPL = False

p1_bar = 10          # compressor inlet pressure (bar)
T1_C = 28.7          # compressor inlet temperature (°C)
p2_bar = 32          # compressor discharge pressure (bar)
T2_measured_C = 82.6    # measured compressor outlet temp (°C) (for diagnostics)
m_comp = 29.22          # kg/s (compressor mass flow)

# Turbine / generator (given)
P_elec_baseline_MW = 166.0   # baseline electrical (MW)
eta_gen = 0.85
eta_mech = 0.80
eta_overall_conv = eta_gen * eta_mech

# Combustor / fuel
LHV = 50e6           # J/kg fuel (50 MJ/kg)
eta_comb = 0.80
```

```
n_stages = 2
eta_stage = 0.83          # isentropic efficiency per stage
intercool_eff = 0.85       # intercooler effectiveness between stage1 and stage2
T_coolant_C = T1_C         # intercooler coolant
TIT_target_K = 1400.0      # turbine inlet temp (K)
eta_turbine = 0.75
pressure_loss_frac = 0.02

def cp_air_coeffs():
    b = 0.1
    A = 1003.5 - 0.1 * 300.0
    return A, b  # cp(T) = A + b*T

def cp_gas_coeffs():
    b = 0.12
    A = 1150.0 - 0.12 * 1000.0
    return A, b

R = 287.0
gamma_air = 1.4
gamma_gas = 1.33


def to_K(T_C): return T_C + 273.15
def bar_to_Pa(p_bar): return p_bar * 1e5

T1 = to_K(T1_C)
T2_measured = to_K(T2_measured_C)
p1 = bar_to_Pa(p1_bar)
p2 = bar_to_Pa(p2_bar)
p_turb_out = 1.0 * 1e5
P_mech_baseline_W = (P_elec_baseline_MW * 1e6) / eta_gen
assumed_spec_work_J_per_kg = 400e3
m_turb_baseline = P_mech_baseline_W / assumed_spec_work_J_per_kg

def delta_h_from_linear_cp(A, b, T1, T2):

    return A * (T2 - T1) + 0.5 * b * (T2 * T2 - T1 * T1)

def delta_s_from_linear_cp(A, b, T1, T2, p1_local, p2_local):
    # Δs = ∫(A + b*T)/T dT - R ln(p2/p1)
    # ∫(A/T + b) dT = A ln(T2/T1) + b*(T2 - T1)
    return A * math.log(T2 / T1) + b * (T2 - T1) - R * math.log(p2_local / p1_local)

def two_stage_compressor(T_in, p_in, p_out, m_dot, n_stages=2, eta_stage=0.86,
intercool_eff=0.9, T_coolant=None):
    """
    Returns:
      dict with keys:
```

```
        - stage_info: list of dicts for stages
        - T_out: final outlet T (K)
        - total_work_per_kg (J/kg)
        - P_shaft_W (W)
        - stage_entropy_changes (list)
    """
    if T_coolant is None:
        T_coolant = T_in
    # per-stage pressure ratio (equal split)
    pr_stage = (p_out / p_in) ** (1.0 / n_stages)
    stage_info = []
    T_stage_in = T_in
    total_work_per_kg = 0.0
    total_entropy_change = 0.0
    A_air, b_air = cp_air_coeffs()

    for i in range(1, n_stages + 1):
        p_stage_in = p_in * (pr_stage ** (i - 1))
        p_stage_out = p_in * (pr_stage ** i)
        # isentropic (ideal) outlet temp
        T2s = T_stage_in * (pr_stage ** ((gamma_air - 1.0) / gamma_air))
        # actual outlet temp using stage efficiency
        T2_act = T_stage_in + (T2s - T_stage_in) / eta_stage

        dh_stage = delta_h_from_linear_cp(A_air, b_air, T_stage_in, T2_act)
        # entropy change for compression stage
        ds_stage = delta_s_from_linear_cp(A_air, b_air, T_stage_in, T2_act, p_stage_in,
p_stage_out)
        total_work_per_kg += dh_stage
        total_entropy_change += ds_stage
        stage_info.append({
            "stage": i,
            "p_in_bar": p_stage_in / 1e5,
            "p_out_bar": p_stage_out / 1e5,
            "T_in_K": T_stage_in,
            "T2s_K": T2s,
            "T2_act_K": T2_act,
            "dh_stage_J_per_kg": dh_stage,
            "ds_stage_J_per_kgK": ds_stage
        })

        if i < n_stages:

            T_after_ic = T2_act - intercool_eff * (T2_act - T_coolant)

            ds_cool = (A_air * math.log(T_after_ic / T2_act) + b_air * (T_after_ic - T2_act))
            total_entropy_change += ds_cool
            T_stage_in = T_after_ic

            stage_info.append({
```

```
            "stage": "IC_after_stage_" + str(i),
            "T_after_ic_K": T_after_ic,
            "ds_cool_J_per_kgK": ds_cool
        })
    else:
        T_stage_in = T2_act

    T_out = T_stage_in
    P_shaft_W = m_dot * total_work_per_kg
    return {
        "stage_info": stage_info,
        "T_out_K": T_out,
        "total_work_per_kg_J": total_work_per_kg,
        "P_shaft_W": P_shaft_W,
        "total_entropy_change_J_per_kgK": total_entropy_change
    }

def combustor_fuel_flow(m_air, T_air_in, T_TIT, LHV, eta_comb):
    # cp_gas coefficients
    A_gas, b_gas = cp_gas_coeffs()
    A_air, b_air = cp_air_coeffs()

    h_air_in = delta_h_from_linear_cp(A_air, b_air, 0.0, T_air_in)  # h referenced from 0 K
(consistent use cancels)
    h_gas_TIT = delta_h_from_linear_cp(A_gas, b_gas, 0.0, T_TIT)
    # energy balance: m_air*h_air_in + m_fuel*LHV*eta_comb = (m_air +
m_fuel)*h_gas_TIT
    # rearrange: m_fuel*(LHV*eta_comb - h_gas_TIT) = m_air*(h_gas_TIT - h_air_in)
    denom = (LHV * eta_comb - h_gas_TIT)
    if denom <= 0:
        return float('nan')
    m_fuel = m_air * (h_gas_TIT - h_air_in) / denom
    return m_fuel


def turbine_expansion_power(m_air_total, m_fuel, T_TIT, p_in, p_out, eta_turbine):
    # total mass through turbine
    m_total = m_air_total + m_fuel
    # cp_gas linear coeffs
    A_gas, b_gas = cp_gas_coeffs()
    # isentropic exit temperature
    T_exit_s = T_TIT * (p_out / p_in) ** ((gamma_gas - 1.0) / gamma_gas)
    # actual exit temperature
    T_exit = T_TIT - eta_turbine * (T_TIT - T_exit_s)
    # enthalpy change per kg of gas (h_in - h_out)
    h_in = delta_h_from_linear_cp(A_gas, b_gas, 0.0, T_TIT)
    h_out = delta_h_from_linear_cp(A_gas, b_gas, 0.0, T_exit)
    dh_per_kg = h_in - h_out
    gross_power_W = m_total * dh_per_kg
    return gross_power_W, T_exit, T_exit_s, dh_per_kg
```

```
comp_res = two_stage_compressor(T1, p1, p2, m_comp, n_stages=n_stages,
                                eta_stage=eta_stage, intercool_eff=intercool_eff,
                                T_coolant=T_coolant_C + 273.15)

T_comp_out = comp_res["T_out_K"]
P_comp_MW = comp_res["P_shaft_W"] / 1e6
w_comp_kJ_per_kg = comp_res["total_work_per_kg_J"] / 1000.0

m_air_total = m_turb_baseline + m_comp

m_fuel = combustor_fuel_flow(m_air_total, T_comp_out, TIT_target_K, LHV, eta_comb)

if math.isnan(m_fuel) or m_fuel <= 0:
    print("TIT infeasible with current assumptions (LHV/eta_comb/cp).")
else:

    p_turb_in = p2 * (1.0 - pressure_loss_frac)
    gross_turb_W, T_exit_actual, T_exit_s, dh_per_kg =
turbine_expansion_power(m_air_total, m_fuel, TIT_target_K,
                                                  p_turb_in, p_turb_out, eta_turbine)
    gross_turb_MW = gross_turb_W / 1e6
    net_elec_after_MW = (gross_turb_W * eta_overall_conv) / 1e6 - P_comp_MW
    net_gain_MW = net_elec_after_MW - P_elec_baseline_MW


    print("=== Two-stage compressor integrated with turbine (results) ===")
    print(f"Compressor: inlet p={p1/1e5:.3f} bar, inlet T={T1:.2f} K")
    print(f"Compressor: outlet p={p2/1e5:.3f} bar, computed outlet T={T_comp_out:.2f} K")
    print(f"Compressor: total specific work = {w_comp_kJ_per_kg:.4f} kJ/kg")
    print(f"Compressor shaft power = {P_comp_MW:.6f} MW (for m_comp={m_comp}
kg/s)")
    print()
    print(f"Baseline turbine mass flow used (estimate) = {m_turb_baseline:.3f} kg/s")
    print(f"Total air into combustor = m_turb_baseline + m_comp = {m_air_total:.3f} kg/s")
    print(f"Fuel flow required to reach TIT={TIT_target_K:.1f} K: m_fuel = {m_fuel:.6f}
kg/s")
    print()
    print(f"Turbine gross power = {gross_turb_MW:.6f} MW")
    print(f"Turbine exit actual T = {T_exit_actual:.2f} K (isentropic exit {T_exit_s:.2f} K)")
    print(f"Net electrical after integration (accounting mech+gen conv and compressor load) =
{net_elec_after_MW:.6f} MW")
    print(f"Net gain vs baseline = {net_gain_MW:.6f} MW")
    print()


print("Compressor stage-by-stage details:")
for entry in comp_res["stage_info"]:
    stage_name = str(entry.get("stage", ""))
    if stage_name.startswith("IC_after"):
```

```
        print(f"  {stage_name}: T_after_ic={entry['T_after_ic_K']:.2f} K,
ds_cool={entry['ds_cool_J_per_kgK']:.6f} J/kgK")
    else:
        print(f"  Stage {stage_name}: "
              f"p_in={entry['p_in_bar']:.3f} bar, p_out={entry['p_out_bar']:.3f} bar, "
              f"T_in={entry['T_in_K']:.2f} K, T2s={entry['T2s_K']:.2f} K,
T2_act={entry['T2_act_K']:.2f} K, "
              f"dh_stage={entry['dh_stage_J_per_kg']/1000.0:.4f} kJ/kg,
ds_stage={entry['ds_stage_J_per_kgK']:.6f} J/kgK")


def cp_ab_air():
    return cp_air_coeffs()  # returns A,b

def cp_ab_gas():
    return cp_gas_coeffs()


A_air, b_air = cp_air_coeffs()
A_gas, b_gas = cp_gas_coeffs()

# s1 = 0 (reference)
s1 = 0.0
s2 = s1 + (A_air * math.log(T_comp_out / T1) + b_air * (T_comp_out - T1) - R *
math.log((p2) / (p1)))
# combustor heat add is at p ~ p_turb_in (slightly different), use p2 -> p_turb_in as approx
s3 = s2 + (A_gas * math.log(TIT_target_K / T_comp_out) + b_gas * (TIT_target_K -
T_comp_out) - R * math.log(p_turb_in / p2))
s4 = s3 + (A_gas * math.log(T_exit_actual / TIT_target_K) + b_gas * (T_exit_actual -
TIT_target_K) - R * math.log(p_turb_out / p_turb_in))

# prepare arrays for plotting closed loop [1->2->3->4->1]
TS_S = [s1, s2, s3, s4, s1]
TS_T = [T1, T_comp_out, TIT_target_K, T_exit_actual, T1]

if HAS_MPL:
    plt.figure(figsize=(8,6))
    plt.plot(TS_S, TS_T, marker='o', linestyle='-', label='Cycle (with 2-stage + intercool)')

    plt.text(s1, T1, "1\n(T1)")
    plt.text(s2, T_comp_out, "2\n(comp out)")
    plt.text(s3, TIT_target_K, "3\n(TIT)")
    plt.text(s4, T_exit_actual, "4\n(turb out)")
    plt.xlabel("Entropy (J/kg·K) (relative)")
    plt.ylabel("Temperature (K)")
    plt.title("T–S diagram: Brayton cycle with 2-stage compressor (intercool between stages)")
    plt.grid(True)
    plt.legend()
    plt.show()
else:
```

print("matplotlib not available; T-S plot not shown. Install matplotlib to see the diagram.")

```
Compressor: inlet p=10.000 bar, inlet T=301.85 K
Compressor: outlet p=32.000 bar, computed outlet T=372.98 K
Compressor: total specific work = 129.1527 kJ/kg
Compressor shaft power = 3.773843 MW (for m_comp=29.22 kg/s)

Baseline turbine mass flow used (estimate) = 421.320 kg/s
Total air into combustor = m_turb_baseline + m_comp = 450.540 kg/s
Fuel flow required to reach TIT=1400.0 K: m_fuel = 11.179316 kg/s

Turbine gross power = 377.737324 MW
Turbine exit actual T = 692.00 K (isentropic exit 595.45 K)
Net electrical after integration (accounting mech+gen conv and compressor load) = 364.576709 MW
Net gain vs baseline = 198.576709 MW

Compressor stage-by-stage details:
  Stage 1: p_in=10.000 bar, p_out=17.889 bar, T_in=301.85 K, T2s=356.41 K, T2_act=365.30 K, dh_stage=63.8823 kJ/kg,
  IC_after_stage_1: T_after_ic=308.19 K, ds_cool=-171.185238 J/kgK
  Stage 2: p_in=17.889 bar, p_out=32.000 bar, T_in=308.19 K, T2s=363.91 K, T2_act=372.98 K, dh_stage=65.2705 kJ/kg,
```