# Optimising Gas Turbine Characteristics Using Shaft Speed on Gas Path Analysis

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## *Abstract*

*This research endeavors to utilize gas path analysis and vibration monitoring techniques to develop a proactive model for analyzing gas turbine (GT) characteristics based on shaft speed. The study begins by formulating the governing equations of the rotor shaft system to accurately capture its dynamics. Specifically, the modeling of shaft speed dynamics is conducted to understand its impact on GT performance. Various sensors are employed for detecting faults in gas path parameters, crucial for maintaining turbine health. The results are presented which shows the 4976rpm is highest value of shaft speed and the lowest value is 452rpm. faults are likely to occur at very high speed and this is shown in this research as fault probability is increasing with an increase in shaft speed. Surface plots illustrating the effects of shaft speed in conjunction with pressure and temperature on identified faults are shown in this research. This research aims to provide valuable insights into optimizing GT operations and mitigating potential issues, ultimately enhancing turbine reliability and efficiency.*

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## 1.0 INTRODUCTION

Gas turbine engines (GTEs) have garnered extensive acceptance within the offshore oil and gas sector. They have been embraced to offer a cost-effective solution for meeting the increasing need for lightweight, high-powered generation equipment suitable for offshore applications. By employing GTs as the primary mover in offshore installations, it enables enhanced production capacity with reduced capital investments. The inception of GTE usage dates back to 1791 when John Barber patented the technology (Amit and Yogita, 2017).

Every constituent of mechanical systems experiences vibrations due to shaft rotation based on its speed. This vibration is characterized as the oscillating motion of mechanical systems around a central point, which can manifest as rectilinear or torsional. Torsional Vibration (TV) specifically refers to the periodic angular movement of elastic shafts with circular rotors rigidly attached. This type of vibration is unique to gas turbine (GT) rotors, which comprise blades, discs, and shafts. The compressors and turbines of GTs are regularly impacted by vibration (Ogbonnaya et al., 2013a).

The rotor plays a pivotal role in absorbing pressure energy, converting it into mechanical energy, and subsequently transforming it into electrical energy and other technological processes. Imbalance, misalignment, eccentricity, and looseness issues are prevalent in GT rotors and can lead to catastrophic failures and prolonged downtimes.

The importance of shaft speed vibration is to trigger operations and maintenance departments to formulate and design effective programme that will restore the system to work (Amos, 2007). It is on this ground that vibro activity is built into the design, operation and maintenance programme of the GT plant.

The power plant industry is concerned with the production of power plants with increase of unitary power, decrease of specific capital expenditure and increase in operational media environment (Grzadziela and Charchalis, 2011). Maintenance is the process or art of restoring and prolonging the life and operation of a component or system. Over the years, maintenance programme such as routine checks, preventive maintenance and complete overhaul (corrective) of systems at breakdown were applied and benefits achieved.

Thereafter it was discovered that certain expenditures can be eliminated or reduced by predicting failures (condition monitoring) and scheduling a maintenance programme at the right time (proactive maintenance). Proper implementation of preventive maintenance and condition monitoring programmes has also reduced capital expenditures by 25% (Ogbonnaya, et al 2013a).

The power plant industry is therefore adopting proactive maintenance policies that change the reactive maintenance culture to proactive programme. It includes vibration monitoring activities and root cause analyses of major systems and components. This proactive maintenance strategy receiving high recognition is capable of reducing capital expenditures, increasing productivity resulting in high reliability and availability of plants (Ogbonnaya, et al 2013b).

This research is intended to employ gas path analysis, vibration monitoring techniques and come up with a proactive model to analyse GT characteristics using shaft speed. It also considered the use of Matlab software to carry out this analysis.

## 2.0 Formulation of Governing Equation of Rotor Shaft system

In the formulation of governing equation of rotor dynamic systems, classical dynamic equations are employed. These include the Newton's laws of motion, the beam theory, principle of conservation of energy. Finite element method has been found very useful in analysis of multi degree of freedom systems. The Holzer method is very useful in torsional vibration analysis.

Some of these methods are presented below including different rotor effects all arriving at the same equation of motion. The analysis for one system may be different from another (Bently 2002). The equations governing rotor dynamic machines for axial, lateral and traverse vibration are very similar to those of torsional vibration. As such the displacement parameter ($x$) and the mass (m) of the former must be replaced with the angular displacement ($\theta$ ) and moment of inertia (I) for torsional vibration analysis (Bently and charlse 2002). The rotor system takes dynamic forces as input and produce vibration as an output. According to Bently and Hatch 2002, modeling of a physical system use the following structured process:

- Stating the assumptions that will be used
- Define a coordinate system
- Describe the forces that act on the system which depend on displacement, velocity and acceleration
- Develop a free body diagram

- ➢ Derive the equation of motion
- ➢ Solve the equation of motion
- ➢ Compare the predicted behavior to the observed behavior of the machine.
- ➢ Adjust the model if description is not adequate.

## 2.1 Stating the Assumptions

To simplify the analysis, the following assumptions may be taken for modeling of rotor shaft system as shown in figure 1

- The rotor model may have one, two degree or multiple degrees of freedom system in the complex plane and will have the same number of differential equations.
- The rotor system parameters may be isotropic ie properties maybe radially symmetric. Parameter are properties of the system mass, stiffness and damping).
- Gyroscopic-effects may be ignored: It can cause a speed-dependent shift in rotating system Natural frequency.
- The rotor system may have significant fluid interaction (in the circular region i.e. fluid film bearing seals)
- Damping may be viscous and due to fluid interaction
- Model may be linear, although machine behavior may be nonlinear, but can be approximated to linear for easy calculation.
- Fluid will completely surround the rotor system (360° $2\pi$, 180°, $\pi$)
- A Non synchronous rotating external force will be applied to rotor system (Bently and Hatch, 2002)

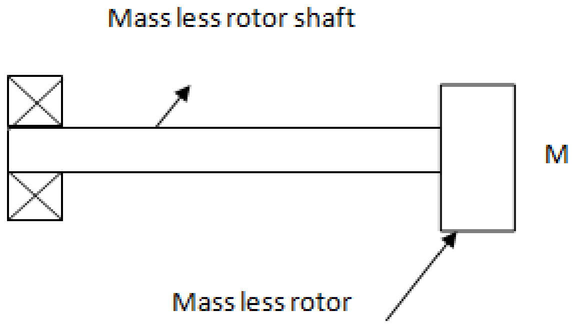

**Fig 1:** Rotor system (Tiwari & Bhaduri, 2017)

## 2.2 Coordinate System of the Rotor Shaft

$$r = x + jy \qquad (1)$$

$$j = \sqrt{-1} \quad j2 = -1 \qquad (2)$$

$$A = |r| = \sqrt{x^2 + y^2} \qquad (3)$$

*r =Rotor position vector*

$$\theta = arc\ tan\ \frac{y}{x} \qquad 4$$

$A =$ *length, magnitude of r*

$$\left.\begin{aligned} x &= A\cos\theta \\ y &= A\sin\theta \end{aligned}\right\} \qquad 5$$

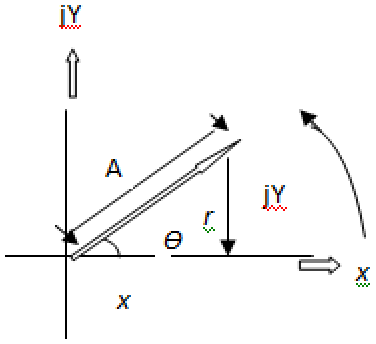

Fig 3.6 Motion from X to Y

$$r = x + jy = Aej\theta \ \ (Euler\ eqn) \qquad 6$$

$$e = 2.718\ 28\ base\ of\ Nat\ log.$$

## 2.3 Displacement and Acceleration

If r rotates around origin w(rad/s)

$$\theta = \omega t + \alpha \qquad 7$$

$\theta$ = absolute phase angle at t = 0

$$r = Ae^{j(\omega t+\alpha)} \qquad 8$$

$$\text{but velocity } v = \frac{dr}{dt} = \dot{r} = j\omega Ae^{j(\omega t+\alpha)} \qquad 9$$

$$\text{acceleration } a = \frac{dv}{dt} = \ddot{r} = -\omega^2 Ae^{j(\omega t+\alpha)} \qquad 10$$

Lambda ($\lambda$) model: $\lambda$ Reduce the complex behavior of fluid in relative motion to a single parameter (Walker, 2004). When a viscous fluid is contained in the annular region between two concentric cylinders rotating at different angular velocities, the fluid will be dragged into relative motion which is complicated. Fluid in annular region with average angular velocity $\Omega = 0.5\ \Omega$

$$\lambda = \frac{V_{average}}{\Omega} \qquad 11$$

$$V_{average} = \lambda\Omega \qquad 12$$

$$\text{Radial force component } F_B = -K_b r \qquad 13$$

$$\text{Tangential force component } F_T = jD\lambda\Omega r \qquad 14$$

$$Fs = -\,Kr \qquad 15$$

$\lambda$ = Fluid circumferential average velocity ratio

$\lambda$ = 0.35 to 0.45 (hydrodynamic bearings) and stiffness is the Bearing force

$K_b$= bearing stiffness constant

D = damping constant of bearing

j = direction of $F_T$ 90° relative to r

K = all spring stiffness in rotor system (shaft, bearing, support etc).

## 2.4 Modelling Shaft Speed

To develop a mathematical model for shaft speed dynamics in a gas turbine, we can start with the basic principles of rotational motion. Let's denote the shaft speed as ω\omegaω (in radians per second). The dynamics of shaft speed can be described using Newton's second law for rotation, which relates the torque applied to the shaft to its rotational inertia and angular acceleration.

$$\boldsymbol{\tau} = \boldsymbol{I} * \boldsymbol{\sigma} \qquad 16$$

Given that the torque on the shaft is typically the difference between the torque supplied by the turbine and the torque required by the load, we can write:

$$\boldsymbol{\tau} = \mathrm{T_{turbine}} - \eta_{\mathrm{load}} \qquad 17$$

Assuming that the load torque is constant, we can express the turbine torque as a function of shaft speed ω

$$\mathrm{T_{turbine}} = f(w) \qquad 18$$

considering the relationship between torque and shaft speed. This can be described using the power equation shown

$$\mathrm{T_{turbine}} = \frac{\mathrm{P_{turbine}}}{\mathrm{w}} \qquad 19$$

Combining these equations, we get

$$\frac{\mathrm{P_{turbine}}}{\mathrm{w}} = \eta_{\mathrm{load}} + I * \boldsymbol{\sigma} \qquad 20$$

Assuming negligible losses, we can express the power output of the turbine as a function of shaft speed

$$\frac{\mathrm{g_{(w)}}}{\mathrm{w}} = \eta_{\mathrm{load}} + I * \boldsymbol{\sigma} \qquad 21$$

Where

τ is the torque applied to the shaft (in Newton-meters).

I is the moment of inertia of the rotating parts about the axis of rotation (in kilograms per meter squared).

α is the angular acceleration of the shaft (in radians per second squared).

P turbine is the power output of the turbine (in watts).

## 2.5 Membership Functions (MF)

The MF parameters used in this work was collected and used for this analysis. It comprises the membership function type, linguistic variable associated with each group of functions, their universe of discourse, and fuzzy linguistic terms.

i. Linguistic Variable/Universe of Discourse

The linguistic variables and the universe of discourse (range of values for each variable) used in this work is presented below as shown in table 1;

Table 1: Universe of Discourse

| **Linguistic Variable** | **Universe of Discourse** | **Description** |
|---|---|---|
| Temperature | [ 0, 55 ] | Temperature (input variable) |
| Pressure | [ 0, 1000 ] | Pressure (input variable) |
| Shaft Speed | [ 0, 5000 ] | Shaft Speed (input variable) |
| Failure | [0 1] | Failure Probability (Output Variable) |

i. Membership Function Model

This work makes use of the triangular membership function. This function is presented mathematical as shown below;

$$f(x;a,b,c) = \begin{cases} 0\,, & x \le a \\ \frac{x-a}{b-a}, & a \le x \le b \\ \frac{c-x}{c-b}\,, & b \le x \le c \\ 0\,, & c \le x \end{cases} \qquad 22$$

Where:

X: is the crisp input

A: is the left leg of the membership function

B: is the centre of the function

C: is the right leg of the membership function

## 2.5 Sensors Used For The Detection of Fault in the Gas Path Parameters

*i.* *Temperature sensors:* is a temperature sensing equipment used in the radiation pyrometer. It is located in a number of places within the GT

*ii. Pressure Sensors:* The gas turbine uses the differential pressure type transducer for pressure sensing

ii. *Speed Sensor:* The sensor used in merasuring the shaft speed of the gas turbine is the tachometer which is connected to the turbne shaft and generates a DC signal having a magnitude proportional to the actual speed of the turbine.

## 3.0 Result and Discussion

### 3.1 Surface Plot

Surface plots are diagrams of three dimensional data. Rather than showing the individual data points, surface plots show a functional relationship between a designated dependent variable Y, and two independent variables X and Z. surface plots are important because it helps to organize and analyse in a well structured form making it easier to interpret data. The dependent variable is the fault on the y axis while the independent variables are the shaft speed on the x axis and pressure on the z axis. Fig. 2 shows the effect of shaft speed and pressure on fault. For For the shaft speed plot, the highest value of shaft speed at point 7, 8, 9, 10 (x axis) is 4976 (y axis) and the lowest value for the shaft speed is at point 1 (x axis) is 452 (y axis) in which fault is like to occur at this point. The shaft speed value used in the work during system testing is presented in figure 3; The importance of shaft fault analysis for an early detection and reliable solution of machine malfunctions is well demonstrated in this research.

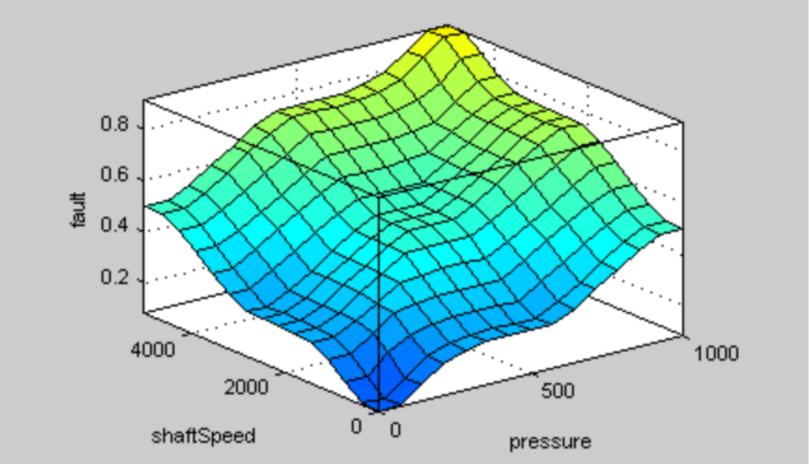

Figure 2: Effect of shaft speed and Pressure on Fault

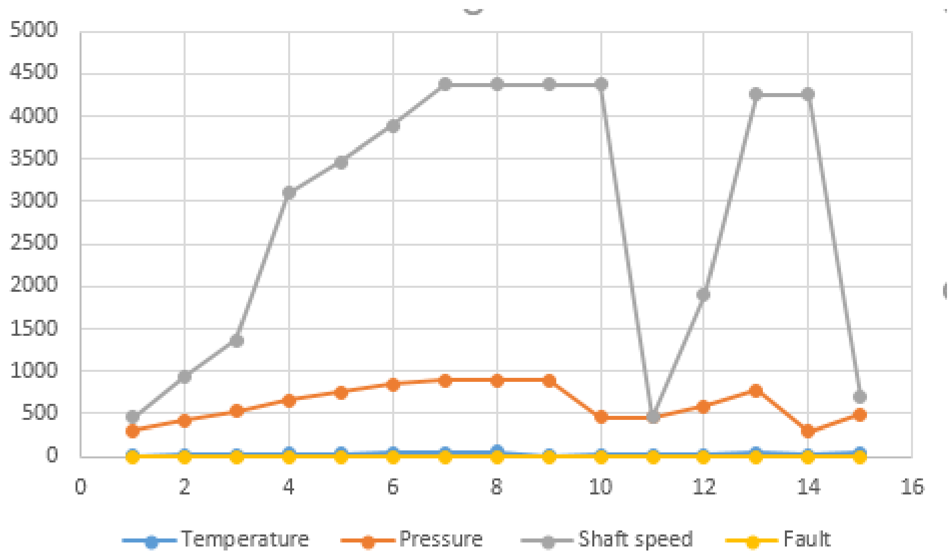

Figure 3: Shaft Speed Plot

These measurements enable an improved identification of common machine malfunctions such as rub contacts, different types of unbalance sources or instability problems. Their added value for the detection of excessive shaft vibrations that risk remaining unnoticed by standard absolute bearing vibration sensors is also well illustrated in the this result. The dependent variable is the fault on the y axis while the independent variables are the shaft speed on the x axis and temperature on the z axis. Fig. 4 shows the effect of the shaft speed and temperature on fault.

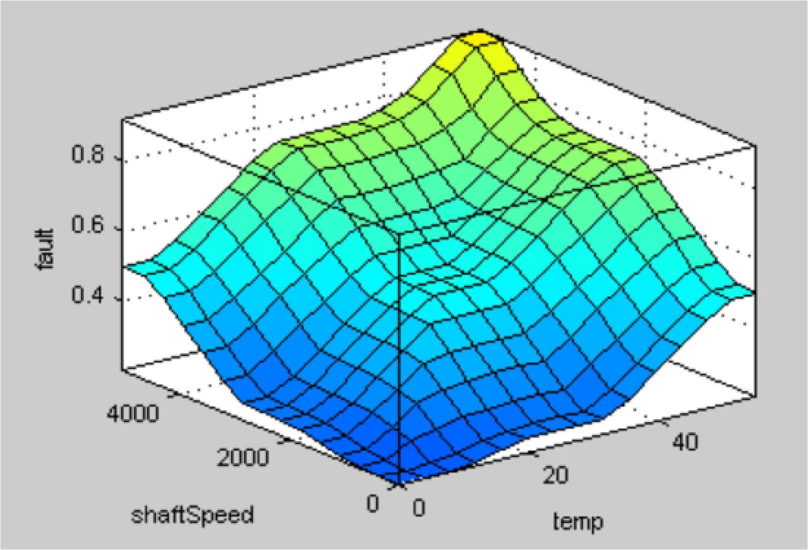

Figure 4: Effect of shaft speed and temperature on fault.

## 4.0 Conclusion

Optimizing gas turbine characteristics through varying shaft speed, as elucidated by gas path analysis, represents a significant advancement in the field of turbine engineering. By delving into the intricacies of gas path dynamics of the turbine system, engineers can meticulously fine-tune operational parameters to achieve enhanced efficiency, reliability, and performance.

The utilization of shaft speed as a focal point in gas path analysis offers a nuanced approach to optimizing turbine operations. By precisely controlling shaft speed, engineers can optimize the interaction between various turbine components, ensuring harmonious operation and minimizing detrimental effects such as vibration, imbalance, and inefficiencies.

Leveraging gas path analysis to optimize gas turbine characteristics holds promise for addressing contemporary challenges in energy production, including the quest for increased power output, reduced emissions, and improved fuel efficiency. By gaining deeper insights into the intricate interplay between gas flow dynamics and turbine components, engineers can devise innovative strategies to maximize turbine performance while minimizing environmental impact.

The pursuit of optimizing gas turbine characteristics through the strategic manipulation of shaft speed via gas path analysis represents a paradigm shift in turbine engineering. This approach not only promises to enhance turbine efficiency and reliability but also holds the potential to drive forward the transition towards sustainable and environmentally conscious energy production. As researchers and engineers continue to refine and expand upon these methodologies, the future of gas turbine technology appears brighter than ever before.

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