

Article

Thermodynamic Performance of Heavy-Duty Gas Turbines with Hydrogen–Ammonia Co-Fuel by Inlet Guide Vane Variations

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Abstract

When methane (natural gas) fuel gas turbines switch to hydrogen–ammonia co-fuel, their thermodynamic performance undergoes significant changes. To expand the operating range of heavy-duty gas turbines when using hydrogen–ammonia co-fuel and to improve their thermodynamic performance, based on the 255.6 MW heavy-duty gas turbine at the Banshan Power Plant in Hangzhou, China, a simulation model was established. A strategy based on changing the angle of the compressor inlet guide vanes (IGVs) was proposed. The thermodynamic performance, turbine stage supersonic flow velocity, and flow matching characteristics of gas turbines were studied under different hydrogen–ammonia mixing ratios. The results indicate that the developed model can accurately predict the performance of the gas turbine under rated operating conditions, yielding a rated output power of 254.59 MW and an efficiency of 36.33%, with relative errors of −0.4% and −1.54% compared with the design values, respectively. When hydrogen–ammonia blended fuel is employed, the outlet Mach numbers of the second- and third-stage turbine stators exceed the safety limit unity. Reducing the IGV angle effectively decreases the turbine stator outlet Mach number and improves operational safety, although a slight reduction in gas turbine efficiency is observed. As the ammonia volumetric fraction in the blended fuel increases, the gas turbine output power increases while the efficiency decreases slightly, accompanied by a reduction in turbine stator outlet pressure and an increase in outlet temperature. Further investigation shows that, after IGV regulation, the combustor outlet pressure, gas turbine power output, and efficiency all increase. Under a fixed IGV opening condition, the gas turbine efficiency gradually decreases with increasing ammonia volumetric fraction. Under off-design fuel flow conditions, increasing the relative fuel flow leads to higher combustor outlet pressure and temperature, whereas increasing the ammonia volumetric fraction causes a slight reduction in these parameters. This research can provide theoretical support for the optimal design and operation of gas turbines using hydrogen–ammonia mixed fuel.

Keywords: heavy-duty gas turbine; hydrogen-ammonia co-fuel; thermodynamic performance; IGV variation; Mach number

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1. Introduction

As a green fuel, the hydrogen–ammonia co-fuel gas turbine has great potential for development and has become a current research hotspot [1–3].

In terms of modeling, Zhang Zhaoyu et al. [4–7] conducted systematic research on heavy-duty gas turbine dynamic modeling based on operational data, addressing the challenges of strong parameter coupling, high nonlinearity, and the limitations of traditional first-principles models during start-up and off-design operation. Using field operation data, a nonlinear autoregressive model with exogenous inputs (NARX) neural network was introduced to develop dynamic models for turbine start-up and loading processes. The proposed approach enables time-series prediction of key operating parameters such as rotational speed, temperature, and output power, demonstrating the feasibility and effectiveness of data-driven methods in gas turbine dynamic modeling.

Jordal et al. [8] developed a stage-by-stage turbine cooling model to evaluate the impact of blade cooling on gas turbine performance. Based on known blade geometric parameters, an iterative calculation was performed along the blade span to determine the temperature distribution on both blade surfaces. The required cooling air flow was then derived by enforcing the maximum allowable material temperature, and the total cooling air consumption was obtained for the entire blade. The developed cooling model was further coupled with a gas turbine thermodynamic cycle model to analyze part-load steady-state performance under varying turbine inlet temperature and pressure ratio conditions.

Mehrpanahi et al. [9,10] developed a gray-box dynamic modeling approach for industrial three-shaft gas turbines under loading and unloading conditions. By combining thermodynamic first-principles models with empirical correlations in the Simulink environment, the model reduces computational complexity while improving accuracy and sensitivity. The proposed model is capable of capturing dynamic system responses under both design and off-design conditions, and has been applied to analyze transient behaviors during start-up, load variation, and fuel change processes in heavy-duty gas turbines.

For large-scale gas turbine systems, Bothien et al. [11] investigated the applicability of sequential combustion under high-hydrogen conditions based on the GT36 gas turbine. The results show that by adjusting the fuel distribution between multiple combustor stages, the turbine inlet temperature can be maintained without hardware modification, enabling stable operation over a wide range of hydrogen blending ratios. When the hydrogen volume fraction is below approximately 70%, rated operation can be achieved, while at higher hydrogen fractions, reducing the combustion temperature is required to ensure safe operation.

Kim et al. [12] investigated the application potential of hydrogen, ammonia, and their blended fuels in gas turbine systems from a thermodynamic perspective. Based on a steady-state thermodynamic model, variations in output power, cycle efficiency, and key thermodynamic parameters under different fuel compositions were systematically analyzed, with particular emphasis on the performance improvement induced by hydrogen addition to ammonia-based gas turbines. The results indicate that an appropriate hydrogen–ammonia blending ratio can mitigate the adverse effects of fuel substitution on cycle performance and help maintain thermal efficiency under various operating conditions.

Pashchenko et al. [13] proposed a chemically recuperated gas turbine concept based on exhaust heat-driven ammonia cracking to improve system efficiency and combustion performance in hydrogen–ammonia fuel systems. In this approach, high-temperature turbine exhaust is utilized to provide the reaction heat required for ammonia decomposition, producing hydrogen-rich fuel for combustion without the need for external hydrogen supply. Comparative analyses among conventional ammonia-fired gas turbines, combined cycles, and chemically recuperated systems indicate that exhaust heat-driven ammonia cracking can significantly improve cycle efficiency. In addition, the generated

hydrogen-rich fuel enhances flame propagation speed and combustion stability due to the increased hydrogen content. From a system integration perspective, this “ammonia cracking–hydrogen production–combustion” concept demonstrates strong potential for improving the performance of hydrogen–ammonia gas turbine systems.

However, the change of fuel will significantly alter the required fuel flow rate, causing changes in the thermal performance of the gas turbine, resulting in the turbine intake volume exceeding the flow capacity limit, and causing supersonic flow velocity within the turbine [13]. This leads to safety hazards and performance deterioration of the gas turbine [14].

Wu Haonan et al. [15] and Lv et al. [16] pointed out that when methane gas turbines switched to hydrogen-rich fuels, the outlet flow velocity of the stator blades encountered a supercritical problem. They proposed the method of increasing the installation angle of the turbine blades. The gas turbine was successfully operated safely under different hydrogen content conditions, and its power and efficiency did not experience significant reduction.

Yu et al. [17] investigated flow blockage in an operational F-class heavy-duty gas turbine during hydrogen co-firing. Based on actual operational data, they developed an integrated gas turbine model. Under safety constraints such as maintaining the compressor surge margin, they systematically analyzed the effects of varying hydrogen blending ratios on thermodynamic performance and flow matching characteristics. Furthermore, they proposed a matching strategy involving reduced turbine inlet temperature, which successfully lowered both turbine inlet and outlet temperatures while preserving the surge margin, thereby ensuring safe operation. Li Yongyi et al. [18] studied the constraints of different operating conditions on the flow matching of the gas turbine, and obtained the effects of different air separation and nitrogen injection on the performance of the gas turbine.

Wei et al. [19] investigated the impact of modified compressor and turbine flow capacity on the overall matching characteristics of a retrofitted gas turbine. To address the difficulty in reconstructing flow characteristics after component modification and the complexity introduced by cooling air and off-design operation, a turbine through-flow model was developed based on fundamental governing equations and loss correlations. The model considers both subcritical and supercritical operating conditions as well as cooled and uncooled cases, and enables system-level through-flow matching analysis using an equivalent flow path approach.

Zhang et al. [20] investigated gas turbine systems operating on low- and medium-calorific-value fuels, focusing on the effects of reduced fuel heating value on increased fuel flow rate, altered flow capacity, and compressor–turbine operating point rematching. Several adjustment strategies were proposed, including increasing compressor pressure ratio, adjusting inlet guide vanes, reducing turbine inlet temperature, implementing compressor bleed, and enlarging turbine flow area. Thermoeconomic and operational safety analyses of simple-cycle gas turbines under different fuel conditions indicate that these strategies exhibit significant differences in efficiency and safety performance. The results further suggest that differentiated through-flow and operating strategies should be adopted for different low-calorific fuels such as coke oven gas and coal-based syngas.

Pashchenko et al. [13] proposed a chemically recuperated gas turbine (CRGT) concept driven by exhaust waste heat to enhance the system efficiency and combustion performance of hydrogen–ammonia-fueled gas turbine systems, and systematically evaluated its thermodynamic performance. Comparative analyses were conducted among the conventional ammonia-fueled gas turbine, the combined cycle system, and the chemically recuperated combined cycle configuration.

Paolo C et al. [21] investigated the feasibility of replacing natural gas with hydrogen fuel in large-scale heavy-duty gas turbines. To address challenges arising from hydrogen combustion, including the significant increase in working fluid volumetric flow rate, deterioration of turbine cooling performance, and mismatch between compressor and turbine flow characteristics, several adaptive modification strategies were proposed, including the application of variable guide vanes, increasing the compressor pressure ratio, and redesigning the overall gas turbine configuration.

Camporeale P et al. [22] conducted a study on system matching and performance variation of an in-service F-class heavy-duty gas turbine under hydrogen co-firing conditions. Based on actual operating data, an integrated gas turbine model was established. Considering operational safety constraints such as compressor surge margin, the effects of different hydrogen blending ratios on the thermodynamic performance and flow matching characteristics of the gas turbine were systematically analyzed.

In addition, several studies have investigated the combustion and emission characteristics of hydrogen–ammonia fuels in gas turbines [23,24], providing mechanistic insights into approaches for improving combustion performance and reducing NO_x emissions.

Companies such as Siemens, General Electric, Ansardo, and Shanghai Electric are developing gas turbines that can use hydrogen and ammonia fuels, and have studied the impact of hydrogen blending ratios on emissions and combustion performance [25,26]. To reduce NO_x emissions, some researchers have developed corresponding technologies, significantly improving the emission performance [27,28]. The research focus remains on limiting NO_x emissions, and the hydrogen blending ratio in heavy-duty gas turbines is relatively low [29].

This paper establishes a thermodynamic dynamic model of a heavy-duty gas turbine considering the influence of cooling air, and verifies the model's accuracy. The thermal performance of the gas turbine using hydrogen–ammonia fuel (with ammonia proportions ranging from 0% to 100%) is calculated. By changing the angle of the IGV, the gas turbine can adapt to different proportions of mixed fuels. The thermodynamic performance of the gas turbine under the condition of using hydrogen–ammonia blended fuel is analyzed. The research results provide technical references for the design of hydrogen–ammonia blended gas turbines.

2. Mathematical Modeling of Gas Turbines

This study refers to the heavy-duty gas turbine in the gas–steam combined cycle of Hangzhou Banshan Power Plant. This heavy-duty gas turbine compressor has a total of 18 stages. There are cooling exhaust holes at the 9th, 13th and 17th stages. The turbine has three stages. The rotor and stator blades of the 1st and 2nd stages, as well as the stator blades of the 3rd stage, are cooled by air. The schematic diagram of the gas turbine compressor is shown in Figure 1.

Using the CREAFORM MetraScan scanning equipment, a three-dimensional scan of the compressor blades of the 9FA unit at Hangzhou Banshan Power Plant was conducted to obtain the point cloud data of the blades. After noise reduction and smoothing processing of the data, the geometric parameters of the blade profiles at different blade heights were extracted. Then, third-order B-spline curve approximation fitting was carried out. Finally, through section profile lofting in the SolidWorks 2025 SP02 LITE software, the three-dimensional reconstruction of the IGV, R0 and S0 blades was achieved.

The characteristic parameters of the blade profiles at the mid-diameter for IGV, R0 and S0 are shown in Table 1.

The three-dimensional blade shape scanning, processing of blade point cloud data, and reconstruction of the blade cross-sectional profiles are shown in Figure 2. The actual aerodynamic and geometric parameters of the blades are obtained from physical blade 3D

scanning measurements. Based on the scanned data, key geometric and aerodynamic characteristics are extracted and subsequently used in the stage-by-stage thermodynamic and aerodynamic modeling of the compressor and turbine.

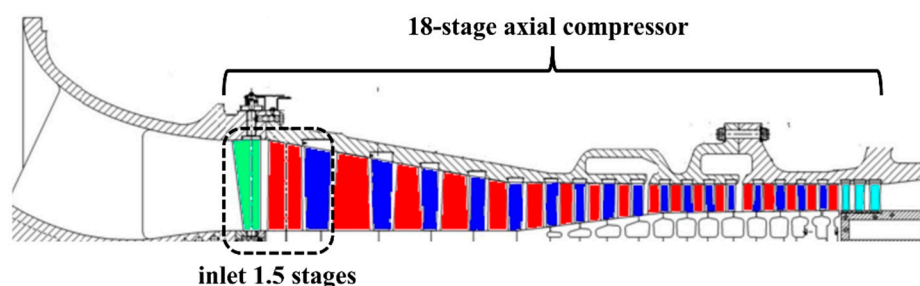
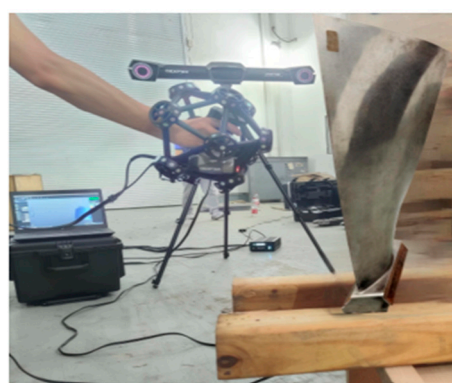


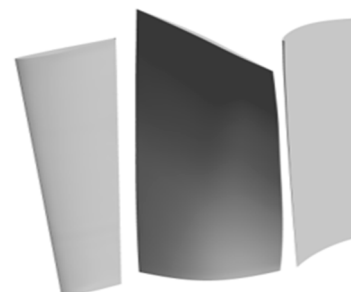
Figure 1. Schematic diagram of an 18-stage axial flow compressor for heavy-duty gas turbine.

Table 1. Characteristic profile parameters at the mean diameter of IGV, R0, and S0 blades.

Blade Profile Characteristic Parameters	IGV	R0	S0
Leading edge radius/mm	0.65	0.81	0.98
Trailing Edge Radius/mm	0.51	0.67	0.60
Pitch/mm	98.18	194.39	132.49
Chord length/mm	129.95	240.68	148.21
Maximum Thickness/mm	6.64	11.99	11.64
Blade Setting Angle/(°)	4.30	47.19	26.53
Blade Camber/(°)	17.95	10.12	30.09



(a)



(b)

Figure 2. Three-dimensional point cloud scanning and reconstruction of compressor IGV, R0 and S0 blades. (a) On-site scanning of compressor blade parameters. (b) Point cloud models of IGV, R0 and S0 blades.

The schematic diagram of the gas turbine structure and the distribution of cooling air is shown in Figure 3. In this paper, the compressor is divided into three sections according to the positions of the cooling exhaust ports. The 0–9, 10–13, and 14–17 represent the stages of the three sections of the compressor. The cooling air flow rates for the three sections are based on the values in Reference [16].

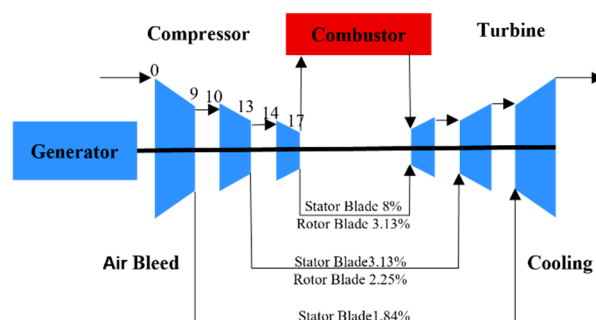


Figure 3. Schematic diagram of air distribution for compressor cooling.

2.1. Compressor Model

The compressor raises the temperature and pressure of air before delivering it to the combustion chamber. Its flow characteristics can be determined using a compressor characteristic fitting formula [27]:

$$\dot{G} = A \times \left(1 - \left(\frac{\dot{n}}{B}\right)^2\right)^{1/2} \quad (1)$$

where A and B are fitting coefficients, $\dot{G}$ is the corrected mass flow rate, and $\dot{n}$ is the corrected rotational speed. Their relationship is given as follows:

$$A = 4.328 \times \dot{n}^2 - 3.623 \times \dot{n} + 0.975 \quad (2)$$

$$B = 1.618 \times \dot{n} - 0.3968 \quad (3)$$

$$\dot{G} = \frac{G\sqrt{T}}{P} \bigg/ \frac{G_0\sqrt{T_0}}{P_0} \quad (4)$$

$$\dot{n} = \frac{n}{\sqrt{T}} \bigg/ \frac{n_0}{\sqrt{T_0}} \quad (5)$$

where G_0 and G are the compressor inlet mass flow rates ($\text{kg}\cdot\text{s}^{-1}$) under the rated condition and the actual operating condition, respectively; P_0 and P are the compressor inlet pressures (Pa) under the rated condition and the actual operating condition, respectively; n_0 and n are the gas turbine speeds (rpm) under the rated condition and the actual operating condition, respectively; $\dot{n}$ is the specific corrected speed; T_0 and T represent the compressor inlet temperatures (K) under the rated condition and the actual operating condition, respectively.

In Figure 4, when the IGV angle is decreased, the airflow entering the compressor is reduced, leading to a modification of the compressor characteristic map. In this study, the effect of IGV variation on compressor performance is represented using an empirical correction method. Specifically, for each 1° reduction in the IGV opening angle, both the corrected normalization mass flow rate and the normalization pressure ratio are assumed to decrease by 1% under standard conditions [29,30]. The implementation of this correction follows the method described in refs. [29,30].

C_s denotes the relative corrected speed [29,30]:

$$C_s = 100 \times \frac{n}{n_0} \sqrt{\frac{RT_0}{G_0} \bigg/ \frac{RT}{G}} \quad (6)$$

where R is the gas constant.

When the IGV angle changes, the compressor efficiency also changes. Its influence on efficiency is given by the following formula [22,30,31]:

$$\eta = \eta_n \cdot (1 - \text{angle} \cdot \text{VEC}) \quad (7)$$

where η is the compressor efficiency; *angle* is the IGV closing angle($^{\circ}$), with the subscript n denoting the corresponding point on the original characteristic curve, η_n represents the compressor efficiency under the fully open IGV condition, and *VEC* is the correction coefficient, taken here as 0.0022 [22].

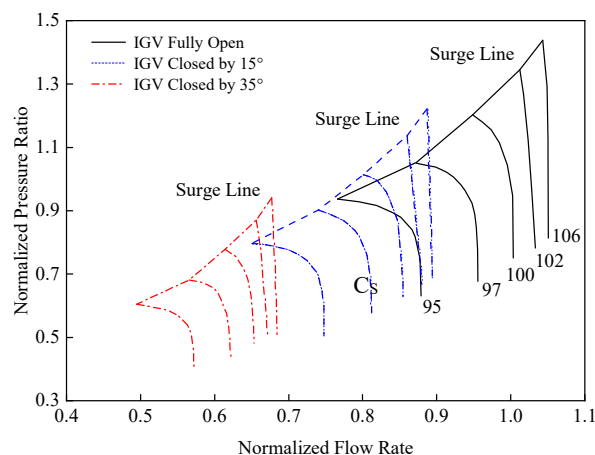


Figure 4. Characteristic curves of compressor performance under different inlet guide vane closure angles and normalized flow rate.

The computational process for the compressor stages is divided into two parts: the rotor blades and the stator blades. The related calculations primarily revolve around the velocity triangles and the modeling is conducted using a stage-by-stage accumulation method. Inter-stage pressure and mass flow losses caused by cooling air extraction must be considered in the model, for which the specific methodology can be referenced from the previous work of our research group [15]. The procedure for the stage-by-stage and segment-by-segment calculations of the compressor is entirely analogous, with the specific workflow illustrated in Figure 5.

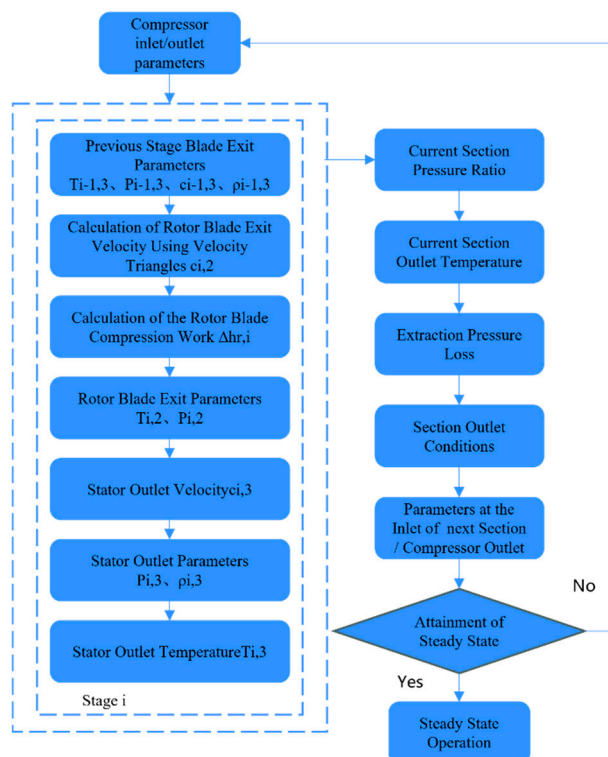


Figure 5. Calculation process for each stage and section of compressor.

Where $i = 0\text{--}17$ denotes the compressor stage number; $j = 1\text{--}3$ represent the rotor inlet, rotor outlet, and stator outlet sections, respectively; $T_{i,j}$, $p_{i,j}$, $c_{i,j}$, and $\rho_{i,j}$ denote the temperature (K), pressure (Pa), absolute flow velocity (m/s), and flow density (kg/m³) at section j of the i -th compressor stage, respectively; and $\Delta h_{r,i}$ is the compression work of the rotor in the i -th compressor stage (J/kg).

2.2. Combustion Chamber Model

The combustion process of the working fluid within the combustion chamber can be modeled as a constant-pressure heating process. This study primarily focuses on the combustor outlet temperature and the internal pressure loss. The dynamic combustion process inside the combustion chamber is governed by a differential equation derived from the energy conservation principle [32].

$$\tau_{cc} \frac{dT_{out}}{dt} = \frac{G_{in}h_{in} + G_{fuel}(h_{fuel} + \eta_{cc}LHV) - G_{out}h_{out}}{G_{out}c_{pout}} \quad (8)$$

where η_{cc} is combustion chamber efficiency, taken as 0.99 in this study. τ_{cc} is combustion chamber time constant. G_{in} , G_{out} are mass flow rates of gas entering and leaving the combustion chamber, respectively. G_{fuel} is fuel mass flow rate. h_{in} , h_{out} are enthalpy of air entering and leaving the combustion chamber. h_{fuel} is enthalpy of the fuel. LHV is lower heating value of the fuel. c_{pout} is enthalpy of the flue gas at the combustion chamber outlet. The calculation method for the combustion chamber time constant τ_{cc} used in this paper refers to [22,33].

Pressure losses occur within the combustion chamber. To account for the pressure loss of the gas flowing through the chamber, the total pressure recovery coefficient ε_{cc} is introduced, which is taken as 0.975 [22,32] in this text. The calculation formulas for the specific enthalpy and specific heat of the relevant gases are adopted from ref. [33].

$$p_{3,cc}^* = p_{2,cc}^* \cdot \varepsilon_{cc} \quad (9)$$

where $p_{2,cc}^*$, $p_{3,cc}^*$ are the stagnation pressures at the combustor inlet and outlet, respectively.

Different fuels consist of hydrogen–ammonia mixtures with different volumetric blending ratios. Assuming complete combustion, the composition of the combustion products is determined according to the fuel composition. The fuel flow rate is obtained iteratively while maintaining a constant turbine inlet temperature. The specific enthalpy and specific heat at constant pressure of the relevant gases are treated as functions of temperature. The detailed calculation method can be found in Ref. [34].

2.3. Turbine Model

The flow characteristics of the turbine can be determined using the Flügel formula [35]:

$$\frac{G_T}{G_{T0}} = \sqrt{\frac{T_{30,T}^*}{T_{3,T}^*}} \sqrt{1 - 0.4 \frac{\Delta n}{n_0} \sqrt{\frac{p_{3,T}^{*2} - p_{4,T}^{*2}}{p_{30,T}^{*2} - p_{40,T}^{*2}}}} \quad (10)$$

where G_{T0} , G_T are the turbine inlet mass flow rates under the rated condition and the actual operating condition, respectively; $T_{30,T}^*$, $T_{3,T}^*$ represent the turbine inlet stagnation temperatures under the rated condition and the actual operating condition, respectively; $p_{30,T}^*$ and $p_{3,T}^*$ are the turbine inlet stagnation pressures of the working fluid at the initial turbine inlet section under the rated condition and the actual operating condition, respectively; Δn is the rotational speed variation.

The turbine efficiency is calculated using the following formula [36]:

$$\eta_T = \left[1 - 0.3 \left(1 - \frac{n}{n_0} \right)^2 \right] (2q - q^2) \quad (11)$$

$$q = \frac{nG_{T0}}{n_0G_T} \quad (12)$$

Based on the measured dimensional data of the turbine blades and in conjunction with the velocity triangles of the turbine stage as shown in Figure 6, the thermodynamic parameters for each turbine stage can be determined, where c_1 and c_2 are the absolute velocities of the airflow at the outlet of the stator blade and the inlet of the rotor blade respectively; w_1 and w_2 are the relative velocities of the airflow at the outlet of the stator blade and the inlet of the rotor blade respectively; u is the circumferential velocity of the rotor blade.

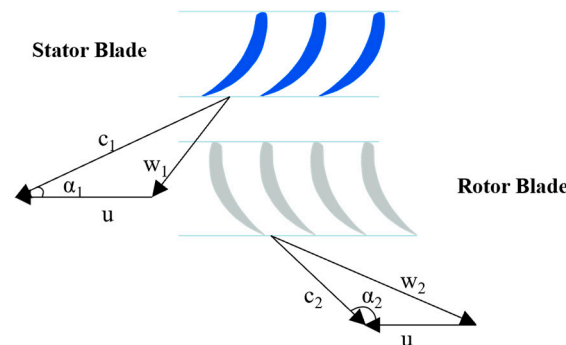


Figure 6. Velocity triangle diagram of turbine stage.

In Equation (13), the Mach number at the stator outlet can be calculated using the outlet absolute velocity (c_1), the outlet static temperature (T_s), the average gas constant (R), and the average specific heat ratio (γ).

$$Ma = \frac{c_1}{\sqrt{RT_s\gamma}} \quad (13)$$

2.4. Rotor Module Modeling

In a gas turbine, the rotor shaft connects the compressor, turbine, and generator, constituting the largest inertial element in the dynamic model of the engine. This component is modeled through a differential equation for rotational angular velocity.

$$\frac{d\omega}{dt} = \frac{P_t - P_c - P_E - P_f}{J\omega} \quad (14)$$

where ω is the angular velocity of the gas turbine; J is the moment of inertia of the rotor; P_t is the output work of the turbine; P_c is the work consumption of the compressor; P_f is the mechanical loss due to friction; P_E is the generator load.

2.5. Volume Module Modeling

In the modeling process, volume modules are primarily placed between the compressor and the combustion chamber, between the combustion chamber and the turbine, and between turbine stages. Their fundamental governing equation is as follows

$$\frac{v_p}{mRT_{out}} \frac{dP_{out}}{dt} = g_{in} - g_{out} \quad (15)$$

where P_{out} is the outlet pressure of the module; g_{in} , g_{out} are the inlet and outlet mass flow rates of the module, respectively; T_{out} is the temperature of the module; R is the gas constant; m is the polytropic index, which in this study is approximated by the adiabatic index k .

2.6. Gas Turbine Simulation Modeling

On the Matlab R2024 simulation platform, the individual modules of the gas turbine are connected to obtain the model shown in Figure 7. This model mainly consists of three

major modules: compressor, combustion chamber, and turbine, and five auxiliary modules: fuel, environment, rotor, load, and outlet.

The model assumes complete combustion of the fuel in the combustor, while the formation of nitrogen oxides (NO_x) and other pollutants is neglected. In addition, the working fluid in the combustor is assumed to be spatially uniform, and detailed chemical reaction processes as well as flow non-uniformities are not considered. Therefore, the present model is primarily applicable to the analysis of thermodynamic performance and flow matching characteristics of gas turbines, rather than emission prediction or high-fidelity flow field analysis [37].

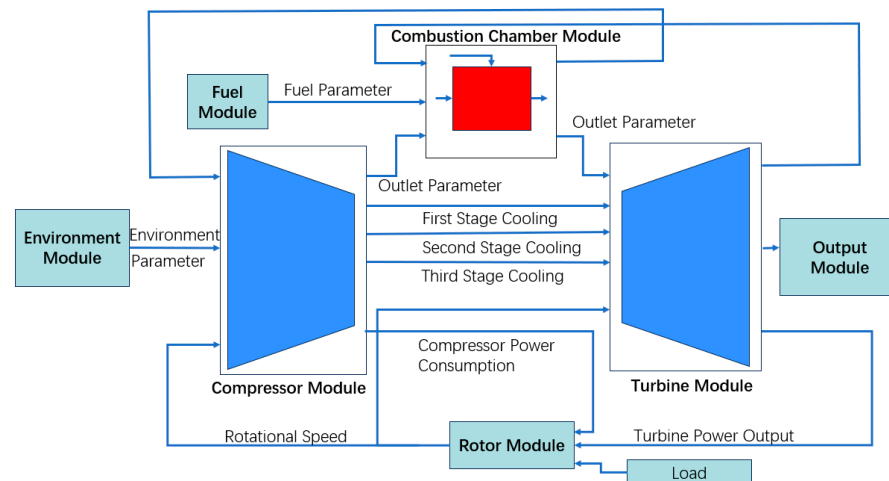


Figure 7. Schematic diagram of the gas turbine simulation model.

The developed thermodynamic model is based on a stage-by-stage calculation method using velocity triangle analysis, and it exhibits a certain degree of generality. The computational procedure can be applied to gas turbines with different power ratings for thermodynamic performance analysis. However, due to differences in geometric dimensions, design parameters, and component characteristics among different gas turbine configurations, the model parameters must be redefined for the specific machine under consideration. Therefore, the numerical results obtained in this study are only valid for the investigated configuration, while the proposed methodology can provide useful reference for hydrogen–ammonia-fueled gas turbine performance analysis and flow matching studies.

2.7. Model Validation

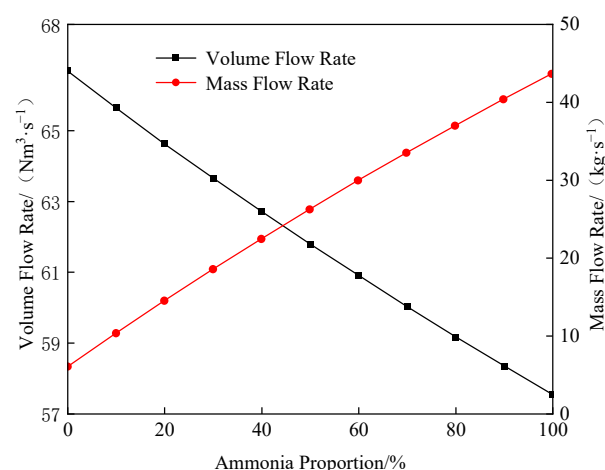
A comparison between the simulation results and the operational data from the F-class gas turbine in Banshan Power Plant is presented in Table 2. The results indicate that the developed model yields a power output of 254.59 MW, an efficiency of 36.33%, and an exhaust gas temperature of 890.25 K. The errors compared to the actual unit are −0.40%, −1.54%, and 0.943%, respectively. This demonstrates a strong agreement between the developed model and the actual unit, confirming that the model provides sufficient accuracy for simulation studies of the gas turbine at the designed point. Due to the lack of actual measurement data for this type of gas turbine under off-designed conditions, the verification of off-designed conditions of the model has not been conducted.

Table 2. Design point parameters of a heavy-duty gas turbine.

Parameter	Design Parameter	Simulated Value	Relative Error
Gas Turbine Power	255.6 MW	254.59 MW	−0.40%
Rated Rotational Speed	3000 rpm	3000 rpm	0
Gas Turbine Efficiency	36.9%	36.33%	−1.54%
Compressor Pressure Ratio	15.4	15.36	−0.26%
Turbine Inlet Temperature	1681 K	1674.3 K	−0.40%
Gas Turbine Exhaust Temperature	882 K	890.25 K	0.94%
Compressor Inlet Mass Flow Rate	638 kg/s	638.8 kg/s	0.13%
Fuel Mass Flow Rate	14.2 kg/s	14.2 kg/s	0
IGV Angle	86°	86°	0

3. Thermodynamic Performance of Hydrogen–Ammonia Mixing Fuel Gas Turbines

After validating the accuracy of the simulation model, the fuel was switched from methane (natural gas) to hydrogen–ammonia co-firing fuels with different ammonia volume fractions (hereinafter referred to as “ammonia fraction”) to obtain the performance parameters of the gas turbine. Given that the volumetric heating value of the hydrogen–ammonia fuel is significantly lower than that of methane, maintaining a constant turbine inlet temperature at the rated speed for different ammonia fractions requires a substantially greater volumetric fuel flow rate compared to the base load condition. However, since ammonia has a higher volumetric heating value and density than hydrogen, the volumetric fuel flow rate decreases as the ammonia fraction increases, while the mass flow rate increases. Under the conditions of constant gas turbine rotational speed and turbine inlet temperature, the required fuel flow rate for each hydrogen–ammonia blending ratio is determined. The fuel mass flow rate is first calculated based on the lower heating values of the fuels, which are taken as 120 MJ/kg for hydrogen and 18.6 MJ/kg for ammonia. The corresponding fuel volumetric flow rate is then calculated using the fuel densities under standard conditions (273.15 K and 101.325 kPa). The variation in fuel demand with the ammonia fraction is shown in Figure 8.

**Figure 8.** Variation in fuel requirements with ammonia fraction.

The increase in fuel volumetric flow rate leads to a significant increase in the volumetric flow rate of the working fluid entering the turbine, which may exceed the maximum flow capacity of the turbine and result in a supercritical flow condition. Since the working fluid within the turbine is under high-temperature and high-pressure conditions, once the flow enters the supercritical regime, the operating performance of the gas turbine deteriorates and potential operational safety risks may arise. The Mach numbers at the

stator outlets of each turbine stage after using hydrogen–ammonia fuel are shown in Figure 9.

As illustrated in the figure above, the Mach number increases with the progression of turbine stages but decreases as the ammonia volume fraction rises. However, even under pure ammonia conditions, the Mach number at the outlet stage of the turbine remains greater than 1. Additionally, when the ammonia volume fraction is less than 0.4, the flow at the outlet of the second-stage stator blades will also exhibit supercritical conditions. To ensure the safe operation of the turbine, through-flow matching must be implemented.

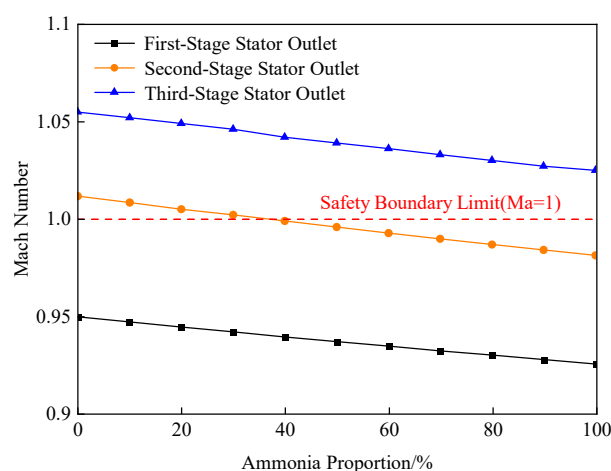


Figure 9. Mach numbers at stator blade outlets of turbine under different ammonia ratios.

3.1. Flow Matching Method by IGV Variation

Common through-flow matching methods include reducing the IGV angle, adjusting the turbine blade installation angle, increasing the compressor pressure ratio, and lowering the turbine inlet temperature. Among these, closing the IGV angle is widely adopted in industry due to its technical maturity and operational simplicity. In this study, a constant turbine inlet temperature (TIT) strategy is employed. Under this strategy, the Mach number within the turbine is reduced by decreasing both the IGV angle and the fuel flow rate. The calculated results for the IGV angle and the corresponding mass flow rates at the compressor and turbine inlets under different ammonia blending ratios are presented in Figure 10.

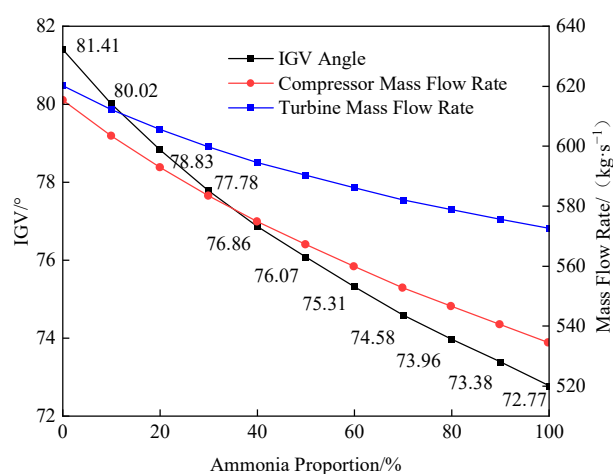


Figure 10. IGV angle and gas turbine flow characteristics under different ammonia percentages.

When fuels with different ammonia blending ratios are used, the fuel demand varies, and the composition of the working fluid after combustion also differs, leading to

corresponding adjustments in the IGV angle. The required IGV adjustment increases with the ammonia blending ratio. This is primarily because a higher ammonia fraction enhances the water vapor content in the working fluid, thereby improving its work capability and significantly increasing the power output of the gas turbine. To comply with power limitations, the IGV must be closed further. Under pure hydrogen and pure ammonia conditions, the IGV angle needs to be reduced by 4.59° and 13.23° , respectively. The closure of the IGV angle further results in a decrease in the mass flow rates at the compressor and turbine inlets. As the ammonia blending ratio increases, the mass flow rates at the compressor and turbine inlets decrease from 615.40 kg/s and 507.20 kg/s to 534.34 kg/s and 474.16 kg/s, respectively.

3.2. Flow Matching Performance

Following the implementation of through-flow matching, the Mach numbers at the stator outlets across all turbine stages are maintained below unity, as illustrated in Figure 11.

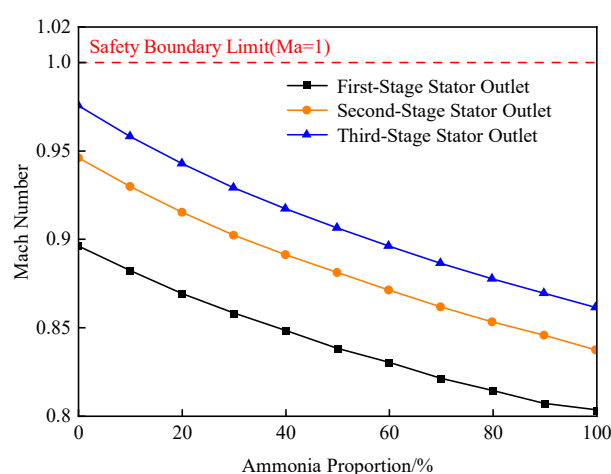


Figure 11. Mach numbers at the outlet of stator blade after flow matching.

As illustrated in Figure 11, the Mach number within the turbine decreases with the increase in the ammonia volume fraction. The maximum Mach number is 0.97 at the outlet of third stage under the pure hydrogen condition, while the minimum is 0.86 under the pure ammonia condition.

Figure 12 shows the relationship between the power and efficiency of the gas turbine and the proportion of ammonia in the gas. P_{Plan 1} is a method of jointly adjusting the IGV Angle and fuel flow rate while keeping the gas turbine speed and turbine inlet temperature unchanged. P_{Plan 2} is a method that only adjusts the fuel flow rate while keeping the gas turbine speed and IGV Angle unchanged. P_{Plan 1} can keep the gas turbine at its rated power under different ammonia ratios.

Overall, when using Plan 1, the efficiency of the gas turbine increased from 37.01% of pure hydrogen to 37.24% of ammonia at 10%, and finally dropped to 36.64% of pure ammonia. When using Plan 2, to ensure that the turbine flow rate does not exceed the critical value, the power of the gas turbine was significantly lower. It could only reach the rated power under pure ammonia conditions, and at this time, the efficiency was 40.13%. Under pure hydrogen conditions, the power of the gas turbine was 202.21 MW, and the corresponding efficiency was 37.65%.

The total pressures at various stages of the turbine after throughflow matching are shown in Figure 13, where IP, SP, and DP represent the total pressures at the turbine stage inlet, stator outlet, and rotor outlet, respectively. The total pressures at different locations remain largely unchanged with the increase in ammonia volume fraction, except at the

turbine inlet. This is because the increase in ammonia fraction leads to a rising trend in the turbine outlet pressure, but simultaneously results in a greater closure value of the IGV, which causes a more significant reduction in the turbine inlet pressure. Under the combined effect of these two factors, the total pressures within the internal stages of the turbine exhibit minimal variation. Specifically, as the ammonia volume fraction increases, the total pressure at the inlet of the first turbine stage decreases from 1315.71 kPa to 1221.92 kPa, decreased by 93.79 kPa. The total pressures at the outlets of the stator and rotor of the first stage decrease from 1086.80 kPa to 1080.55 kPa and from 808.01 kPa to 806.73 kPa, respectively. In contrast, the total pressures at various locations of the second and third turbine stages show an increasing trend with higher ammonia content. The smallest relative increase is observed at the inlet of the second stage, where the total pressure rises from 761.94 kPa to 764.63 kPa, representing a relative increase of 0.35%. The largest relative increase occurs at the outlet of the third stage, where the total pressure increases from 173.72 kPa to 176.35 kPa, corresponding to a relative increase of 1.52%.

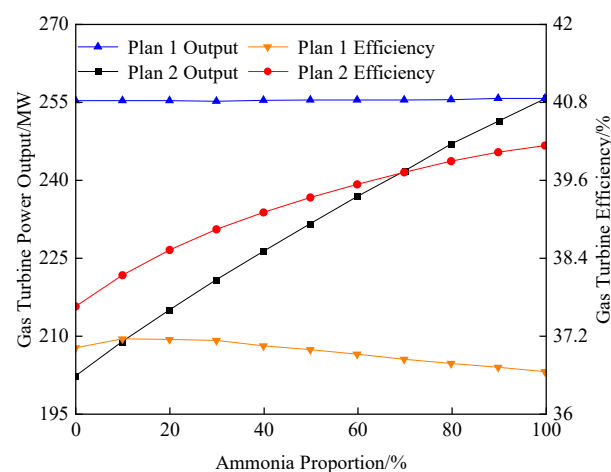


Figure 12. Flow matching performance under IGV modulation strategy and partial load operation strategy.

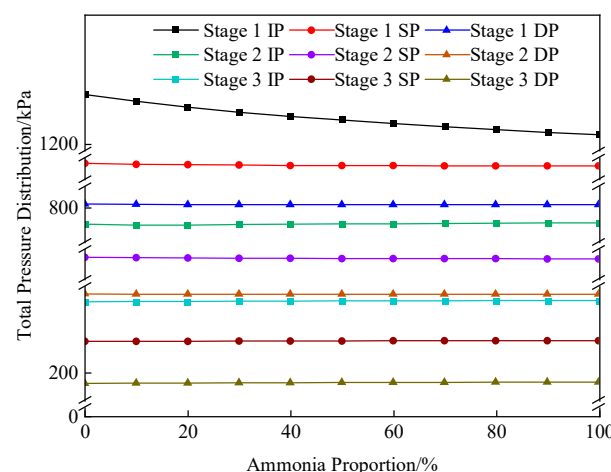


Figure 13. Turbine stage outlet pressures with varying ammonia mixing ratios after flow matching.

Figure 14 illustrates the temperature variations at different locations within the turbine after through-flow matching. Due to differences in fuel composition, the working fluid entering the turbine exhibits varying composition and consequently experiences different cooling effects. As the ammonia volume fraction increases, the volume proportion of water vapor in the working fluid rises, which reduces the cooling effect experienced by the working fluid. Simultaneously, the increase in the average specific enthalpy of the

working fluid enables it to deliver the same amount of work output with a smaller temperature drop. The turbine inlet temperature remains virtually constant with increasing ammonia blending ratio, with a total variation of less than 3 K. The variation in the outlet temperature at each stage increases progressively with the turbine stage number. The turbine inlet and outlet temperatures change from 1589.97 K and 956.60 K under the pure hydrogen condition to 1586.96 K and 983.73 K under the pure ammonia condition, respectively. This results in an overall reduction of the total temperature drop by 30.14 K and an increase in the outlet temperature by 27.13 K.

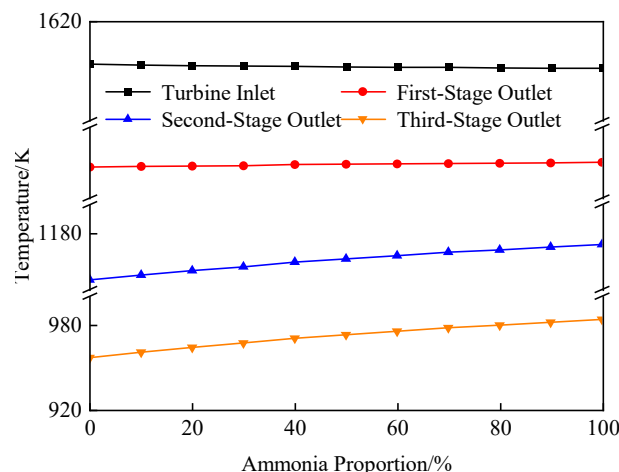


Figure 14. Temperature distribution across the turbine after flow matching.

3.3. Performance Characteristics of a Hydrogen–Ammonia Mixing Fuel Gas Turbine

In the previous section, the flow matching characteristics of a hydrogen–ammonia mixing fuel gas turbine under variable IGV conditions were investigated. Through flow matching regulation, the developed model is able to determine the corresponding operating parameters, including IGV opening and fuel flow rate, for different hydrogen–ammonia blending ratios, thereby providing a basis for subsequent off-design performance analysis. Based on this framework, five fuels with different volumetric compositions—100% H₂, 80% H₂ + 20% NH₃, 50% H₂ + 50% NH₃, 20% H₂ + 80% NH₃, and 100% NH₃—are selected as the research objects in this section, which are denoted as Gas1–Gas5, respectively (the same notation is used hereafter). The operating characteristics of the gas turbine under off-design conditions are then further analyzed.

3.3.1. IGV Variation

The composition of the five fuels (Gas1–Gas5) remained unchanged, and the turbine speed and inlet temperature also remained unchanged. The power, efficiency of the gas turbine, as well as the outlet pressure of the combustor are calculated when the IGV and the fuel flow changed. The IGV variation can be expressed by the following equation:

$$IGV = 1 - \frac{\alpha_{max} - \alpha}{\alpha_{max} - \alpha_{min}} \quad (16)$$

where α_{max} and α_{min} represent the actual maximum and minimum IGV angles during gas turbine operation, with values of 86° and 45.9°, respectively, and α denotes the actual IGV angle.

As shown in Figure 15, under variable IGV opening conditions, the combustor outlet pressure gradually increases with the increase in IGV opening, and this trend is approximately linear. Taking the Gas1 condition as an example, the combustor outlet pressure increases from 1119.64 kPa at an IGV opening of 60% to 1247.21 kPa at an IGV opening of 80%. Under the same IGV opening, there are certain differences in the combustor outlet

pressure corresponding to different fuel compositions. Moreover, as the volumetric fraction of ammonia in the fuel increases, the combustor outlet pressure shows a slight overall increase.

Figure 16 illustrates the variation in gas turbine power. As shown in Figure 16, under different IGV openings, the gas turbine output power exhibits a monotonically increasing trend with the increase in IGV opening. Although the variation trends of gas turbine power with IGV opening are generally similar for different fuel compositions, the power curves show inflection points at different IGV openings. For the Gas1–Gas5 conditions, as the IGV opening increases from 60% to 80%, the gas turbine power increases from 199.22 MW, 203.43 MW, 210.77 MW, 221.32 MW, and 229.22 MW to 238.51 MW, 248.30 MW, 255.39 MW, 267.02 MW, and 271.51 MW, respectively, indicating a relatively wide variation range in power output.

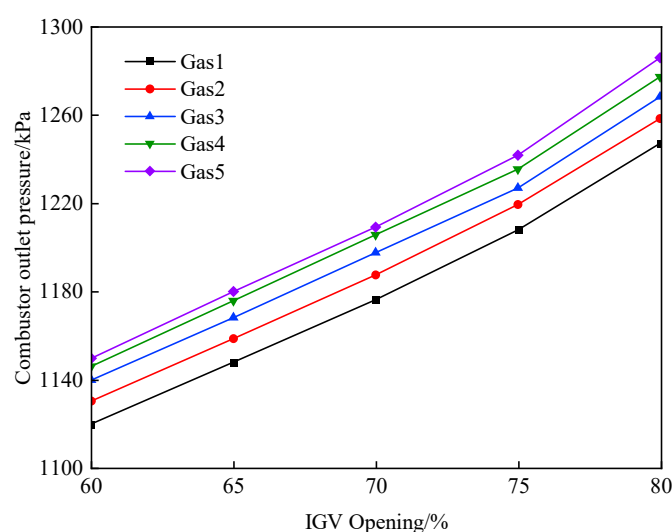


Figure 15. Combustor outlet pressure at different IGV variations.

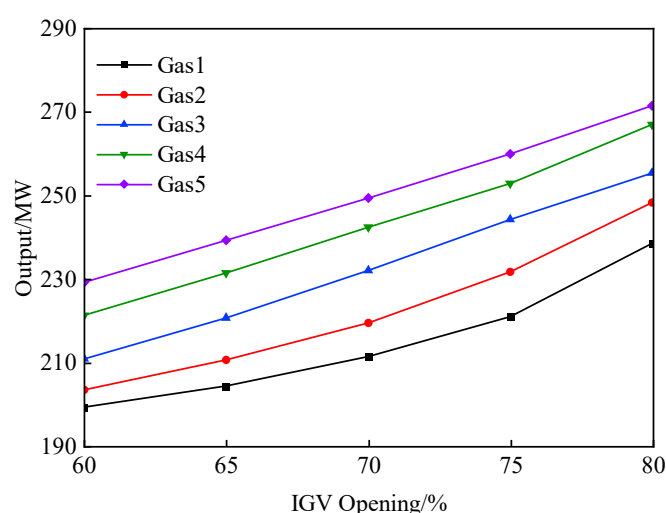


Figure 16. Gas turbine power at different IGV variations.

As shown in Figure 17, under variable IGV opening conditions, the gas turbine efficiency generally exhibits a continuously increasing trend with the increase in IGV opening. This is because, as the IGV opening gradually increases, the operating condition of the unit approaches the design point, thereby improving the efficiency level of the gas turbine. For gas1, the efficiency is 34.65% when the IGV is 60% and 36.50% when the IGV is 80%.

Under the same IGV opening, certain differences in gas turbine efficiency can be observed for different fuel compositions. Moreover, as the volumetric fraction of ammonia in the fuel increases, the gas turbine efficiency shows a gradual decreasing trend. Taking the case of an IGV opening of 80% as an example, the gas turbine efficiencies under the Gas1–Gas5 conditions are approximately 36.50%, 36.23%, 35.73%, 35.62%, and 34.95%, respectively.

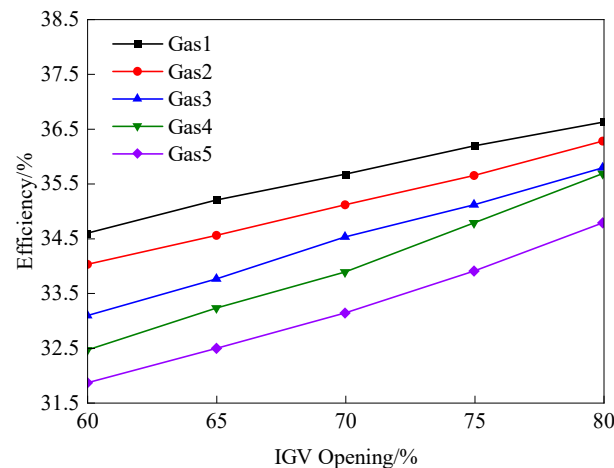


Figure 17. Gas turbine efficiency at different IGV variations.

3.3.2. Fuel Flow Rate

For the variable fuel flow rate condition, the fuel flow rate obtained from the flow matching calculation is taken as the rated value, and the ratio of the actual fuel flow rate under each operating condition to this rated value is defined as the relative fuel flow rate. By varying the relative fuel flow rate while maintaining the gas turbine rotational speed at the rated value of 3000 r/min, the operating characteristics of the gas turbine under different fuel supply conditions are analyzed.

Figure 18 illustrates the variation in combustor outlet pressure for five hydrogen–ammonia blended fuels (Gas1–Gas5) under variable fuel flow rate conditions. As shown in Figure 18, within the investigated range of relative fuel flow rate, the combustor outlet pressure generally increases with increasing fuel flow rate, and this trend is approximately linear.

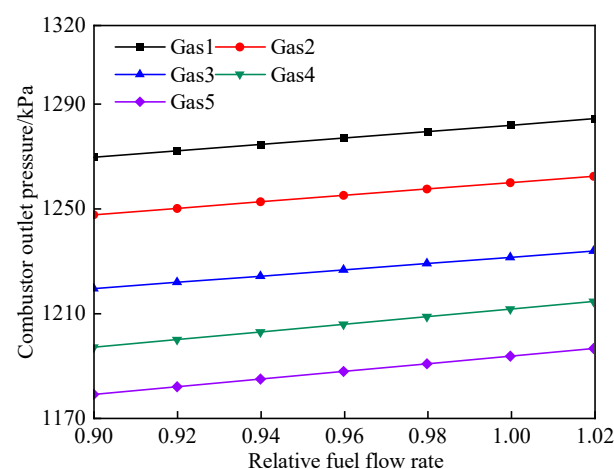


Figure 18. Combustor outlet pressure under variable fuel flow rate conditions.

From the comparison of different fuel compositions, it can be observed that, under the same relative fuel flow rate, fuels with a lower volumetric fraction of ammonia

correspond to higher combustor outlet pressures. As the ammonia fraction increases, the combustor outlet pressure gradually decreases, although the decreasing rate becomes less pronounced. Taking the case with a relative fuel flow rate of 1.02 as an example, the combustor outlet pressure is approximately 1284.34 kPa by Gas1, while the corresponding pressure is 1196.38 kPa by Gas5.

Under variable fuel flow rate conditions, the combustor outlet temperature is further calculated in this section, and the corresponding results are presented in Table 3. With the variation in relative fuel flow rate, the combustor outlet temperature increases with increasing fuel flow rate. When the relative fuel flow rate increases from 0.9 to 1.02, the combustor outlet temperature under the Gas1–Gas5 increases by 111.38 K, 110.38 K, 107.69 K, 104.56 K, and 103.12 K, respectively.

Table 3. Combustor outlet temperature at different fuel flow rates.

Relative Fuel Flow Rate	Gas1 (K)	Gas2 (K)	Gas3 (K)	Gas4 (K)	Gas5 (K)
0.90	1588.29	1587.99	1588.42	1589.46	1588.66
0.92	1606.67	1606.37	1606.94	1607.43	1606.63
0.94	1625.05	1624.75	1625.46	1625.51	1624.71
0.96	1643.64	1643.34	1643.52	1643.70	1642.90
0.98	1662.47	1662.17	1662.04	1662.35	1661.55
1.0	1681.00	1681.00	1681.00	1681.00	1681.00
1.02	1699.67	1698.37	1696.11	1694.02	1691.78

Under the same relative fuel flow rate, the variation in combustor outlet temperature corresponding to different hydrogen–ammonia blending ratios is relatively small. Taking the case with a relative fuel flow rate of 1.02 as an example, only minor differences in combustor outlet temperature can be observed for different fuel compositions. Specifically, the combustor outlet temperature is 1699.67 K by Gas1 and 1691.78 K by Gas5, indicating that the outlet temperature decreases with the increase in the volumetric fraction of ammonia in the fuel.

As shown in Figure 19, with the increase in relative fuel flow rate, the overall efficiency of the gas turbine under different fuel compositions exhibits a similar variation trend, characterized by an initial increase followed by a decrease, reaching the maximum value when the relative fuel flow rate is 1.00. In addition, the variation in overall gas turbine efficiency differs with the ammonia fraction in the fuel. Under the same relative fuel flow rate, the gas turbine efficiency generally decreases with the increase in the volumetric fraction of ammonia in the fuel, decreasing from 37.22% under the pure hydrogen to 36.69% under the pure ammonia 100%.

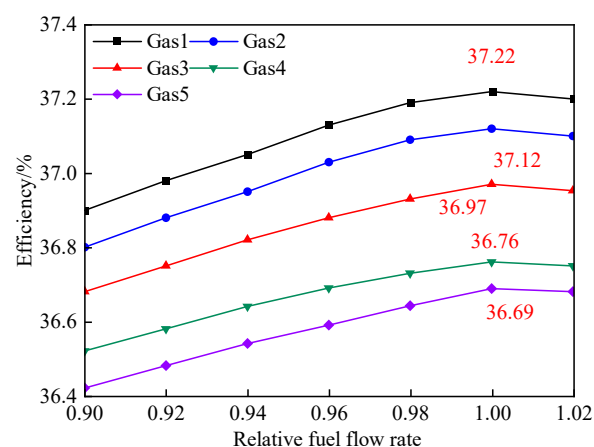


Figure 19. Gas turbine efficiency at different fuel flow rates.

4. Conclusions

This study established a simulation model for the investigated heavy-duty gas turbine, incorporating inter-stage compressor bleed air and stage-by-stage turbine cooling. To address the supercritical flow in turbine blades when switching from methane to hydrogen–ammonia mixing fuel, a through-flow matching method by the inlet guide vanes (IGV) was proposed. The conclusions are as follows:

1. The simulation model was verified using actual power plant operation data. The calculated power of the model was 254.59 MW, the efficiency was 36.33%, the exhaust temperature was 890.25 K, and the errors were -0.4% , -1.54% , and 0.94% respectively. The errors of other parameters were all below 0.5% . The results indicate that the model has sufficient accuracy.
2. As the volume proportion of ammonia in the hydrogen–ammonia mixed fuel increases, the adjustment amount of the IGV also increases. From 4.59° in the pure hydrogen condition to 13.23° in the pure ammonia condition, this can prevent the outlet flow velocity of the turbine from exceeding the critical value, and can ensure the safe operation of the gas turbine.
3. When the inlet temperature of the turbine remains constant, the Mach number of the outlet flow velocity of the turbine stage decreases as the volume proportion of ammonia in the mixed fuel increases. The maximum Mach number of the stator blade at the turbine outlet is 0.97 under the pure hydrogen condition, and the minimum value is 0.86 under the pure ammonia condition.
4. As the ammonia volumetric fraction increases, the gas turbine output power can be maintained at the rated value of 255.6 MW, the turbine inlet and outlet temperatures change from 1589.97 K and 956.60 K under the pure hydrogen condition to 1586.96 K and 983.73 K under the pure ammonia condition, respectively. This results in an increase in the outlet temperature of 27.13 K. The maximum efficiency of 37.22% is achieved at 100% hydrogen fuel (Gas1), whereas the minimum efficiency of 36.64% is obtained at 100% ammonia.
5. When the opening degree of the IGV angle increases from 60% to 80%, the outlet pressure of the combustion chamber, the power of the gas turbine, and the efficiency all show an upward trend. Taking 100% hydrogen fuel (Gas1) as an example, the outlet pressure of the combustion chamber increases from 1119.64 kPa to 1247.21 kPa; the efficiency is 34.65% when the IGV is 60% and 36.50% when the IGV is 80%.
6. As the fuel flow increased from 90% of the rated value to 102%, the outlet pressure and temperature of the combustion chamber rose accordingly. Specifically, when the fuel flow was 102%, the outlet pressure of the combustion chamber for 100% hydrogen fuel (Gas1) and 100% ammonia fuel (Gas5) was 1284.34 kPa and 1196.38 kPa respectively, and the outlet temperature of the combustion chamber increased by 111.38 K and 103.12 K respectively.

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