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Performance Evaluation of the Compressor Unit of a Gas Plant: A Case Study of Tunnu Gas Booster Station

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ABSTRACT

This research work presents the results of Performance of a compressor unit carried out on Tunu Booster Station facility. The coefficient of performance, isentropic efficiency and power supply to the compressor unit was evaluated from the Gas plant. The effects of the compressing system was investigated by using thermodynamics models to compare the performance of the existing system. The results indicated that the power of the compressor, isentropic efficiency and coefficient of performance of the compressor gave values of 33899kw, 0.83 and 1.2. The deviations from the existing plant values were 35.6%, 31.7% and 20% respectively. Other parameters that were evaluated include the failure rate (ʎ) and the mean time between failures (MTBF). During the evaluations, unexpected breakdowns, decrease in productivity and high maintenance cost associated with gas compressing system failures were minimized by using maintenance models to checked the performance of the compressor. The fault tree analysis (FTA) was primarily employed to evaluate the root cause of failure of previous gas compressing systems performance while the linear regression model derived from Weibull's exponential distribution was employed to obtain the shape and scale parameters. The results of the Failure rate function and the mean time between failures (MTBF) of the system obtained shows that the failure rate (λ) for the existing and proposed models were 0.016% and 0.052% estimated at 1 failure/year during its 7200 hours of operation. Also, the gas compressing system's probability of failure and the mean time between failures (MTBF) were 34.5% and 4.12 years respectively for the proposed model. Results also indicated that the performance of the system were 65.5% and 51.3% in the proposed and existing models respectively for 7200 hours operation yearly. This also confirms the fact that the proposed model had significant improvement when compared to the existing model.

Keywords: gas compressing; Booster Station; Gas plant

1. INTRODCTION

1.1 Background of the Study

Failure of the plant or equipment of the plant has negative safety, environmental and economic impact to plant owner and host community. Reliability, operation

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and maintainability of process plant are what determine its performance. For continuous operation of process plant and for the plant to continue to deliver its required output from time to time, the components of the plant need to be replaced, repaired, overhauled or removed. Deterioration of equipment that result from wear, tear, and ageing, increases chance of failure and leads to decreased performance of equipment. Increase cost of operation, technical ageing, poor quality and low-level production are as a result of gradual regression of operational life of equipment [1]. Maintenance approach can be used to mitigate this impact of failure. When Maintenance approach or strategy is not proper or appropriate for the plant, it can lead to increase cost and without justifiable improvement in equipment performance. Maintenance must be synchronized with production requirement and demand to ensure optimum equipment availability, minimal downtime and production loss. Equipment with high production demand will require more maintenance attention than equipment with less production demand [2].

Natural gas has been used for electricity generation in periods of peak electricity demand [3]. Besides being used as a source of electricity generation, natural gas has several application ranging from electricity generation, cooking, domestic heating, air conditioning and refrigeration unit application. It is used in the manufacturing of fabrics, anti-freeze, fertilizer and some polymer materials like plastics. It is a major source of energy in the pulp and paper, chemical, petroleum, refining, paint, glass, metal and food industries [4].

1.2 Aim of the Research

The aim of the research work is to evaluate and improve on the performance of the Compressor unit of Tunu Booster Station Gas Plant.

1.3 Objectives of the Research

The objectives of this research encompass

i. Use thermodynamics models to evaluate the performance of the compressor unit of Tunu Booster Station Gas Plant.
ii. Extract data from the plant operational data history file for the evaluation of the performance of the compressor unit of the Gas plant.
iii. Apply existing models and proposed maintenance models for the evaluation of Tunu Gas Booster Station performance.
iv. Use Probability models and fault tree analysis to evaluate the performance of the compressor unit of the gas plant.

2. MATERIALS AND METHODS

2.1 Materials

The various materials that aid the achievement of this research work are as follows:

i. Inlet manifolds
ii. Test separator
iii. High pressure separator
iv. Low pressure separator
v. Surge vessels
vi. LP gas inlet vessel
vii. Compressor

Performance and maintenance models were major tool apply in this research work

2.2 Methods

The methodology adopted for this research work is the collection of data from the company for the evaluation of thermodynamics parameters. The data was statistically analyzed to obtain the relevant thermodynamics parameters; the temperatures, the pressure and the flow rate. The equation is been applied for the gas and liquid flow rates evaluation from the production wellhead to TUNU BOOSTER STATION. Hence, to actualized this research work, the four methods applied is been summarized as;

i. The application of established thermodynamics models in conjunction with plant operational data were been used to evaluate thermodynamics parameters performance.
ii. Plant operational data result were presented for thermodynamics parameters evaluation.
iii. Existing maintenance model and proposed maintenance model of Tunu Gas Booster Station were evaluated.
iv. Probability model development (Fault Tree Analysis) were been carried out to evaluate the plant performance.

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2.2.1 Temperature, Pressure, Power and Coefficient of Performance Models

An equation was established to monitor the flow rate of gas and liquid from production wellhead to HP metering separator in Tunu booster station. [5] carried out a research work on gas and liquid flow rate. During their finding, the following equation for gas and liquid flow rate was established; [6]

$$\lambda_l = \frac{V_{sl}}{(V_{sl}+V_{sg})} = \frac{M_l}{M_l+M_g} \tag{1}$$

λ_l =Liquid input content/Liquid volume fraction

$V_{sl} = The$ Superficial liquid velocity

$V_{sg} = The$ Superficial gas velocity

M_l =Mass flow rate of liquid

M_g =Mass flow rate of gas

From equation (1);

$$V_{sl} = \frac{M_l}{\rho_l A)} \tag{2}$$

where:

ρ_l=Density of liquid

A= Cross sectional area of the pipe

$$V_{sg} = \frac{Mg}{(\rho_g A)} \tag{3}$$

ρ_g = Density of gas

$$V_m = V_{sl} + V_{sg} \tag{4}$$

V_m = The mixture velocity

$$\lambda_g = 1 - \lambda_g = \frac{Mg}{Ml+Mg} \tag{5}$$

λ_g = Gas void fraction

$$H_g = 1 - H_l$$

H_g =Gas hold up fraction

H_l =Liquid hold up fraction

$$H_l = a \tag{6}$$

The compression of gases is been founded on the following laws; Boyle's, Charle's, Avogadro's, Ideal Gas and Bernoulli's. The thermodynamic of gas compression are derived from these laws. In the process industry, it is only few gases that obeyed gas relationship; hence, real gas equation should be characterized and used for the behavior of most process gases. This research work

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focuses on the Tunu Booster reciprocating compressor type. Figure 1 shown below is a pressure volume diagram depicted on the compression process. The process is represented as an adiabatic process with constant (k) and related as:

PV^k = constant (7)

The compression stage ratio should take 3:1. Reciprocating compressors always have discharge relief valves for safety against exceeding parameters in other to attained design limit since the delivered pressure is highly expected to overcome the discharge back pressure. Since this type of compressor is associated with centrifugal type, the compressor does not have choke limitations as well as surge [7].

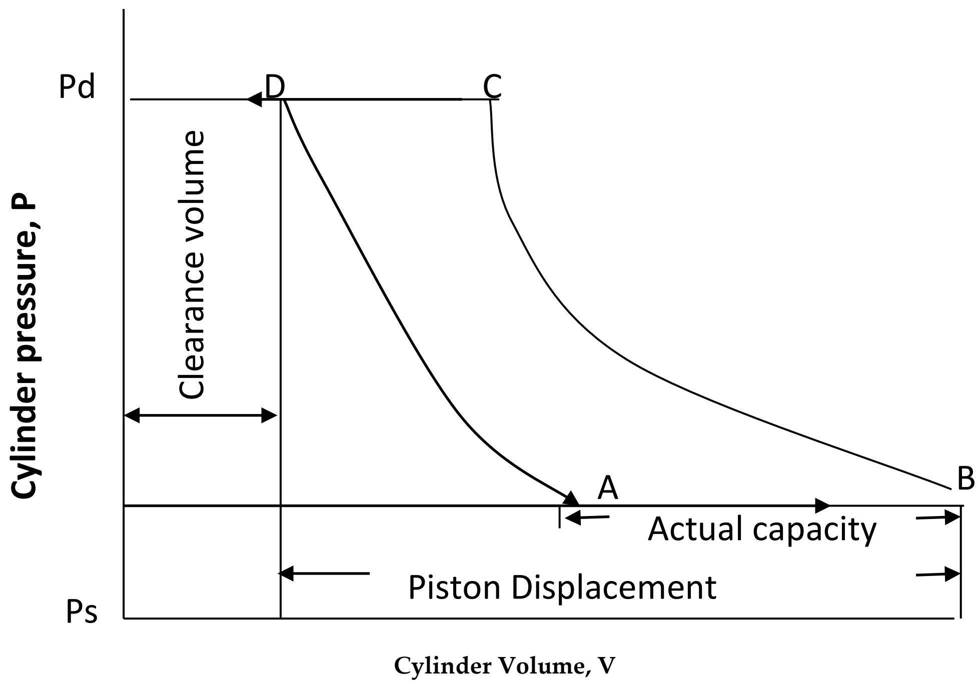

Figure 1: P- V Diagram for Reciprocating Compressor

The work/power required for adiabatic compression for real gas compression process is given as: [8]

$$H_{ad} = \frac{Z_{avg}\, R\, T_S}{MW\,(k-1)/k} \left[\left(\frac{P_d}{P_S} \right)^{\frac{k-1}{k}} - 1 \right] \tag{8}$$

The volumetric efficiency, E_V defined as the ratio of the gas handled at inlet conditions to the theoretical volume displaced by the compressed as a percent:

$$E_V = \frac{Actual\ capacity}{Piston\ Displacement} = 100 \left[\left(\frac{Z_S}{Z_d} \right) r_p^{\frac{1}{k}} - 1 \right] C - L \tag{9}$$

where:

r_p = pressure ratio based on absolute pressure

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C = Clearance volume

L = Effects from internal leakage, gas friction, pressure drop through valves, etc (approx. 5% for lubricating compressors)

It is also noted that as the pressure ratio across the cylinder increases, E_V decreases with an increase in internal cylinder clearance volume. E_V is not a constant parameter, it varies with fixed diameter, stroke, speed, internal clearance volume.

The theoretical discharge temperature is calculated as:

$$T_d = T_s \left[\left(\frac{P_d}{P_s} \right)^{\left(\frac{k-1}{k} \right)} \right] \quad (10)$$

where:

k = $\frac{C_P}{C_V}$ = Specific heat ratio

T_d, T_s = Absolute temperatures, K.

Calculating of efficiency of compressor

$$\eta_{isen} = \frac{Isentropic\ Enthalpy\ Compressor}{Actual\ Enthalpy\ Change}$$

$$= \frac{h_{2isen} - h_1}{h_2 - h_1} \quad (11)$$

The compressor power can be calculated as:

$$\text{Power } (\dot{W}_{in}) = ni\,(h_2 - h_1) = \frac{\dot{m}(h_{2isen} - h_1)}{\eta_{isen}} \quad (12)$$

where:

η_{isen} = isentropic efficiency

h_1 = suction enthalpy calculated at P_1, T_1, and composition, (2i)

h_{2isen} = isentropic discharge enthalpy at P_2, (or T_2), $S_{2isen} = S_1$, and composition (2i)

h_2 =Discharge enthalpy calculated at P_2, T_2 and composition (2i)

$\dot{m}$ = mass flow rate, kg/s.

The computation of compressor efficiency or power involves two steps:

i. Determination of the ideal or isentropic (reversible and adiabatic) enthalpy change ($h_{2,isen} - h_1$) of the compression process.

ii. Determination of the actual enthalpy change ($h_2 - h_1$).

Procedure to determine power based on equation of state:

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i. Assume steady state i.e. feed composition remain the same.

ii. Calculate suction enthalpy; $h_1 = f\left(P_1, T_1, and\ z_i\right)$

iii. Assume isentropic process and set;

$$S_{2,isen} = f\left(P_2, T_{2,isen}, z_i\right) = S_{f(1)} = f\left(P_1, T_1, and\ z_i\right)$$

i. Calculate the ideal enthalpy, $\left(h_{2,isen}\right)$ at discharge condition for known $z_i, T_2\left(or\ P_2\right)$ and $s_{2,isen}$

ii. Calculate the actual enthalpy, $\left(h_2\right)$ at discharge condition for known $z_i, T_2\ and\ P_2$

iii. Calculate isentropic efficiency and power.

The efficiency of compressor is estimated by:

$$k = [1.46 - 0.1(\eta - 0.55)](1 - 0.067\eta - AT) \tag{13}$$

Where:

T = Temperature, $K\left({}^{O}R\right)$

η = Gas relative density; ratio of gas molecular weight to air molecular weight

$A = 0.000272\,(0.000151)$

The Actual discharge temperature based on isentropic path can be estimated: (Rajput, 2007)

$$T_2 = T_1\left[1 + \frac{\left[\dfrac{P_2}{P_1}\right]^{\left(\frac{k-1}{k}\right)} - 1}{\eta_{isen}}\right] \tag{14}$$

$$\Rightarrow \eta_{isen} = \left(\frac{T_1}{T_2 - T_1}\right)\left[\left(\frac{P_2}{P_1}\right)^{\left[\frac{k-1}{k}\right]} - 1\right] \tag{15}$$

on a polytropic path, the actual discharge temperature can be estimated as:

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} \tag{16}$$

$$n = \left[1 - \frac{\left(\dfrac{T_2}{T_1}\right)}{\left[\dfrac{P_2}{P_1}\right]}\right]^{-1} \tag{17}$$

Whereas the actual discharge temperature based on polytropic path can be given as:

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$$\eta_{poly} = \frac{\left(\frac{k-1}{k}\right)}{\left(\frac{n-1}{n}\right)} = \frac{n(k-1)}{k(n-1)} \tag{18}$$

The isentropic head is calculated as:

$$Head_{isen} = \left(\frac{n}{n-1}\right)\left(\frac{z_a R T_1}{MW}\right)\left[\left(P_2 / P_1\right)^{\left(\frac{n-1}{n}\right)} - 1\right] \tag{19}$$

Power is calculated as shown below based on an isentropic (reversible and adiabatic) process: (Rajput, 2007)

$$Power = \left(\frac{k}{k-1}\right)\left(\frac{z_a}{\eta_{isen}}\right) T_1 q_s \left(P_S / T_S\right)\left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{k-1}{k}\right)} - 1\right] \tag{20}$$

Also, power is been determine in a polytropic process using:

$$Power = \left(\frac{n}{n-1}\right)\left(\frac{z_a}{\eta_{isen}}\right) T_1 q_s \left(P_S / T_S\right)\left[\left(\frac{P_2}{P_1}\right)^{\left(\frac{n-1}{n}\right)} - 1\right] \tag{21}$$

Alternatively:

$$Power = \dot{m}\frac{\left(Head_{isen}\right)}{\eta_{isen}} = \dot{m}\frac{\left(Head_{poly}\right)}{\eta_{poly}} \tag{22}$$

Where:

$Head = Compressor\ Head, m$

$Power = Compressor\ Power, kW (HP)$

$R = Universal\ gas\ constant, 8.48 kgm/kmol.K$

$P_S = Standard\ Condition\ Pressure, kPa$

$P_1 = Suction\ Pressure, kPa$

$P_2 = Discharge\ Pressure, kPa$

$T_S = Standard\ Condition\ Temperature, K$

$T_1 = Suction\ Temperature, K$

$T_2 = Discharge\ Temperature, K$

$q_S = Gas\ volumetric\ rate\ at\ the\ standard\ condition, 5m^3/d$

$$z_a = Average\ gas\ compressibility\ factor = (z_1 + z_2)/2 \tag{23}$$

$z_1 = Gas\ compressibility\ Factor\ at\ suction\ condition$

$z_2 = Gas\ compressibility\ Factor\ at\ the\ discharge\ condition$

$MW = Gas\ Molecular Weight$

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For cycle analysis as seen in equation (24)

$$\text{Work per cycle} = \left(\frac{n}{n-1}\right) P_{in} V_{ind} \left[r_P^{\frac{n-1}{n}} - 1 \right] \tag{24}$$

where

r_P = ratio of P_2 to P_1

Mass delivered = mass induced

$$\frac{P_2 (V_2 - V_3)}{RT_2} = \frac{P_1 (V_1 - V_4)}{RT_1} \tag{25}$$

$$m = fc \frac{P_1}{RT_1}(V_1 - V_4) = fc \frac{P_2}{RT_2}(V_2 - V_3) \tag{26}$$

where:

fc = compressor rotational frequency, Hz.

The volumetric efficiency, η_v is also the gas actual volume sucked to the theoretical volume that could have been sucked on the absent of clearance volume.

$$\eta_V = \frac{V_1 - V_4}{V_2 - V_1} = \frac{\dot{m}}{\dot{m}s} \tag{27}$$

where;

$\dot{m}s$ = swept volume mass flow rate

$$m_S = fc \frac{P_1 - V_S}{RT_1} \tag{28}$$

V_s = swept volume

The theoretical work required for gas compressor is given by: [11]

$$\dot{W} = \begin{cases} fc \dfrac{n}{n-1} P_1 (V_1 - V_4) \left[\left(P_2 / P_1 \right)^{\frac{n-1}{n}} - 1 \right] & n = 1 \\ fc\ P_1 (V_1 - V_4) \left(P_2 / P_1 \right) & \end{cases} \tag{29}$$

For polytropic process of exponent, n:

The power supplied in adiabatic compression

$$\dot{W} = P = \dot{m}_e CpT \left[\left(P_2 / P_1 \right)^{2/7} - 1 \right] \tag{30}$$

Power supplied in isothermal compression is given by:

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$$\dot{W}iso,e = \dot{m}e\,RT_1 \quad In\left(\frac{P_2}{P_1}\right) \tag{31}$$

The shaft power is given by:

$$\dot{W}shaft,e = \dot{W}_{actual,e_e} + \dot{W}\ friction \tag{32}$$

The power required only for gas compression is known as the actual power

The overall efficiency η is given as:

$$\eta = \frac{\dot{W}ad,e}{\dot{W}shaft,\ e} \tag{33}$$

Adiabatic efficiency

$$\eta_{ad,\ e} = \frac{\dot{W}ad,e}{\dot{W}actual,\ e} \tag{34}$$

The isothermal efficiency is given as:

$$\eta_{iso,\ e} = \frac{\dot{W}iso,\ e}{\dot{W}actual,\ e} \tag{35}$$

Mechanical efficiency is calculated as:

$$\eta_{mech} = \frac{\dot{W}actual,\ e}{\dot{W}shaft,\ e} \tag{36}$$

Table 1: Thermodynamic Parameters from the Plant (Tunu Flow Station)

S/N	Parameters	Symbols	Values (Unit)
1.	Discharged pressure	Pd	10.3 barg
2.	Piston pressure	Ps	2.6barg
3.	Average compressibility factor of the gas	Zavg	0.960
4.	Pressure ratio	Rp	82.67
5.	Clearance volume	C	0.21
6.	Gas molecular weight	Mw	21.73g/gmol
7.	Universal gas constant	R	8.314J/mol K
8.	Mass flow rate	M	1.72Kg/s
9.	Constant pressure for specific heat capacity	Cp	1.9282KJ/kg^{0}C
10.	Compressors factor for piston	Zs	0.964
11.	Discharge compressor factor	Zd	0.986
12.	Effects from internal leakage	L	0.081
13.	Suction enthalpy	h_1	-15771.0KJ/kg
14.	Isentropic discharge enthalpy	h_{2isen}	-617.4KJ/kg
15.	Plant temperature	T_1	37^{0}C
16.	Pressure induced	P_1	3.5barg
17.	Discharge pressure	P_2	10.3barg
18.	Rotational frequency	F_c	2.1896m^2/s
19.	Volume induced	V_{ind}	0.18m^2/s

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20.	Actual Volume flow	V_4	$0.1m^2/s$
21.	Discharge enthalpy	h_2	3937.9KJ/kg

The input parameters for the simulation of the thermodynamic models derived above, are obtained from the TUNU flow station as shown in table 1.

3. RESULTS AND DISCUSSION

3.1. Results

The results obtained from the computation of the performance maintenance models presented in the previous pages, for the gas compressing system are discussed in this section.

Table 2: Summary of Calculated Thermodynamic Parameters

S/N	Parameters	Symbol	Value (Unit)
1	Discharge Temperature	T_d	49.75^{0}C
2.	Adiabatic enthalpy	h_{ad}	21.77KW/kg
3.	Volumetric efficiency	E_v	0.78
4.	Compressor efficiency	η	0.83
5.	Actual discharge temperature	T_2	60^{0}C
6.	Adiabatic constant	K	1.274
7.	Polytropic constant	N	1.81
8.	Compressor power	P	33899KW
9.	Polytropic efficiency	η_{pd}	0.48
10.	Work per cycle	w_{pc}	8.7J/rad
11.	Theoretical power	W_t	115KW
12.	Power supplied	P_{in}	371W
13.	Isothermal compressor power	P_{is}	4.785KW

Table 2 is the thermodynamic upgraded result for the Tunu compressor station carried out. The isentropic efficiency calculated is 0.83 as to that of the plant which is 0.63, the power of the compressor is 33899KW compared to 25,000KW generated at a feed rate of 21.73Kg/s. The coefficient of performance of the compressor is 1.2 times that originally generated of 1. Thus, the result thermodynamically proved viable and better compared to the plant result.

Table 3: Probability of Failure of Basic Events

S/N	Fault	Probability of failure (%)
1	Valve	4
2	Pressure packing ring	1.5
3	Piston ring	2
4	Piston rod cylinder	2
5	Vibration	15
6	Wear	5
7	Material failure	3
8	Corrosion	2

Table 3 shows the probability of failure of the gas compressing system components obtained using fault tree analysis (FTA). In determining the shape and scale parameters, failure rate function and the mean time between failures (MTBF) of the gas compressing system, the linear regression model derived from Weibull's exponential distribution was employed.

The probability of system failure F (t), which is the sum of failure of crucial components presented in Table 3, was obtained as 34.5%. Probability of failure below 50% indicates system effectiveness. The system performance was evaluated based on the probability that the system will operate for a specified period of time without failure, was obtained as 65.5%. This implies that the

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system will operate effectively without failure for 7200 hours, which is the average operational time for a year. Performance above 50% implies that the system is effective and would carry out its intended purpose without failing, for the specified period of time. The failure rate (λ) is the frequency of failure of the gas compressing system at any given time during its operational time. It was obtained as 0.00016%, which implies that the probability of failure of the system at any given point in time during its 7200 hours of operation is 0.052%, which is estimated at 1 failure/year. Lastly, the mean time between failures (MTBF) of the system is one of the best performance indicators applied to repairable systems. It is the average length of operating time without failure between one failure and the next failure. It was obtained as 4.12 years, which implies that based on the current number of hours the system is operated, it may likely fail in four years' time.

Table 4: Linear Regression Analysis

	N	$\sum Ft$	$(\sum Ft)^2$	$\sum (t)^2$	A	B	A	B
1A	64	2099.2	4406641	89440	**0.000578**	**-3.0608**	**43.01**	**0.0226**
1H	64	215.2	46311.04	63				
1V	64	6634.5	44016590	2099.2				
2A	64	149.8	22440.04	215.2				
2H	64	4559.8	20791776	6634.5				
2V	64	30.5	930.25	149.8				
3A	64	1001.7	1003403	4559.8				
3H	64	186.7	34856.89	30.5				
3V	64	3448.6	46958.89	420.1				
4A	64	1185.1	1404462	89440				
4H	64	977.5	170238.3	216.7				
4V	64	718.24	132080	26.8				

Table 4 shows the linear regression analysis at various test situations. See Appendix B for complete analysis. As seen from Table 4, Points (1A – 4V) denotes the points at which reading were taken, when the gas compressing system was operating in the axial, horizontal and vertical directions. In order to obtain the scale (α) and shape (β) parameters, which are crucial components in determining the failure rate and mean time between failure (MTBF) of the system, the mean reading at each of the four points were plugged into our regression model so that the constants a and b can be obtain.

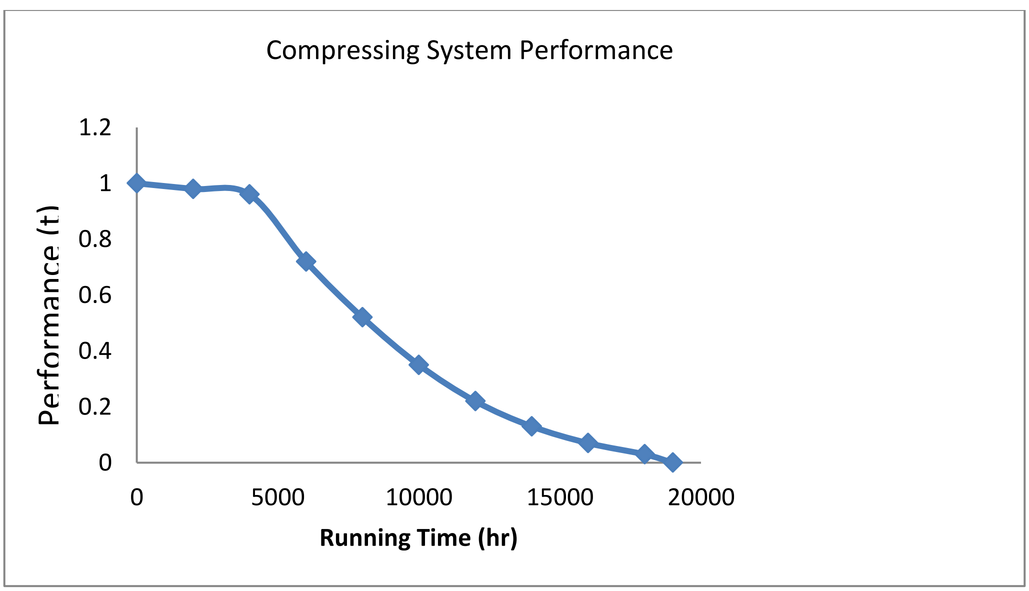

Figure 2: Compressing System Performance

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The performance of the gas compressing system, in terms of probability shows that the gas compressing system will operate for a number of time without failure, is shown in Figure 2. As the number of operating hours' increases, the performance of the system decreases. This implies that the system's performance is dependent on the number of operating hours. Also, the average number of operating hours of the gas compressing system at Tunu gas Booster station is 7200 hours, which is equivalent to 65.5% as seen in the Figure 2. Since the performance of the system is above 50%, it means that the system will continue to function properly for its 7200 hours.

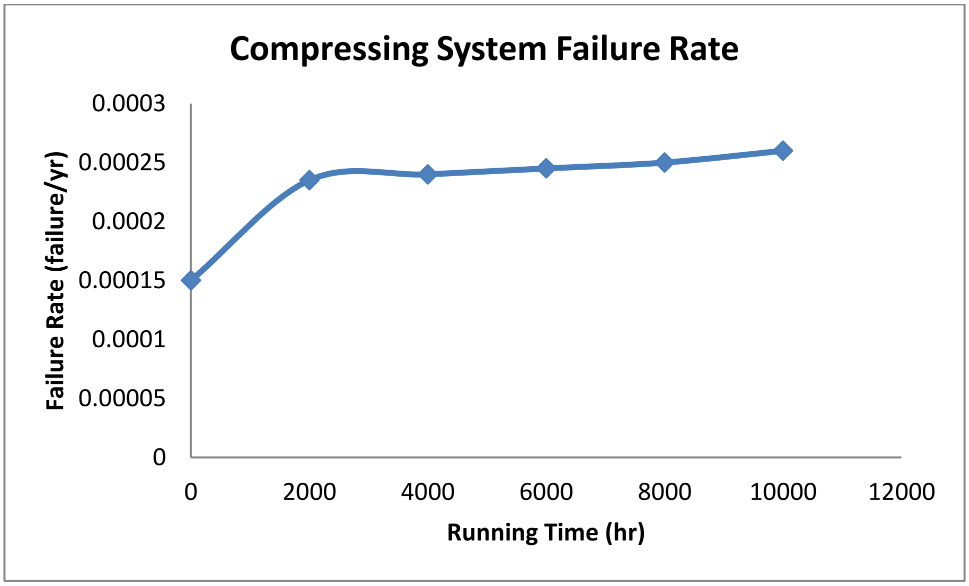

Figure 3: The Failure Rate of the Gas Compressing System

Figure 3 shows the failure rate of the compressing system. The failure rate is the frequency at which the system fails at any given point in time during its operating period. As seen in Figure 3, as the number of the system operational hours increases, the failure rate also increases. This implies that an increase in the operating time, results in a higher probability that the system may fail at any point. Also, at 7200 hours, which is the average number of hours the system is operated at Tunu Booster Station, the failure rate was obtained as 0.052%, which is equivalent to an estimate of 1 failure/year.

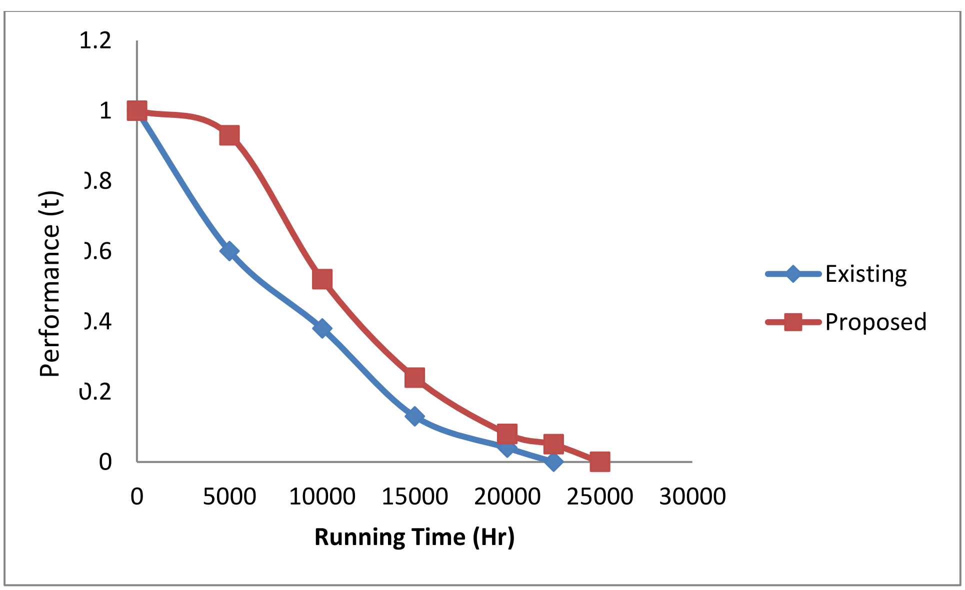

Figure 4: Existing Versus Proposed Gas Compressing System Performance

The mean time between failures (MTBF), been the key performance indicator for repairable systems, shows the system robustness measured in the past and predicts the near future rate of failure based on that measurement. As seen in Table 4, the

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mean time between failure (MTBF) of 4.12 years indicates that from previous root cause of failure analysis (RCFA) and predictive maintenances carried out, the gas compressing system at Tunu Booster Station may likely fail in 4 years-time. This means that the number of preventive and reactive maintenances carried out all through the 7200 hours in which the system is available, should be minimized. This would save cost associated with frequent preventive and reactive maintenances since the system will not fail in the next four years.

Comparison of the performance and failure rate between the existing maintenance model at Tunu Booster Station and the proposed performance maintenance model is shown in Figure 4. As seen in Figure 4, the performance of the system decreases in both models as the number of hours the system is in use increases. Although the performance of both models decreases with increased operation time, the proposed model is more reliable, as it has greater values at same operating times. Also, at 7200 hours, which is the average number of hours the system is operated yearly at Tunu Booster Station, the performance of the system was 65.6% and 51.3% in the proposed and existing models respectively. This also confirms the fact that the proposed model has significant improvement compared to the existing model.

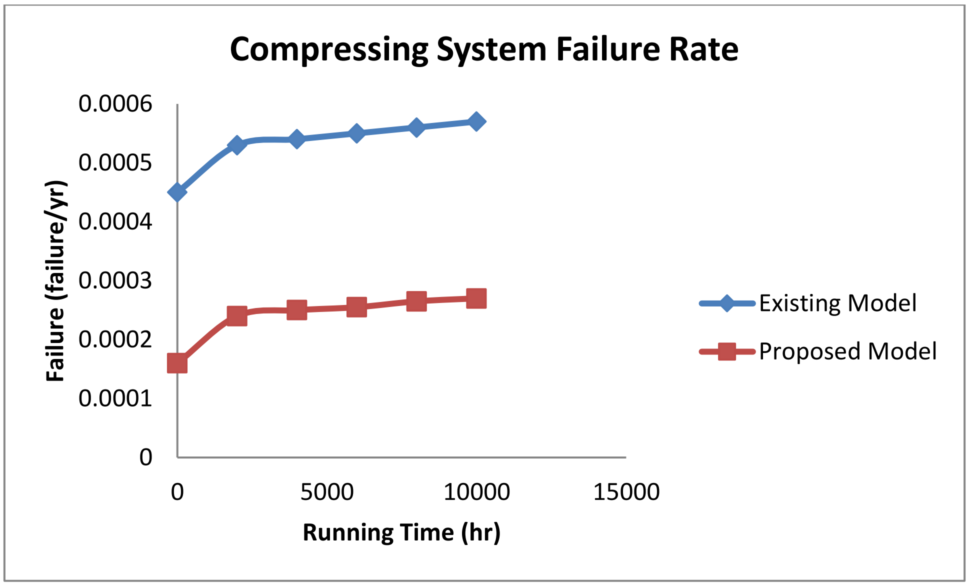

Figure 5: Existing Frequency of Component Failure versus Proposed

Figure 5 shows the frequency of failure of the gas compressing system at any given point in time, during its 7200 hours of operation. As seen in the proposed and existing model, as the number of operating hours increases, the probability that the system will fail at any time, also increases. Although, the failure rate of the existing model is greater than that of the proposed performance maintenance model at the same operating time. Also, at 7200 hours, which is the average number of hours the system is operating yearly at Tunu Booster Station, the failure rate was 0.00052% (4 failures/year estimation) and 0.00016% (1 failure/year estimation) for the existing model and the proposed performance maintenance models respectively. This also confirms that the proposed model has significant improvement when compared to the existing model.

4. CONCLUSION AND RECOMMENDATIONS

The results indicate that the power of the compressor, isentropic efficiency and coefficient of performance (cop) of the compressor gave values of 3399Kw, 0.83 and 1.2. The derivations from the existing plant values were respectively 35.6%, 31.7% and 20%. The model integrates fault tree analysis (FTA) and Weibull's exponential distribution in order to determine the system's performance and its mean time between failure (MTBF).The fault tree analysis (FTA), which is based on the root cause of failure of previous gas compressing systems, was employed to determine the system's performance, while the linear regression model derived from Weibull's exponential distribution was employed to obtain the shape and scale parameters, failure rate function and the mean time between failure (MTBF) of the system. However, results obtained showed significant improvement when compared to the existing maintenance model at the Tunu Gas Booster Station. From the results obtained, the proposed performance maintenance model showed significant improvement in terms of the system's performance and frequency of failure (failure rate) when compared to the existing model at Tunu Gas Booster Station, which means lower maintenance cost and higher availability.

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The mean time between failures (MTBF) of 4.12 years indicates that the gas compressing system may likely fail in 4 years' time. Hence, the number of preventive and reactive maintenances carried out on the system yearly, should be reduced, so as to save cost arising from maintaining functional components since the system will not result in functional failure in the next 4 years.

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Conflicts of interests

The authors declare that there are no conflicts of interests.

Data and materials availability

All data associated with this study are present in the paper.

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