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Research article

A model for booster station matching of gas turbine and gas compressor power under different ambient conditions

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ABSTRACT

Transporting natural gas across different locations require compressor stations to provide the pressure needed to keep the gas moving. This paper presents a model for matching the gas turbine and gas compressor power required under different environmental conditions to support the continuous gas transmission across other locations. The trans-Saharan gas pipeline (TSGP) project proposed to transport gas from Nigeria to Algeria has been used as a case study in this paper. The TSGP project is a Nigerian Government initiative to rejig its gas development and transportation infrastructure to meet its internal and external market demand. The numerical method used in this paper integrates the effect of the ambient temperature in the power matching of the gas turbine and gas compressors. There are 18 compressor stations across the TSGP network, and compressor station 2 is used as the reference point. The daily temperature fluctuation is segmented into hours of the day, emphasising considerable ambient temperature variation at 3:00 h, 9:00 h, 15:00 h. One benefit of the model against others in the open literature is accounting for changes in the ambient temperature along the pipeline network and gas compression stations. Accounting for changes in ambient temperature provides accuracy to near real-life operational experience for gas distribution via pipelines. The model also accounts for variations in turbine entry temperature (TET) to compensate for changes in the ambient conditions to meet the power requirements of the gas turbine and the gas compressor. The results show that for every 1% increase in ambient temperature, a 3.5% increase in power is required to drive the gas compressor and a 1% decrease in gas turbine output power. The effect of the 1% increase in ambient would require a 3.5% increase in TET to meet both the gas turbine and gas compressor requirement.

1. Introduction

Natural gas is a cleaner and environmentally friendly energy source than crude oil, making it an attractive energy option [1]. However, vast quantities of natural gas reserves are found in remote locations requiring transport to the consuming market [2] through gas transmission networks across distances in different locations. A typical gas transmission network would consist of large numbers of pipes, valves, and regulators, with injection, delivery points and compressor stations (consist of gas compressor and the driver) [2]. The compressor stations are placed at suitable intervals to ensure a steady flow of the natural gas through the pipeline network and carefully monitored to provide high operational reliability and proper power matching between the gas compressor and the driver. According to Lubomisky et al. [3], every gas pipeline transportation system's success depends mainly on how compressor stations are constructed, the type of compressor employed, selection of driver, and proper matching of both the driver and the driven at various operating conditions. An important factor favouring gas turbine-driven compression stations is gas availability, which is the energy source for gas turbines.

Centrifugal compressors, driven by multi-shaft gas turbines, are widely utilized in gas transmission systems to meet the optimum alternate process requirement. Centrifugal compressors are used in most gas transmission networks because of their compactness, reliability, and relatively low-pressure ratios compared with axial compressors [4, 5]. However, it is crucial to verify the proper integration and matching of the centrifugal compressor and its driver to ensure the operational needs at design and off-design conditions are satisfied. Most of the centrifugal compressors used in the oil and gas industry operate within a specified operating envelope, consisting of many operating points, speeds, and

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power levels. These operating points change due to changes in the process conditions, such as gas composition, pressure, temperature, and throughput.

The selection of the drivers (e.g. gas turbine) for the gas compressors located across the gas transmission network is influenced by the power demand, capital, maintenance and operating cost, reliability, and availability. The gas turbine is expected to provide enough power to the centrifugal compressor operating points throughout its operations. Understanding the technical requirement for matching the gas compressors and drivers is essential when selecting both pieces of equipment for smooth transportation of the natural gas from one location to another. Proper matching of the gas turbine and the centrifugal compressor improves the overall system efficiency, reduces gas turbine fuel consumptions, and enhances system availability and reliability. Generally, simulating the gas transmission network could pose an arduous task because of the environment's impact on the pipeline and compressor stations [6]; hence, the model developed in this article assumes non-isothermal steady-state conditions. Lopez-Benito et al. [7] used a similar comprehensive modelling and simulation approach to differentiate the pipeline's pressure profile as a non-isothermal process that allows the model to account for ambient conditions changes.

To this end, there has been growing research on pipeline systems and driver's performance analysis for the natural gas transmission network. Kurz and Mokhatab [7] developed a simulation tool to study the effect of pipeline systems and compressor configuration in response to operational changes [3, 8]. The authors used the different compressor arrangement (sequential and parallel) to assess the power requirements and emissions and transient behaviour of the system to low demand in the gas flow [9]. Lubomirsky et al. s. [3]. work on the impact of distance between compressor stations showed that there was less flow resistance when more compressors are installed along the pipeline network.

It appears that most previous works were focused on optimising the gas compressor configuration performance for natural gas processing. Albusaidi and Pilidis [10] developed an iterative method to understand the centrifugal compressor's behaviour at various operating conditions to understand the optimal performance requirement for gas transportation. Some research looked at control optimization for multiple gas turbine-driven compressors for various stations. For example, Uster and Dilaveroglu [11] developed a large integrated scale mixed-integer non-linear optimization model to ascertain pipelines in the network and compressor stations capacities. Zamotorin et al. [12] looked at simple models to control the gas turbines' varying station load requirements to improve efficiency and operating range. Their method considers loading the gas turbines evenly by running all compressors at the same distance from their respective surge lines while the gas turbines maintained the same load setting. Rashidzadeh et al. [13] focused on the gas turbine for mechanical drives when the gas turbine operates at part-load. Their goal was to understand the behaviour of the gas turbine components at different operational settings. Taher and Meher-Homji [14] attempted to show the matching requirement for the compressor and gas turbine. However, their work only provided the rating philosophy and provided guidelines for effective evaluation.

Another important factor in stimulating the natural gas transmission process is the pipeline and hydraulics models, as the governing equations should account for the real gas effect on the system. Chaczykowski [15] highlighted the effects of selecting the correct equation of state for the pipeline gas flow model. This equation should account for the real gas effect on the hydraulics of natural gas transmission. In their work, the authors utilised a non-isothermal gas flow model to understand its influence on flow parameters, especially on the gas temperature and pipeline line-pack. Similarly, Adeosun et al. [16] developed an unsteady-state Weymouth equation for gas volumetric flow rate in horizontal and inclined pipes. The unsteady-state equation gives a functional relationship between flow rate, inlet and outlet pressure at any given time and could account for initial transience in gas volumetric flow rate [16]. Osiadacz and Chaczykowski [17] compared isothermal and non-isothermal pipeline gas flow models, accounting for significant variation in the pressure profile along the pipeline.

This paper presents the technical requirement for matching the gas compressor (driven) and gas turbine power (driver) requirement based on the ambient temperature of each booster station along the trans-Saharan gas pipeline at a different time interval of the day. This is important because, as the gas is transported via a pipeline network from one location to another, the compressor power requirements and the gas turbine plant are affected by the gas pressure demand and the environment's ambient temperature. Proper matching of both pieces of equipment would improve the overall system efficiency, cost savings, improve overhaul time intervals and many more. One of the assumptions considered in this study is that the pipeline pressure will remain constant; hence any fluctuation is compensated by the centrifugal compressor to maintain a steady flow. Although the pipeline network has eighteen compressor stations, a reference station is used to demonstrate this study's goal.

The paper is structured in five (5) sections. The first section introduces the trans-Saharan gas pipeline project, where the matching model is to be applied. The second section explains the power matching model's underpinning concept for the centrifugal compressor and aero-derivative gas turbine in this paper. The next section discusses the results presented in this paper, and finally, the overarching points from this work and recommendations are presented.

2. Model application and gas transportation

Gas production in Nigeria has been increasing with current reserves of over 190 trillion cubic feet, making it the seventh-largest in the world [18, 19]. The trans-Saharan gas pipeline (TSGP) project was conceived by the Nigerian Government, aimed at exporting natural gas from the Niger Delta region in Nigeria to the consumer market in Europe through Niger and Algeria [20]. The project is expected to transport 30 billion cubic meters per year of natural gas through a 56-inch pipe diameter and total pipeline distance of 4180 km with 18 compression stations at suitable intervals along the pipeline network. In Nigeria, TSGP is regarded as a strategic project because it will generate revenues and open opportunities for the Nigerian gas sector [20]. A feasibility report on this project was conducted in 2006 [20]. It is worth noting that all the reports only mention the number of compressor stations without providing the exact location to be constructed. Table 1 shows the TSGP transmission pipeline data that goes from Nigeria to Algeria [21].

Transporting the gas to the European and Mediterranean market where there is a considerable demand for its use is the TSGP project's goal. However, realising the project goal is with the help of transnational collaboration as the pipeline network would connect through Algeria to the European market, which already consumes the Nigerian liquefied natural gas. This transport network would also benefit domestic supply across Nigeria and throughout its sub-region, such as Niger. Aside from the TSGP project, other ongoing regional initiatives are to transmit gas via a pipeline network to the Benin Republic, Togo and Ghana [18].

One would think there could be conflicting interest in transporting the gas via Algeria, considering that Algeria has gas production in abundance [22] and a major gas supplier in the European market. Although there has been a decline in Algeria gas production in the last 5 years [23], its current gas reserves are over 2.9 trillion cubic feet, making her the tenth-largest in the world. There is speculation that Algeria's varying market interest and transnational economic policies could harm the TSGP project. Despite how genuine the argument on varying interest could be, it is an area the authors of this article will not want to be drawn into. The business interest for the TSGP project for Algeria, Nigeria and Niger is set at 45%, 45% and 10%, respectively, with a project cost estimate of over $13 billion [21].

Table 1. Pipeline data [20].

Parameters	Values
Gas flow	30 bcm/year
Nominal Diameter	56 inches
Outer Diameter	56 inches (1422.4mm)
Inner Diameter	54.38 inches (1381.2mm)
Wall Thickness	0.812 inches (20.62mm)
Design Pressure	100 bar
MAOP	95 bar
Pipe Grade	API 5L X70
Design Code	0.72
Design Factor	ASME B31.8
Roughness	0.0003 inches (7.62 μm)

3. Theory of model

The daily ambient temperatures for the different gas compressor stations along the Pipeline were obtained and divided into winter, dry, and summer seasons to match the centrifugal compressor power requirement and gas turbine power. Four gas turbines were simulated based on the ambient temperature and the power requirement of the centrifugal gas compressor. The simulation was done with a Cranfield University in-house simulation code to ensure that the gas turbine will provide the centrifugal compressor's power at all the different power operating levels. The modelling approach can be divided as follows:

- ➢ Define the ambient temperature based on the time interval at each of the compressor station locations.
- ➢ Calculate the gas compressor power requirement at each station based on the ambient temperature and time interval.
- ➢ Simulate the gas turbine power based on each station location's ambient temperature and time interval.
- ➢ Find the gas turbine entry temperature (TET) to drive the gas compressor based on the compressor power requirement.

3.1. The centrifugal compressor

The centrifugal compressor is designed to provide the required head at the specified flow rate with the optimum efficiency at its design points. However, most of the time, the centrifugal changes its normal operating point due to variations in its operating environment. Sometimes, these new operating points may require more polytropic head or higher capacities. The ideal gas law is the fundamental thermodynamic law of understanding the operation of the compressor. However, no gas in operation conforms to this law due to varying operating conditions, hence using the compressibility factor (Z) in the real gas equation for accounting for such deviations. In this work, the compressibility factor is obtained from the generalized Nelson-Obert chart [24]. The compressibility factor is plotted as a function of reduced temperature T_r and reduced pressure P_r. The reduced temperature is obtained by dividing the actual temperature of the gas by critical temperature:

$$T_r = \frac{T}{T_c}\ ;\ P_r = \frac{P}{P_c} \tag{1}$$

HYSYS is used for the calculation of gas compressor polytropic efficiency. HYSYS software presently deals with enhanced Peng-Robinson (PR) and Soave-Redlich- Kwong (SRK) equations of state to estimate the gas's thermodynamic properties in operation. PR equation of state supports the broadest array of operating conditions. Over the years, PR and SKR are some of the equations developed to calculate the constants in the van Der Waals equation [25]. The PR and SRK are the most common equations of state used in the oil and gas industry. The Peng-Robinson equation is expressed as follows.

$$a = \frac{0.457247R^2T_c^2}{P_c} \tag{2}$$

$$b = \frac{0.07780RT_c}{P_C} \tag{3}$$

The SRK equation is given as

$$a(T) = 0.4274\left(\frac{R^2T_c^2}{P_C}\right)\left\{1 + m\left[1 - \left(\frac{T}{T_C}\right)^{0.5}\right]\right\}^2 \tag{4}$$

where

$$m = 0.480 + 1.57\omega - 0.17\omega^2 \tag{5}$$

$$b = 0.08664\ \frac{RT_c}{P_C} \tag{6}$$

A. **Gas Compressor Power Estimation**

The n-value approach was adopted in this paper for the estimation of the centrifugal compressor power requirement. The compressor polytropic efficiency calculation is based on the fact that polytropic efficiency is independent of the thermodynamic state of gas undergoing compression, as expressed in the equation below [26].

$$\eta_{poly} = \frac{\frac{n}{n-1}}{\frac{\gamma}{\gamma-1}} \tag{7}$$

The gas compressor discharge temperature is one of the factors that limit the stage compression ratio. Therefore, it should be kept within a reasonable value. The discharge temperature is a function of inlet temperature, pressure ratio, and process exponent

$$T_2 = T_1 r_{p^{(n-1)/n}} \tag{8}$$

The polytropic head is the compressibility factor's function, process exponent, molecular weight, inlet temperature, and pressure ratio is expressed as

$$H_p = \frac{Z_a}{1,000}\ \frac{8,314}{MW} T_1 \frac{n}{n-1}\left[r_p^{\frac{n-1}{n}} - 1\right] \tag{9}$$

The power required to drive the gas compressor is calculated using the equation below.

$$Power_g = \frac{mH_p}{3,600\ \eta_p} \tag{10}$$

$$Ps = Power_g + Lm \tag{11}$$

where *Lm* are mechanical losses associated with bearings.

Based on the centrifugal compressor's power requirement at compression station 2, Dresser-Rand-Datum C centrifugal compressor was selected for this assessment. Table 2 summarizes the characteristics of the centrifugal compressor.

B. **Centrifugal Compressor Map**

The compressor performance curves comprise a graph showing the variation of the polytropic head, efficiency, pressure ratio, and power at several constant speed and at a different flow rate, as shown in Figure 1 and Figure 2. A new compressor performance curve for compressor station 2 was obtained from the scaling of an existing original equipment manufacturer (OEM) curve. There is a need for correction of both the mass flow rate and rotational speed [10, 27]. Albusaidi [28] suggested that the corrected speed is obtained by maintaining a fixed tip Mach number and considering the variation in gasses properties.

$$N_c = \frac{N_{act}}{a_{act}/a_{ref}} \tag{11a}$$

The equation above is used for the same engine where the diameter impeller is constants. To study the effect of the gas properties, then the equation above becomes:

$$N_c = N_{act}\sqrt{\frac{k_{ref}}{k_{act}}}\sqrt{\frac{Z_{ref}}{Z_{act}}}\sqrt{\frac{R_{ref}}{R_{act}}}\sqrt{\frac{T_{ref}}{T_{act}}} \tag{12}$$

Berdanier et al. [29] suggested that the effect of the humidity factor should be ignored because it has an insignificant impact on this correction application. The following equation for corrected mass flow is applied since there is no change in diameter.

$$\dot{m_c} = \dot{m}_{act}\frac{P_{ref}}{P_{act}}\sqrt{\frac{k_{ref}}{k_{act}}}\sqrt{\frac{Z_{act}}{Z_{ref}}}\sqrt{\frac{R_{act}}{R_{ref}}}\sqrt{\frac{T_{act}}{T_{ref}}} \tag{13}$$

New compressor maps were obtained by scaling the corrected values using the factors defined below [10, 27].

$$S_f\ PR = (PR_{DP} - 1) \big/ (PR_{DP\ Map} - 1) \tag{14}$$

$$S_f\ NDW = \left(\frac{W\sqrt{\theta}}{\delta}\right)_{DP} \Bigg/ \left(\frac{W\sqrt{\theta}}{\delta}\right)_{DP\ Map} \tag{15}$$

$$S_f\ EFF = \eta_{DP}/\eta_{DP\ Map} \tag{16}$$

3.2. Gas turbine performance characteristics

Gas turbines are versatile machines that have been used in aerospace and industrial applications for many years. Kurz and Ohanian [30] opined that in gas pipeline transportation, the ideal means of compression is the combination of centrifugal compressor and industrial gas turbine. The gas turbine's design and off-design performance simulation were done using a Cranfield University in-house performance simulation tool [31]. Gas turbine power varies with the environmental conditions, output speed, inlet and exhaust losses, and degradation with the effect of variation of hot ambient temperature on the gas turbine power and heat rate at full load discussed in the next section. An aero-derivative gas turbine is used for the compression station 2 to drive the centrifugal compressor's rated power requirement. Table 3 summarizes the performance characteristics of the gas turbine.

3.3. Modelling the effect of ambient temperature

The daily three hourly temperature measurements for winter, hot, and dry seasons were recorded for station 2, as shown in Figures 3, 4, 5

Table 2. Compressor characteristics at station 2.

Compressor station 2 Configuration	
Actual Volume flow (m^3/h)	10510
Inlet temperature (°C)	56.11
Discharge Temperature (°C)	22.22
Inlet pressure (bar)	65.39
Discharge Pressure (bar)	95
Polytropic head (kJ/kg)	47.32
Power (kW)	10904
Efficiency (%)	0.761

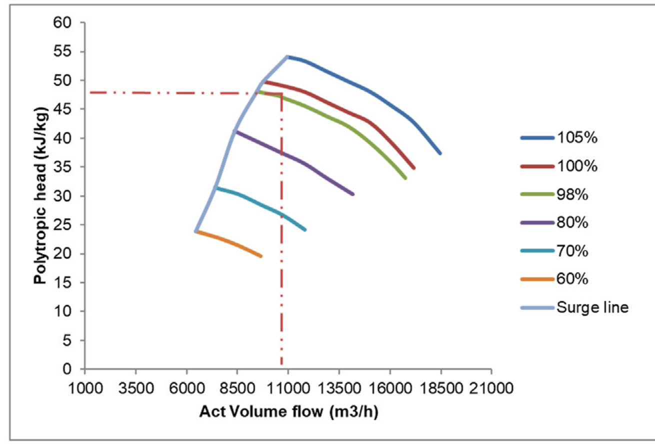

Figure 1. Centrifugal compressor characteristics showing the polytropic head against the actual volumetric flow.

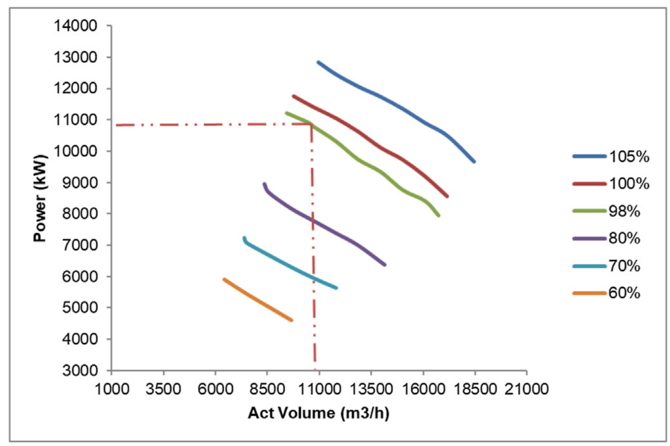

Figure 2. Centrifugal compressor characteristics showing the power demand against the actual volumetric flow.

below. The power requirement for both the centrifugal compressors and gas turbines were simulated and calculated based on these ambient temperatures.

For the centrifugal compressor, the T_1 from the polytropic head (H_p) equation represents the ambient inlet temperature. In this paper, the variations of T_1 depended on the time of the day and the season of the year. However, the discharge temperature of a long-distance gas pipeline is usually in equilibrium with the ambient temperature of the surrounding environment, as shown in Figure 6 below:

The gas discharge temperature is obtained using the heat transfer relationship between the ambient and gas temperature as shown with Eqs. (17), (18), (19), (20), (21), (22), (23), (24), (25)

$$Q = UA\Delta T \tag{17}$$

Where Q is heat flow (W), A is the area (m^2), ΔT is the temperature difference (K), and U is the overall heat transfer coefficient of insulated pipe (W/m^2K) is determined using the following equation.

$$U = \frac{1}{\frac{D_3}{D_1 + h_{in}} + \frac{D_3 * In\left(\frac{D_2}{D_1}\right)}{2 * k_{pipe}} + \frac{D_3\ * In\left(\frac{D_3}{D_2}\right)}{2 * k_{insulation}} + \frac{1}{h_{out}}} \tag{18}$$

where k_{pipe} and $k_{insulation}$ are the thermal conductivities of the pipe and insulation while $h_{in}$is the heat transfer coefficient in the pipe and h_{out} is the heat transfer coefficient at the outside insulation surface. The temperature difference ΔT is also expressed as follows:

$$\Delta T = \frac{T_1 - T_2}{In\frac{T_1 - T_a}{T_2 - T_a}} \tag{19}$$

where T_1 is the inlet temperature, T_2 is the outlet temperature, and T_a is the ambient temperature. Therefore, the heat flow (Q) is determined using the equation below:

$$Q = U\pi dL\frac{T_1 - T_2}{In\frac{T_1 - T_a}{T_2 - T_a}} \tag{20}$$

There is a need to equate the gas and heat flow equation to obtain the gas discharge temperature, as shown in the equation below:

$$mC_p(T_1 - T_2) = U\pi dL\frac{T_1 - T_2}{In\frac{T_1 - T_a}{T_2 - T_a}} \tag{21}$$

$$T_2 = T_a + (T_1 - T_a)exp^{\frac{-U\pi d}{mC_p}L} \tag{22}$$

$$\theta = \frac{-U\pi d}{mC_p}L \tag{23}$$

$$e^{\theta} = 0 \tag{24}$$

$$T_2 = T_a \tag{25}$$

where T_2 is gas temperature and is T_a ambient temperature.

In a long-distance natural gas pipeline transportation system, pipelines are constructed in segments [32]. The exit of one pipeline is the inlet of the centrifugal compressor. Therefore, as ambient temperature changes, the value of polytropic head changes and the gas compressor power requirement.

For the gas turbine, changes in the ambient temperatures are accounted for during the simulation by varying the gas turbine inlet temperature.

4. Discussions

Figures 1 and 2 describe the new hypothetical centrifugal compressor map's design point characteristics at station 2. The map shows the compressor operates at 98% rotational speed, which could be referred to as its design point. As previously mentioned, the map produced from the scaled existing OEM map represents the centrifugal compressor's behaviour at the various operating envelope.

4.1. Effect of ambient temperature on compressor performance

The centrifugal compressor's pressure ratio varies with the change in ambient temperature [6]. Hence, the gas compressor must react to the new changes while maintaining the Pipeline's maximum allowable operating pressure. This is possible because the compressor rotational speed changes directly with the available flow rate. In this work, both the gas composition and volumetric flow is assumed to be constant; hence any change in ambient temperature or the gas temperature affects the volumetric gas flow rate since the compressor suction flow rate is a function of ambient temperature and inlet pressure. The approach is similar to the works of Li and Lapina [5, 26]. Figures 7 and 8 shows the changes in the centrifugal compressor map characteristics as the ambient condition changes. These changes can be explained using Eqs. (26) and (27).

Table 3. Gas turbine specification.

Parameters	Values
Power output (MW)	15.2
Pressure ratio	22
Thermal efficiency (%)	38
Inlet mass flow (kg/s)	46

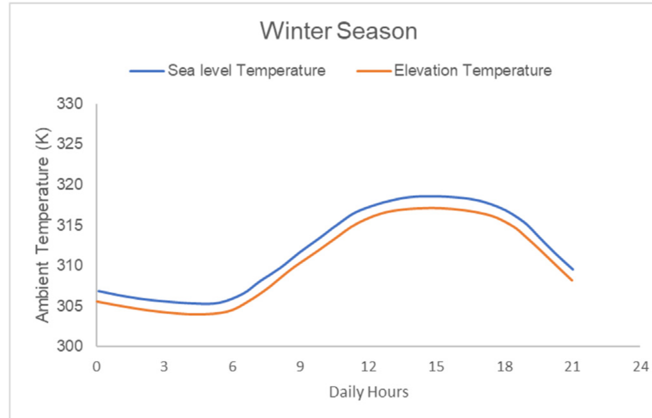

Figure 3. Ambient temperature curve at compressor station 2 during the winter season.

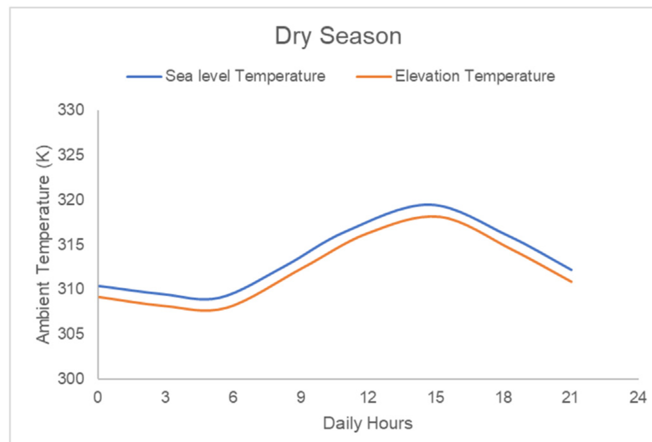

Figure 4. Ambient temperature curve at compressor station 2 during the dry season.

$$Q_2 = Q\left(\frac{P_b}{P_2}\right)\left(\frac{T_{amb}}{T_b}\right)\left(\frac{Z_2}{1.0}\right) \tag{26}$$

The actual mass flows rate is defined as the actual volumetric flow rate multiply by compressor gas inlet density as express below:

$$W_2 = Q_2{}^* \rho_2 \tag{27}$$

An increase in ISO's ambient condition increases the centrifugal compressor's polytropic head, increasing the gas centrifugal compressor power demand. In contrast, the same effect on the gas turbine results in decreased gas turbine output power. An increase in fuel burn would be needed to maintain a constant actual mass flow rate and compensate for the temperature fluctuations.

Figure 9 shows that the gas compressor polytropic head and gas power trend will increase as suction temperature increases for a constant volume flow rate and gas composition. The gas compressor head rises proportionally with the increase in inlet temperature and vice versa [33]. Since the compressor power is a function of the mass flow rate, polytropic head and efficiency, an increase on any of these parameters will impact the compressor power. This ambient temperature factor is of great importance for mechanical drive applications. The ambient temperature strongly influences the centrifugal compressor polytropic head and the

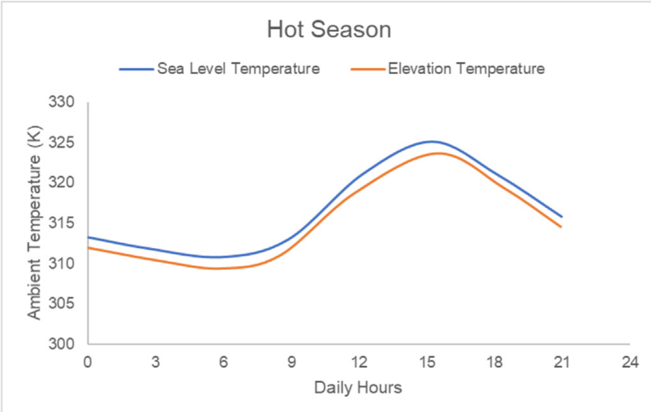

Figure 5. Ambient temperature curve at compressor station 2 during the hot season.

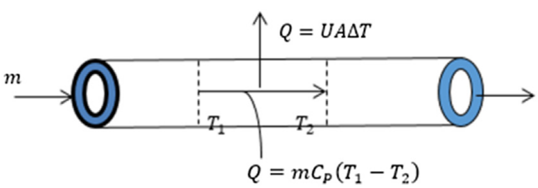

Figure 6. Relation between gas temperature and ambient temperature.

gas turbine output power. For every 1°*C* rise in ambient temperature, the gas turbine power output drops by ~1% and the centrifugal compressor polytropic head increases by ~1.1%, as shown in Figure 10.

The variation in power for both equipment must be compensated to maintain a steady flow of natural gas across the trans-Saharan gas pipeline route. Unless an inlet cooling technique is used, little can be done regarding the ambient temperature. This implies that the gas turbine would require more fuel to compensate for the reduction in the gas turbine shaft power and increase in the centrifugal compressor power to maintain a steady flow rate of the natural gas across the pipeline network.

Figure 11 below shows the matching of the gas centrifugal compressor power demand with the gas turbine shaft output power at different TET as the ambient temperature fluctuates daily in winter, dry and hot seasons across the Trans-Sahara pipeline stations. An increase in ambient condition from ISO increases the centrifugal compressor's

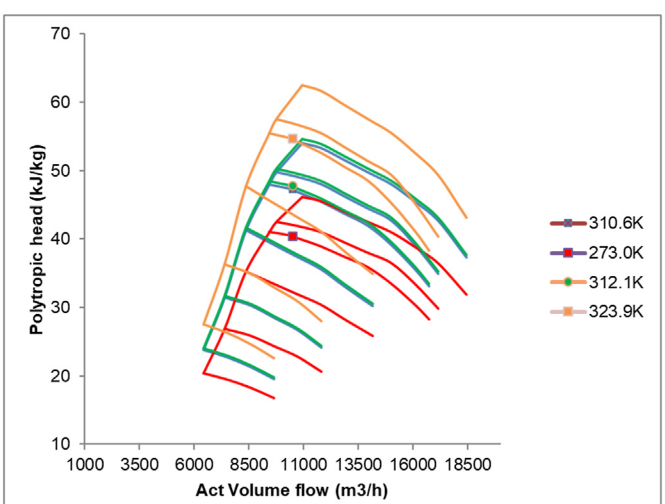

Figure 7. Polytropic head reference point changes in Centrifugal Compressor map due to Ambient temperature.

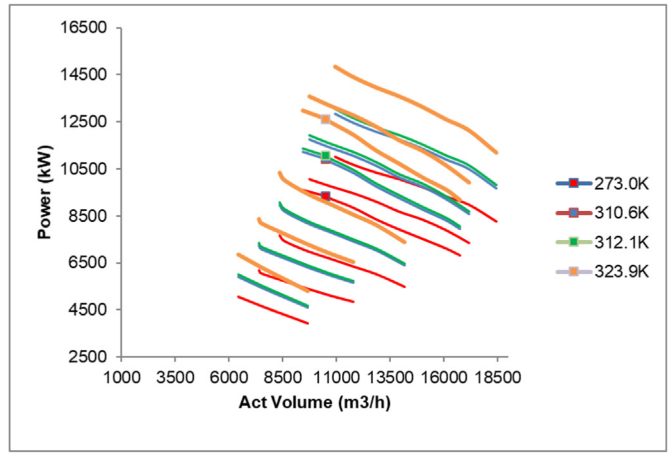

Figure 8. Power demand reference point changes in Centrifugal Compressor map due to Ambient temperature.

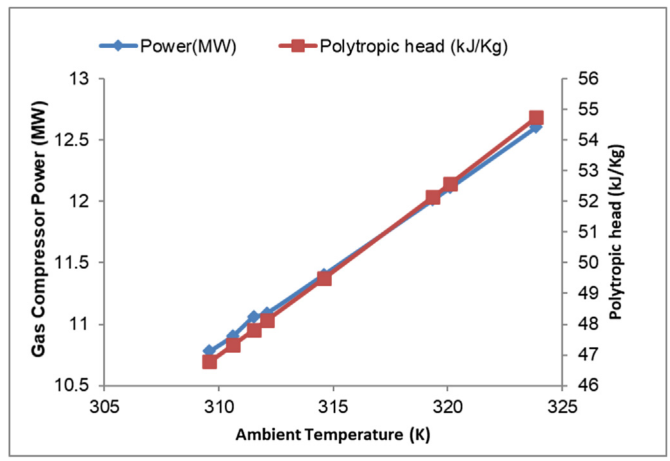

Figure 9. Effect of Ambient temperature on compressor performance.

polytropic head. As such, the gas centrifugal compressor power demand, whereas the same effect on the gas turbine resulting in a decrease in gas turbine output power for the three seasons. For every 1% increase in ambient temperature, a 3.5% increase in power is required to drive the gas compressor and a 1% decrease in gas turbine output power. The gas compressor power demand is matched with the gas turbine shaft output power at the time of day when the ambient temperature was minimum, mid and maximum (3:00 h, 9:00 h, 15:00 h) respectively for the three

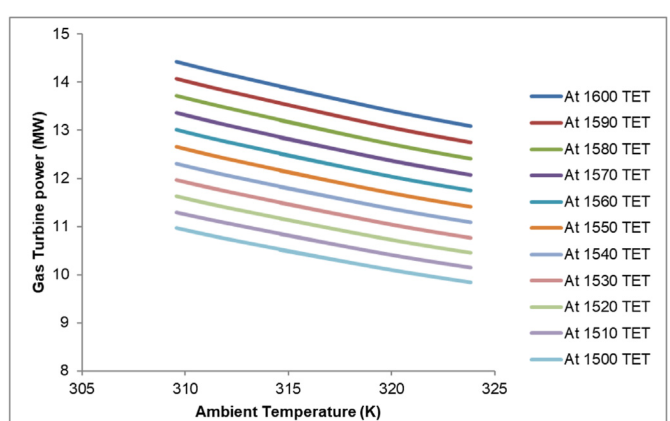

Figure 10. Effect of ambient temperature on gas turbine power.

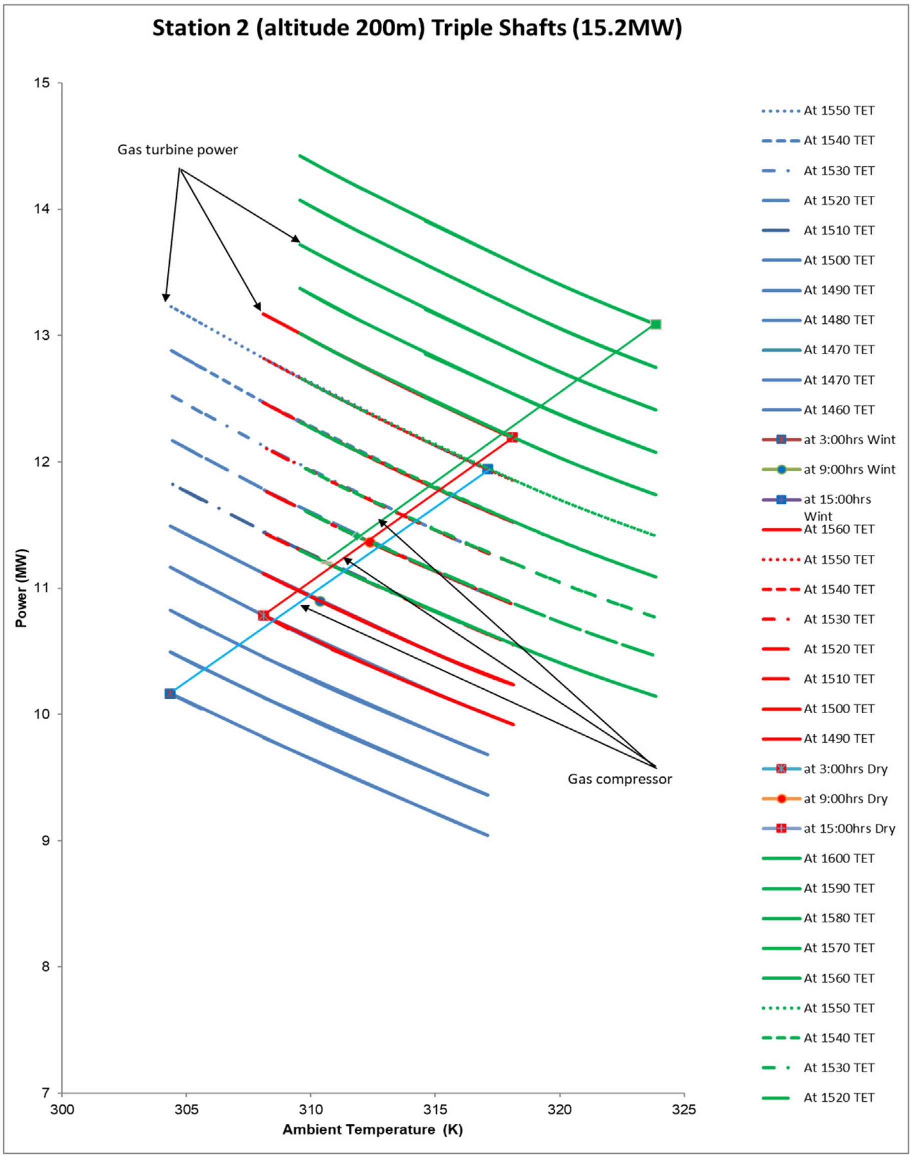

Figure 11. Gas compressor & Gas turbine power matching.

seasons under consideration. The graph shows that both equipment power matches at the intercession of gas turbine shaft output power with the centrifugal gas compressor power at different TET. In the hot season, at 3:00 h, the gas compressor power requirement was 10.90 MW, and the gas turbine TET of 1510 K will provide the power matching for the centrifugal gas compressor. Table 4 summarizes the gas turbine and gas compressors' matching point at 3:00 h, 9:00 h, 15:00 h, and during the hot season.

As the temperature of each season and time of the day varies, it is expected that the gas turbine can provide the power required always to drive the centrifugal gas compressor. This paper did not consider the effect of degradation on the power matching of the centrifugal compressor and the gas turbine, which the authors addressed in other work.

Table 4. Power matching for the hot season.

Hot Season			
Time	Temperature (K)	Gas Compressor Power (MW)	Gas Turbine Power (MW)
03:00 h	310.6	10.90	11.20
09:00 h	311.52	11.06	11.41
15:00 h	323.85	12.61	13.09

5. Conclusion

The TSGP project is a Nigerian Government initiative to rejig its gas development and transportation infrastructure to meet its internal and external market demand. This timely investment drive would ease the current energy transport imbalance caused by a few existing distribution pipelines and add millions of dollars to the country's trade exchange. However, several roadblocks such as the international gas polities, the pipeline route's geopolitical dangers, and project funding could potentially hurt the project concept. This paper aims not to address policy issue but to look at the technical problem of matching the gas compressors with the gas turbine drivers at varying ambient conditions.

In this paper, a steady-state model which simulates and matches the gas compressor and its driver for multiple compressor stations has been analysed using the TSGP project as a case study. The introduced model applied to performance simulation considered the varying effect of ambient conditions on the gas compressor, gas turbine and pipeline pressure requirement. The overarching results on the effect of ambient temperature on both the centrifugal compressor and gas turbine collaborate with previous works, which set the foundation for this study. Rather than consider a single point temperature, the ambient temperature was segmented in seasons and hourly to capture a more realistic scenario for matching the centrifugal compressor power demand with the gas turbine power. Considering a wide possibility of operation envelop would enable proper matching of both pieces of equipment, improving overall system efficiency, saving cost, and improving overhaul time intervals and reliability. One of the assumptions considered in this study is that the pipeline pressure remained constant; hence any fluctuation is compensated by the centrifugal compressor to maintain a steady flow.

The result shows that the centrifugal compressors power requirement increases with an increase in ambient temperatures for each season whilst the gas turbine power decreases with an increase in ambient temperatures. For every 1% increase in ambient temperature, a 3.5% increase in power is required to drive the gas compressor and a 1% decrease in gas turbine output power. The matching point between the centrifugal compressor and the gas turbine output power at varying environmental condition occurs at the point where the turbine entry temperature corresponds with centrifugal power and gas turbine power requirement. For example, the TET increased from 1469 K to 1520 K as the ambient temperature increases from 300 K to 325 K, putting a significant increase in power required by the centrifugal compressor. To meet centrifugal compressor power demand due to changes in the ambient temperature will require raising the gas turbine TET to produce the output power corresponding to this demand. A significant part of this matching would affect the gas turbine's life cycle cost due to an increase in the gas turbine entry temperature to compensate for the variations in power. A detailed life cycle cost assessment has been provided in another publication by the authors. The study did not consider the effect of degradation on both pieces of equipment during the matching procedure analysis, which is addressed in future studies.

Declarations

Author contribution statement

Nasiru Tukur: Conceived and designed the experiments; Performed the experiments; Analyzed and interpreted the data; Wrote the paper.

Emmanuel O. Osigwe: Analyzed and interpreted the data; Contributed reagents, materials, analysis tools or data; Wrote the paper.

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Data availability statement

Data will be made available on request.

Declaration of interests statement

The authors declare no conflict of interest.

Additional information

No additional information is available for this paper.

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