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Optimisation Of Oil And Gas Facilities To Reduce Flaring Activities

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Abstract: *As the world continue to advance towards a cleaner and safer environment to live, it has become so imperative and necessary to reduce the release of toxic gases to the atmosphere. Third world and advancing countries like Nigeria with little or no regulatory policies on gas flaring has opened its atmosphere to dangerous and harmful pollutants through gas flaring activities. Hence, this research work is targeted to totally eradicate this menace. An employed mathematical modelling analysis of a centrifugal compressor stage design were carried out to compress flared low-pressure gases from oil and gas (O&G) wells managed by the Gbarain-Ubie Integrated O&G Production Facility. Results affirms that flared gases as low as 0.5bar can be pressurized to acceptable exporting pressure of 12bar. The compressor stage design analysis results for blade row stages shows an acceptable pressure of 13.62 – 13.93bar at stage three (3) with a high compressor efficiency of 84.34 – 84.03%. In conclusion, research reveals that the wasted flared gas could be turned useful and profitable gas which is capable of boosting the nation's economy. Therefore, adhering to this innovative technological research will minimize the environmental catastrophe from gas flaring.*

Keywords— Compressor, Flaring, Modelling, Oil and Gas, Stage

1. INTRODUCTION

Gas flaring activities as a result of O&G exploration is a major research concern. Meanwhile, flaring is considered as the last safety defense line in the O&G plants that produce flammable toxic and harmful products via flare headers to the atmosphere [1]. Also, it is been unveiled that gas flaring and other petroleum product activities is capable of melting the glaciers and the poles. Thus, in the maritime and shipping industry, these gases cause great limitations. This result to extreme climatic conditions, expulsion of plants, both land and aquatic animals, wild spread of diseases in the environment, etc, [2]. Established findings confirm that O&G exploration activities contribute enormously to global warming through the overloading of carbon-monoxide (CO), carbon (iv) oxide (CO_2), soot, etc in the atmosphere. The excess burning of the black gold products also pose irreversible risk to the environment and causes serious epidemic of different negative influence [3], [4].

Globally, legislation on gas flaring is increasingly intensified to reduce flaring activities to minimum and set up regulatory standard but this also, is not a permanent solution to flaring. Conversely, this research paper is targeted to proffer solutions and optimize the O&G facilities in order to reduce flaring activities by means of redesigning some vital components of the said plant(s). Although, in the redesign consideration of the component(s), of the production plant, flare systems are considered because it forms very critical safety aspect of the plant for continuous pilot gas flaring and non-wasted gas release due to plant upset emergencies.

2. GBARAIN-UBIE INTEGRATED OIL AND GAS PRODUCTION PLANT

The Gbarain-Ubie Integrated O&G Production Plant is a sample facility used for this study. The facility covers an area approximately 650Km^2 in Bayelsa State. The major objective of this facility is to interconnect flow lines and pipeline of O&G from oil wells in remote manifold locations amongst other activities. It has a central processing facility which collects Non-Associated Gas (NAG) from the outlying areas and separates the gas from condensate and associated water. The dried and metered gas is exported to Bonny for economic purposes. Figure 1 shows a typical block flow diagram of the Gbarain-Ubie Integrated O&G Production Facility.

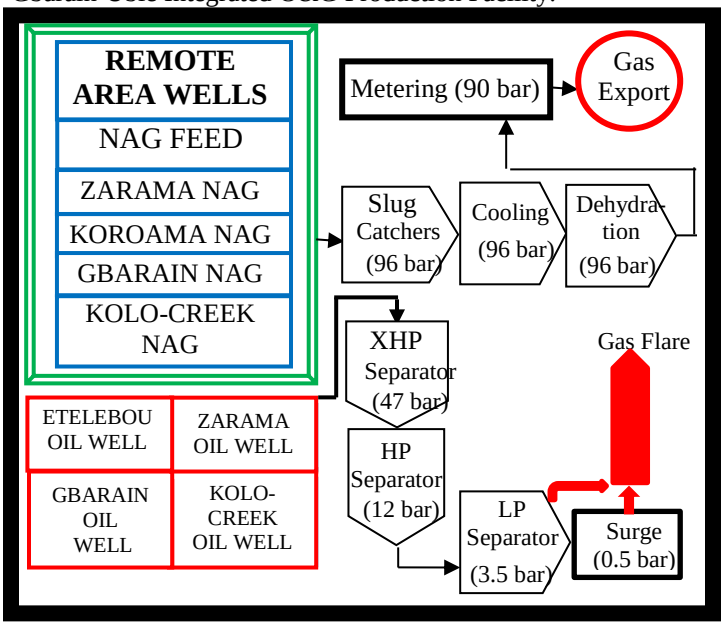

Figure 1: Block Flow Diagram of the Gbarain-Ubie Integrated O&G Production Facility

The production facility from the low pressure (LP) separator and surge vessels of 3.5bar and 0.5 bar respectively

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are flared to the atmosphere which is the concern of this research article. This very low wasted and flared pressures from the corresponding vessels can be boosted up to the required amount of pressure and be exported along the NAGs through a simple optimization process of compressor stages. There are many exceptional scholarly contributions on the design analysis of engine components such as compressors, etc. It reveals in gas turbine studies that understanding the characteristic dimensions and gas flow angles in compressor stages is based on the numerical analysis of the gas flow path. This permits the inlet and outlet cross-sectional area evaluation and the corresponding number of stages needed in a compressor design [5].

The optimum concern of the design is pressure increment and efficiency per stage of the compressor, and perhaps its aerodynamic stability. This is controlled by the tip leakage flow as point of operation becomes nearer to the surge limit as revealed in the open literature [6], [7], [8]. Also, it is unveiled scholarly that improving the technological concept of compression is a reflection of enhanced capacity per stage of the compressor. This will result to increase in rotor speed, aerodynamic loading through the geometric changes, optimization and improved design methods [9], [10], [11].

3. GEOMETRIC DESCRIPTION AND COMPRESSOR PLACEMENT IN THE PRODUCTION FACILITY

A centrifugal compressor is used to develop a mathematical model for the study as shown in figure 2 from a cut-out section of the production facility in the presentation of figure 1.

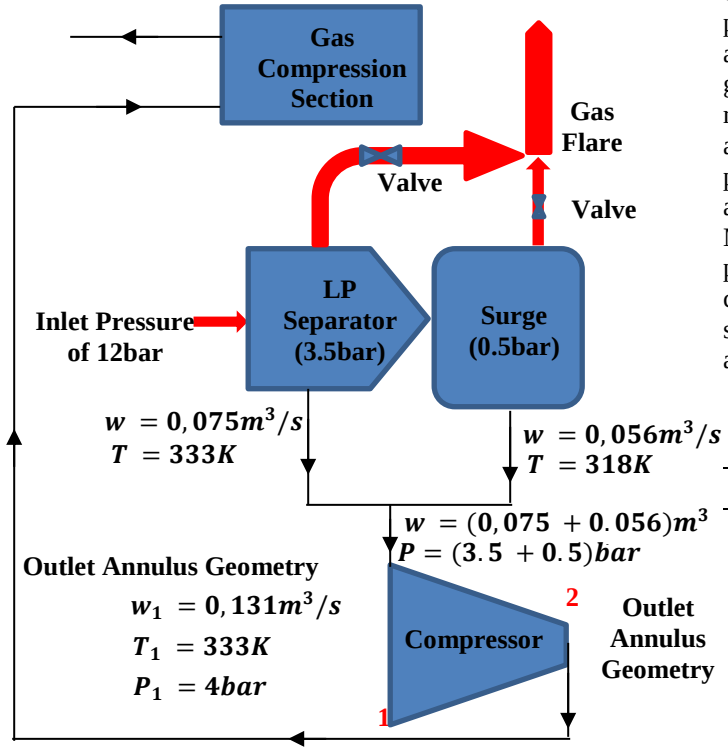
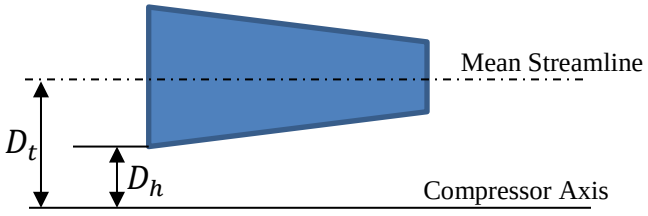


Figure 2: Cut-out Section of the Processing Facility

The developed mathematical model is the pivotal point of the research and it will be used to carry out feasibility studies for the preliminary design analysis of the centrifugal compressor. The essence of this study is to optimize and improve the low-pressure gases of 0.5bar and 3.5bar for the surge and LP separator respectively to a minimum of 12bar for export and reduction of flaring activities.

4. DESIGN POINT SPECIFICATION AND MATHEMATICAL MODELLING ANALYSIS

The cut-out section of the centrifugal compressor as presented in figure 3 is an annulus diagram to establish the compatibility of the design.



D_t = Tip or Casing diameter
D_h = Hub or root diameter

Figure 3: Compressor Annulus Diagram for Constant Mean Diameter

According to a scholarly review, the choice of annulus diagram for a new compressor design depends largely on the mean diameter configuration requirements. This formation is preferably used instead of the rising line configuration of the annulus [9]. It is shown that the rising compressor design for a given mass flow leads to lower outlet blade height and higher in rear stage blade speed. It also leads to higher outlet hub/tip ratio and creates leakage flow in the system which may cause pressure drop [9]. Hence, the choice for constant mean diameter annulus configuration is chosen.

Meanwhile in the compressor design, the component design point parameters are the main focus of the study. These configurations are selected to match the performance specifications of the compressor and are presented in tables 1 and 2.

Table 1: Fixed Design Point Specification Parameters for Compressor

S/N	Parameters	Symbols	Values	Unit
1	Overall Pressure Ratio Compressor	PR	7	
2	Polytropic Efficiency	η_{poly}	88	%
3	Mass Flow Rate	W	0.1314	m^3/s
4	Inlet Pressure	P_1	400 x10^3	Pa
5	Inlet Temperature	T_1	333	K
6	Specific Heat Ratio	γ	1.395	

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Table 2: Assumed Parameters at Design Point

S/N	Parameters	Symbols	Values	Unit
1	Mean Axial Velocity	V_{am}	1	m/s
2	Inlet Hub/Tip Ratio	μ	0.615	
3	Number of Stages		3	
4	Absolute Air Angle at Inlet to each Stage	P_1	0	

Conversely, the numerical analytical process is carried out with the following governing equations as shown in equations 1 – 15.

$$Q_o = \frac{W\sqrt{T_1}}{K_B \times A_1 \times P_1} \quad (1)$$

$$A_1 = \frac{\pi}{4}\left(D_{t_1}\right)^2 \times (1-\mu^2) \quad (2)$$

$$\mu = \frac{D_h}{D_t} \quad (3)$$

$$D_m = \frac{D_h + D_t}{2} \quad (4)$$

$$N = \frac{U_m}{\pi D_m} \quad (5)$$

$$\eta_{compr.} = \frac{(PR_C)^{\frac{\gamma-1}{\gamma}} - 1}{(PR_C)^{\frac{\gamma-1}{\gamma}} + \eta_{poly} - 1} \times 100 \quad (6)$$

$$P_2 = PR_c \times P_1 \quad (7)$$

$$T_2 = T_1 \times \Delta T \quad (8)$$

$$\Delta T = \frac{T_1}{\eta_{compr}} \times (PR_c)^{\frac{\gamma-1}{\gamma}} - 1 \quad (9)$$

$$\Delta T_{stage} = \frac{\Delta T}{required\ number\ of\ stages} \quad (10)$$

$$T_{stage\,1} = \Delta T_{stage} + T_1 \quad (11)$$

$$T_{stage\,2} = \Delta T_{stage} + T_{stage\,1} \quad (12)$$

$$PR_{stage} = (\eta_{stage} \times \frac{\Delta T_{stage}}{T_{1,\ 2,\ 3,\ldots\ldots\infty}} + 1)^{\frac{\gamma-1}{\gamma}} \quad (13)$$

$$P_{stage\,1} = PR_{stage} + P_1 \quad (14)$$

$$P_{stage\,2} = PR_{stage} + P_{stage\,1} \quad (15)$$

The stage analysis of the compressor design is vital and thus, the following thermodynamics equations (ie, equations 16 – 18) are used for the evaluation of the performance efficiency of the compressor.

$$C_P T_{Inlet} = C_P t_{Inlet} + \frac{V^2_{Inlet}}{2} \quad (16)$$

$$C_P T_{Outlet} = C_P t_{Outlet} + \frac{V^2_{Outlet}}{2} \quad (17)$$

$$C_P(T_{Inlet} - T_{Outlet}) = C_P \Delta T = \Delta H \quad (18)$$

Where,
$C_P t =$ specific enthalpy,
$\frac{V^2}{2}$ = Kinetic energy per mass mass,
ΔH = stagnation enthalpy

Also, momentum change of air in the direction of rotation in the stage design analysis of the compressor is considered with the presentation of the stage velocity triangles in figure 4 with its corresponding governing equations in equation 19 – 22.

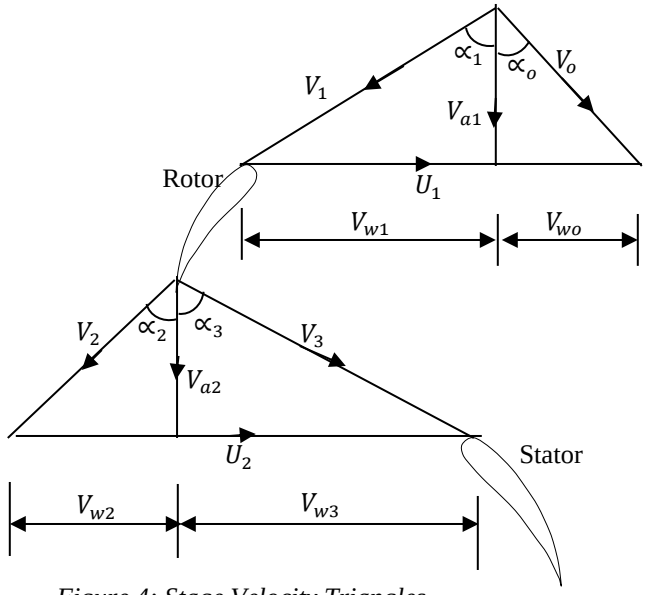

Figure 4: Stage Velocity Triangles

Change in angular momentum $= \delta m(r_2 V_{w3} - r_1 V_{wo})$ (19)

$$Torque, T = \frac{\delta m}{\delta t}(r_2 V_{w3} - r_1 V_{wo}) \quad (20)$$

Work done per unit mass is given as;

$$\frac{W}{\frac{\delta m}{\delta t}} = (U_2 V_{w3} - U_1 V_{wo}) \quad (21)$$

Thus, Euler's equation for compressor stage design is given as;

$$\Delta H_{stage} = (U_2 V_{w3} - U_1 V_{wo}) \quad (22)$$

Finally, the compressor operational envelop at all conditions are evaluated with the off-design variation of the flow path parameters of the compressor as shown in equation 23.

$$\%\ deviation = \frac{DP - ODP}{DP} \times 100\% \quad (23)$$

Where
DP = design parameters

ODP = off design parameters

The achievement of this process lies on the use of the supporting and governing equations with the considerations made in tables 1 and 2. The axial blockage factor, K_B of the compressor inlet is specified as constant unit of 1. Thus, in similar manner, an assumed blade speed at the inlet is considered to be 1m/s mean blade speed. This enhances the calculation of the rotational speed and other vital performance parameters such as polytropic efficiency, stage temperatures and pressures, etc. to validate the research findings.

5. RESULTS PRESENTATION AND DISCUSSION

The following results are achieved from the mathematical modelling analysis of the compressor and it is presented tables 3 and 4; and figures 5 – 8 respectively.

Table 3: Compressor Annulus Design Results

Number of Stages	Stage Pressure (bar)	Stage Temperature (K)	Stage Pressure Ratio
0	4	333	-
1	6.617	390.966	1.321
2	9.677	448.931	1.357
3	13.577	506.897	1.403
4	18.425	564.862	1.464
5	24.329	622.828	1.654

Table 4: Increase in Pressure Ratio Variation

Parameters	Deviation			
	1%	2%	3%	4%
P_2 (bar)	28.28	28.56	28.84	29.12
T_2(K)	624.83	626.81	628.78	630.74
$P_{stage\ 1}$ (bar)	6.64	6.66	6.68	6.70
$T_{stage\ 1}$(K)	391.37	391.76	392.16	392.55
$P_{stage\ 2}$ (bar)	9.73	9.77	9.82	9.87
$T_{stage\ 2}$(K)	449.73	450.52	451.31	452.10
$P_{stage\ 3}$ (bar)	13.67	13.75	13.84	13.93
$T_{stage\ 3}$(K)	508.10	509.29	510.47	511.64
$P_{stage\ 4}$ (bar)	18.57	18.71	18.85	18.99
$T_{stage\ 4}$(K)	566.46	568.05	569.62	571.19
$P_{stage\ 5}$ (bar)	24.54	24.75	24.97	25.18
$T_{stage\ 5}$(K)	624.83	626.81	628.78	630.74
Compr. Eff(%)	84.37	84.39	84.41	84.43

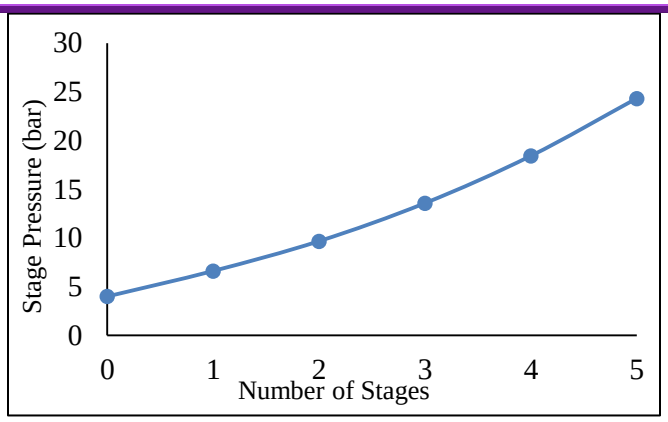

Figure 5: Number of Stage against Stage Pressure

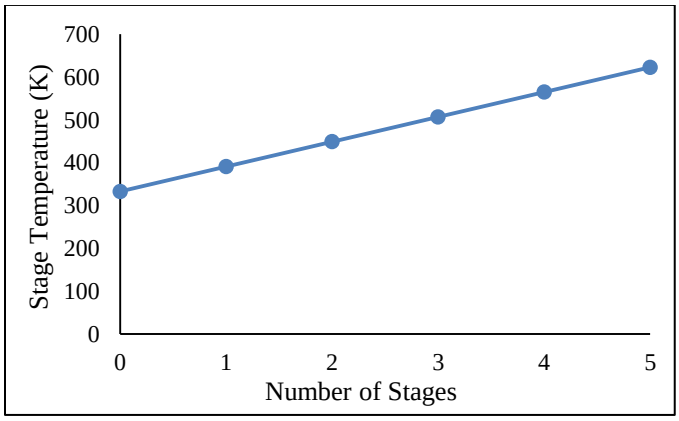

Figure 6: Number of Stage against Stage Temperature

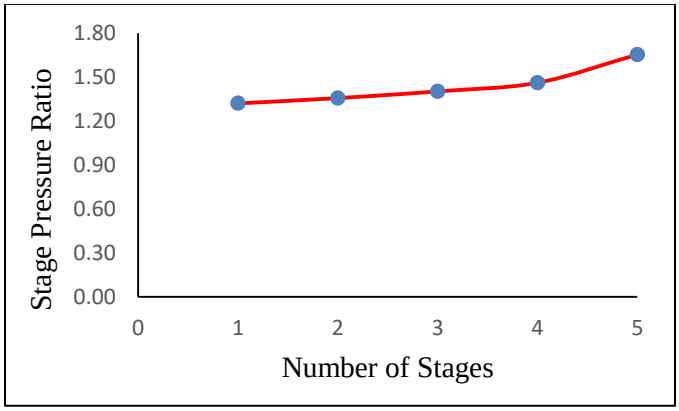

Figure 7: Number of Stage against Stage Pressure Ratio

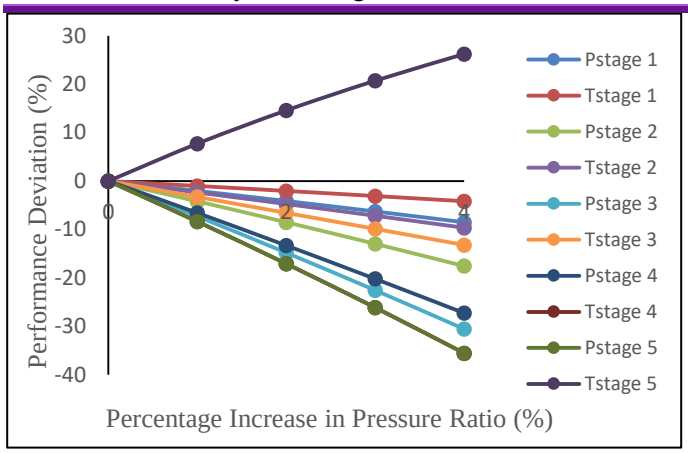

Figure 8: Effect of Increased Pressure Ratio on the Designed Compressor

The graphical results in figures 5 – 7 displays the same trend of plot. Meanwhile, results in figures 5 and 6 shows an increase in stage pressures and temperatures with increasing stage number. Obviously, at stage number three (3), a pressure of 13.577bar was achieved against the required pressure of 12bar. This fulfills the target of the research which is aimed at designing a compressor for a typical O&G production facility to compress its surge and LP separator gases of pressures 0.5bar and 3.5bar respectively to exportable pressure of 12bar. It also has solved the gas flaring devastation and catastrophe to the environment caused by multinational O&G industries. Most especially the research stands to improve the nation's economy from the waste flared gas to source-able economic value to the country's treasury.

However, figures 7 and 8 shows rise in pressure ratio as the number of the design stages in the compressor increases. This is in conformity with a reviewed literature attesting that an axial compressor with the requirement of a large overall pressure ratio will need many stages of blade rows. This enables each stage of the compressor to interact with both up and downstream flows [12], [13]. Meanwhile, in practice gas compressibility effects on a multi-stage compressor ensures steady and constant axial velocity and blade speed at the same rate for a given stage. Consequently, compressor operating at this speed have their front stages approach stall condition whilst rear stages approach choke. However, a rapidly accelerating constant speed compressor have their rear stages tending to approach stall rather than choke. This stalling phenomenon of flow separation from suction surface in aerodynamics is said to increase component performance as reported in a scholarly review [14].

The compressor component performance is a function of stage by stage increase of flow path parameters as such as the pressures and temperatures as observed in table 3. This result has various ranges of increment; at stage 3, it rises from 13.67bar – 13.93bar which is considered as the acceptable pressure range of gas to be exported rather than flaring. Also, in the performance deviation analysis carried out and shown in figure 8, the compressor efficiency also doubles from 84.37% - 84.43%.

6. CONCLUSION

The research work investigated the potentials of increasing low pressure gases that are released to the atmosphere by means of flaring. The flared harmful gases could be converted to useful economic resources to boost the nature economy. The time and risk for engineering design, building and construction of gas flare stock in O&G production facilities can be averted with due improvement and consideration of this research study.

Therefore, O&G production facilities and other likely agencies are to improve on their facilities to incorporate a multi-stage compressor. This will enable to pressurize low pressure gases to meaningful, useful and more economic valued gases for the safety and evergreen protection of the environment.

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