

CRANFIELD UNIVERSITY

AZIAKA, DUABARI SILAS

TECHNO-ECONOMIC STUDY FOR DEGRADED GAS TURBINE
ON PIPELINE APPLICATION IN THE OIL AND GAS INDUSTRY

SCHOOL OF AEROSPACE, TRANSPORT & MANUFACTURING
Centre for Power and Propulsion

DOCTOR OF PHILOSOPHY
Academic Year: 2018 -2019

Supervisors: Dr Igie Uyioghosa and Professor. Pericles Pilidis
November 2019

CRANFIELD UNIVERSITY

SCHOOL OF AEROSPACE, TRANSPORT & MANUFACTURING
Centre for Power and Propulsion

DOCTOR OF PHILOSOPHY

Academic Year 2018 - 2019

AZIKA, DUABARI SILAS

TECHNO-ECONOMIC STUDY FOR DEGRADED GAS TURBINE
ON PIPELINE APPLICATION IN THE OIL AND GAS INDUSTRY

Supervisors: DR. Igie Uyioghosa & Professor. Pericles Pilidis.
November 2019

© Cranfield University 2019. All rights reserved. No part of this
publication may be reproduced without the written permission of the
copyright owner.

ABSTRACT

Gas compression through pipelines is a capital intensive project. Therefore, it is imperative to investigate the viability of investing in such a project. Thus, the techno-economic and environmental risk assessment (TERA) tool to rapidly evaluate the entire natural gas pipeline project becomes vital. This research has investigated the impacts of gas turbine (GT) degradation in the application of TERA for a natural gas pipeline, taking into account the equipment selection, ambient conditions and periodic engine overhaul. Three scenarios (optimistic, medium and pessimistic) defining different levels of deterioration of the GT in comparison with the clean condition were examined in each season of the years (rainy, dry and hot season) based on the location of Trans-Saharan gas pipeline with 18 compression stations. The developed TERA model considered different modules such as the pipeline/gas compressor, performance, emission, a simplified lifing and economic module.

The pipeline/gas compressor module evaluated the performance of the 4180km pipeline and gas compressor power across all compression stations in both isothermal and non-isothermal conditions. Aspen-Hysys/micro-soft excel and MATLAB were used to develop the model. The result showed that for every 1% increase in pipe exit pressure resulted in a 1.8% increase in the volume of the gas flow in the pipeline. Having evaluated the gas compressor (GC) power across the 18 compressor station, the investigation also revealed that for every 1% rise in the gas temperature resulted in a 3.4% rise in the power required by the gas compressor to move the gas. The GT performance was modelled using TURBOAMATCH at fixed power of the engine with respect to the different scenarios under investigation. The performance result was linked with the developed emission, lifing and economic model in MATLAB. The result revealed that for every 1% degradation (reduction in flow capacity and isentropic efficiency) at a constant power of engine operation, between an ambient temperature of 16.2°C and 29°C, CO₂ emission increases between 0.71% and 0.78% when compared with the clean condition. Also, at the same operating condition, the NO_x emission increases between 1.66% and 1.8%. However, NO_x emission at different compressor station varies from one station to another due to the influence of different ambient conditions, engine power settings and number of engines used. Lifing result showed that as the engine degrades, its creep life reduces at high TET to deliver the same power at a fixed number of engines

Net present value (NPV) at different discount rates (DR) (0%, 5%, 10% and 15%) were used to evaluate the economic viability of the project, taking into account engine

divestment and leasing for the redundant fleets after overhaul. The study further investigated how Rescheduling of GT Overhaul (ROH) from the baseline condition affects the economic viability of the pipeline project. The result showed that implementing the ROH reduces the number of GT used for the optimistic, medium and pessimistic scenarios by 8%, 2% and 4% respectively, for the same number of the compressor station and at the same operating conditions when compared with the baseline condition. The result also showed that running the engine on degraded mode increases the life cycle cost while the NPV reduces as the degradation increases. For instance, at 10% DR, the baseline NPV for the clean, optimistic, medium, and pessimistic scenarios were \$21.5, \$19.6, \$18.4 and \$17.1 billion, respectively showing that the NPV decreases with increase in degradation, unlike other studies that analysed the NPV on clean engine operation only. Remarkably, the NPV for engine divestment was 0.2% to 20.3% lower than the NPV for leasing depending on the different scenarios and DR, indicating that NPV leasing gives better benefits than that of engine divestment.

Furthermore, the implementation of on-line compressor washing to investigate the impacts on the pipeline project and emission reduction using TURBOMATCH and MATLAB for the developed model revealed that the CO₂ emission and cost of CO₂ for the optimistic, medium and pessimistic scenarios had a reduction of 5.8%, 6.1% and 6.5% respectively when compared with the baseline condition. Also, at 5% DR, the NPV for the three scenarios after compressor washing increase by 6%, 5.2% and 4.8%, respectively when compared with the baseline case. The proposed methods and result in this research will offer a useful decision-making guide for all pipeline investors to invest in a natural gas pipeline business, taking into account different operating conditions and the impacts of engine degradation.

Keywords: Performance, MATLAB, lifecycle cost, gas compressor, ambient temperature, TURBOMATCH, TERA. Emission

ACKNOWLEDGEMENTS

This research work began from 2016 to 2019 with both good and challenging times. I am most grateful to the Almighty God for giving me Prof. Pericles Pilidis and Dr Uyioghosa Igie who were not just mere supervisors, but they were always there to help and guide at all the time. I give them my profound gratitude for their meticulous supervision in driving this research to its logical conclusion. I thank Dr Li Yiguang and Dr Kim Blackburn for their patience and time spared to examine all the three years of my progress review reports, and for their helpful comments. In the same regard, I must also appreciate both the external and internal examiners, who would finally, examine this thesis for their time, patience and comments. Similarly, I thank all staff in the centre of Power and Propulsion for their numerous contribution and advice towards the successful end of this research work.

I am very grateful to the Tertiary Education Trust Fund (TETFUND) for granting me the scholarship to study a PhD degree in the United Kingdom. This was made possible through Rivers State Polytechnic, Bori. The Rector Dr.Georgewill, Dr Tamuno J.K, Engr. Zukbee N. A and other academics and staff of this institution are acknowledged for their various contribution towards my scholarship.

I sincerely appreciate the gracious input of Dr Nasiru Tukur, Dr Musa Gali, Prof. Jacob Neeka, Dr David Rowland, Dr Bernard Lewis, Engr. Le-ol A, Dr. Kingsley Ibrahim and Dr Obhuo Mafel. The contributions of these guys towards the successful completion of my PhD are unquantifiable. I also thank my colleagues and friend Dr Emmabuel Osigwe, Dr Alozie Okechukwu, Dr Ojo, Dr Israel, Dr Saleh M.M, Dr Luz-Dary L, Dr Ezedem, Dr Evangelia for their encouragement and support and Ass. Prof. Nasir Abdulkarim

A special thanks go to the D.L.C.F Cranfield university, and the brethren of Deeper Life Bible Church, Buckinghamshire Region, Pastor Jonah Sunday and Pst. (Dr) M.D Nsanam for their encouragement and prayers. A very distinctive thanks to my amiable wife, daughter and extended family for their support and prayers. Surely, they must reap the fruits of their labour. Above all, I give thanks to the Almighty God the giver of wisdom and understanding for the successful end of my study

TABLE OF CONTENTS

ABSTRACT	i
ACKNOWLEDGEMENTS.....	iii
LIST OF FIGURES	x
LIST OF TABLES	xviii
LIST OF EQUATIONS	xix
LIST OF ABBREVIATIONS	xxii
1 INTRODUCTION	1
1.1 Background of Study.....	1
1.1.1 Identification of Research Gap and Contribution to Knowledge.....	4
1.2 Motivation and Problem Statement	5
1.3 Aim of Research	7
1.4 Research Objectives.....	7
1.5 Research Contribution	8
1.5.1 The results on the impact of engine degradation on the techno-economic assessment of pipeline investment	8
1.5.2 Development of a model and methodology to evaluate the economic implication of using a degraded gas turbine in a time-based ambient temperature for natural gas pipeline investment.....	9
1.5.3 Development of an emission model and methodology to quantify gas turbine emission and cost of CO ₂ in a timed-based system.....	10
1.6 Scope of Research and Limitation	10
1.7 Publications and Conferences during PhD.....	12
1.7.1 Publications.....	12
1.7.2 Conferences.....	12
1.8 Thesis Structure.....	13
1.9 Summary of Chapter	15
2 LITERATURE REVIEW	17
2.1 Background.....	17
2.1.1 Natural Gas Reserve and Productions by Countries	17
2.2 Natural Gas Transportation System	18
2.2.1 Natural Gas Pipeline Transportation System.....	18
2.3 Pipeline Design Consideration	21
2.3.1 Analysis of Gas Flow.....	22
2.4 Pipeline Components/Accessories.....	28
2.4.1 Meter Stations and Regulators (MSR).....	28
2.4.2 Mainline Valve Stations	28
2.4.3 Supervisory Control and Data Acquisition (SCADA) and Telecommunication System	28
2.4.4 Environmental and Permitting Cost.....	29
2.4.5 Right of Way (ROW) Acquisition	29
2.4.6 Engineering and Construction Management (ECM)	29
2.5 Trans-Saharan Gas Pipeline(TSGP) Project.....	30

2.6 Compressor Stations.....	31
2.6.1 Arrangement of Compressor unit on Natural Gas Pipeline	33
2.6.2 Thermodynamics of Gas Compression	38
2.6.3 Compressor Station's Economics.....	39
2.7 Prime Mover Option for Compressor Station.....	40
2.8 Gas Turbine	41
2.8.1 Gas Turbine Degradation	42
2.8.2 Degradation of Turbine Blades.....	45
2.8.3 Gas turbine Degradation and Evaluation Methods	47
2.9 Engine Maintenance Action: Compressor Washing	48
2.9.1 Manual Procedure	48
2.9.2 Abrasive of Grit-blasting methods	49
2.9.3 Off-line and On-line Compressor Washing	49
2.10 Gas turbine (GT) Engine Overhaul.....	53
2.11 Gas Turbine Emission.....	55
2.11.1 Correlation and empirical modelling of emission	56
2.11.2 Considered Emission for the Present Study and the Correlation between the Emission	57
2.12 TERA Tool	59
2.12.1 TERA Application and progress beyond state-of-the-art.....	60
2.13 Economic Appraisal Methods.....	62
2.13.1 Net Present Value (NPV)	63
2.14 Summary of Chapter	65
3 RESEARCH METHODOLOGY.....	67
3.1 Overview	67
3.2 Pipeline and Gas Compressor (GC) Module	67
3.2.1 Introduction	67
3.3 Pipeline performance Calculation.....	69
3.3.1 Methodology.....	70
3.3.2 Local Condition	70
3.3.3 Pipeline analysis	71
3.4 Result and discussion	75
3.4.1 Effects of Pipe Exit Pressure on Volume of Gas flow at different gas Temperature	76
3.5 Evaluation of Gas compressor	78
3.5.1 Overview	78
3.5.2 Evaluation of Gas compressor Pressure Ratio at Various Compression stations.....	79
3.5.3 Gas Properties and GC power evaluation using Aspen-Hysis.....	80
3.5.4 GC power calculation	81
3.5.5 Polytropic Head Calculation	84
3.5.6 Evaluation of Volumetric Flow	84
3.5.7 Gas Compressor Operating Characteristics	85
3.5.8 Discussion.....	87

3.5.9 Effects of Suction Pressure and Ambient condition on Gas compressor Power	89
3.5.10 Gas Compressor Highest Power for 18 Compressor Station.....	92
3.5.11 Centrifugal Compressor map on Pipeline Compression System	93
3.5.12 Effects of molecular weight and Gas temperatures on Gas compressor performance	95
3.6 TERA framework for the natural gas pipeline application	98
3.6.1 Engine Performance Module	99
3.6.2 Lifting Module	100
3.6.3 Economic Module (EM)	102
3.7 Conclusion	103
4 Gas turbine Selection, Modelling and Performance Simulation	105
4.1 Overview	105
4.2 Introduction	105
4.3 Gas turbine Selection	106
4.4 Performance Modelling	107
4.4.1 Component Map.....	108
4.4.2 Map scaling and Deteriorated mode of the engines	109
4.5 IGTA Performance Specification	110
4.6 IGTA OD Simulation Result	112
4.6.1 Effect of Ambient Temperature and Degradation	112
4.7 IGTB Performance Specification	120
4.7.1 IGTB Compressor Map	122
4.7.2 The Impact of Degradation on the Deviation of engine Parameters from their DP	124
4.8 IGTD Performance Specification	126
4.9 Conclusion	128
5 Influence of Degradation on GT usage and emission evaluation	131
5.1 Overview	131
5.2 The Influence of Degradation in determining the number of Gas Turbine usage on the Pipeline application	131
5.3 Introduction	131
5.3.1 Implementation of Routine Overhaul on the GT Engine	132
5.4 Methodology	133
5.4.1 Estimation of Number of GTs to be used for each CS.....	134
5.4.2 Engine Degraded Scenarios under Investigation.....	137
5.4.3 Consideration and Summary of Implanted degradation in each Season.....	138
5.4.4 Gas turbine placement to each Compressor station	140
5.5 Result and Discussion.....	143
5.5.1 Turbine Entry Temperature (TET) Profile Map (Conjugate Map)	143
5.5.2 Analysis of the Compressor Station TET profile based on Season's of the Year	144

5.5.3 Number of GT used Per Station with respect to Different Season of the Year	151
5.5.4 Number of Engine investment for Project Life due to Degradation	153
5.6 Conclusion	155
5.7 Gas turbine Emission Evaluation on Natural Gas Pipeline	156
5.7.1 Introduction	156
5.8 Methodology	157
5.8.1 NO _x Emission Calculation	159
5.8.2 CO ₂ Emission Calculation	159
5.9 Result and Discussion.....	161
5.9.1 Emissions produced at Each Compressor Station.....	166
5.9.2 Evaluation of emission Tax.....	169
5.9.3 Conclusion	172
6 Development of Economic Model of the Gas Pipeline	175
6.1 Cost Components of the compressor stations under consideration	176
6.1.1 Economic Sensitivity Analysis	177
6.2 Methodology	177
6.3 Component Cost	179
6.3.1 Capital Expenditure (CAPEX) Calculation Method	179
6.3.2 Method to Estimate Fuel and O&M cost	183
6.3.3 Operation and Maintenance Costs (O&M).....	184
6.3.4 Fuel Cost Calculation	185
6.3.5 Economic Sensitivity Analysis	186
6.4 Result and Discussion.....	186
6.4.1 The impacts of GT Degradation on the Project CAPEX.....	186
6.4.2 Impact of GT Degradation on Fuel Cost	190
6.4.3 Season's-based Fuel Cost	191
6.4.4 Fuel Cost at each Compressor Station	194
6.4.5 Creep Life Estimation	196
6.4.6 Maintenance Cost	198
6.4.7 Influence of Degradation on Different Component of O&M Cost	198
6.4.8 Maintenance Costs of Each Compressor Station	203
6.4.9 Life Cycle Costs (LCC) for the Pipeline Project	208
6.4.10 The impact of Escalation Rate of Fuel and O&M cost on LCC	210
6.5 Evaluation of Net Present Value	213
6.5.1 Revenue from the sale of gas.....	213
6.5.2 Loan Repayment.....	214
6.5.3 Annual Cash Flow of the entire Project	214
6.5.4 Engine divestment sales	215
6.5.5 Net Present Value (NPV) Calculation	218
6.5.6 The Influence of Engine Leasing on NPV	221
6.6 Conclusion	225
7 MAINTAINANCE ACTION	227
7.1 General Overview	227

7.2 The Impacts of GT Re-scheduled overhaul on the Techno-economic Assessment of Natural Gas Pipeline Project	227
7.2.1 Overview of section one	227
7.2.2 Introduction	228
7.3 Methodology	228
7.3.1 Consideration of the Implanted Degradation base on ROH	229
7.4 Consideration of the Number of GT used for the Re-scheduled Overhaul ..	232
7.4.1 Result and Discussion	232
7.5 The Influence of Re-schedule Overhaul on the Economics	236
7.6 Result and Discussion	236
7.6.1 The Influence of Re-schedule overhaul on the CAPEX	237
7.6.2 The Influence of re-scheduled overhaul on the Fuel and O&M costs of the Pipeline Project	238
7.7 The Impacts of ROH on GT Emissions	241
7.7.1 The Influence of ROH on the Life Cycle Cost (LCC)	243
7.7.2 The Impacts of Reschedule Overhaul on NPV	244
7.8 Compressor Washing: Maintenance Action –Section 2	248
7.8.1 Overview	248
7.8.2 Methodology	249
7.8.3 Performance Improvement through compressor washing	250
7.8.4 Economic Enhancement after Compressor Washing	253
7.8.5 Emission reduction through compressor washing	260
7.8.6 The Influence of Compressor Washing on the NPV	264
7.8.7 Result and Discussion	266
8 CONCLUSIONS AND RECOMMENDATION FOR FUTURE WORKS	269
8.1 Conclusions	269
8.2 RECOMMENDATION	275
REFERENCES	277
APPENDICES	291
Appendix A	291
Appendix B Compressor map from the OEM	Error! Bookmark not defined.
Appendix C : TURBOMATCH design point input file	295
Appendix D OD results for IGTB	303
Appendix E : Input file for the Degraded Scenario	305
Appendix F : Selected compressor based on GC calculated power	315
Appendix G Cost components of the O&M cost	316
Appendix H The implemented percentage value for the re-schedule overhaul ..	318
Appendix I Emissions and CO ₂ tax for CS 4	319

LIST OF FIGURES

Figure 1-1 Natural Gas Demand and Supply for EU [6]	2
Figure 1-2 Layout of Pipeline with Compressor stations[9]	3
Figure 2-1 Natural gas reserve by countries[28]	18
Figure 2-2 Transportation efficiency for various transportation mode [29]	19
Figure 2-3 Pipeline layout[32]	20
Figure 2-4 Energy of flowing fluid[38]	23
Figure 2-5 Steady flow in gas pipeline	24
Figure 2-6 Trans-Saharan Gas Pipeline route[26]	30
Figure 2-7 Schematics of GT compressor station[47]	32
Figure 2-8 Generalised compressor station schematic[44]	33
Figure 2-9 Schematic diagram of a Compressor Station [48].....	33
Figure 2-10 Parallel Configuration of Compressors[10]	35
Figure 2-11 Compressors in Series[10]	35
Figure 2-12 Types of compressors [57]	37
Figure 2-13 Range of application of different types of compressor[58]	37
Figure 2-14 Mechanism of Degradation and their effects	44
Figure 2-15 Degraded GT air inlet Bellmouth and blading[98]	45
Figure 2-16 Photographs showing the fractured blade[103]	46
Figure 2-17 Effects of compressor washing [111]	48
Figure 2-18 Manual cleaning of the compressor [113]	49
Figure 2-19 Description of online and off-line washing device[128]	51
Figure 2-20 Water cleaner example [133].....	52
Figure 2-21 Types of Degradation Rate[132].....	52
Figure 2-22 Source of greenhouse gas emission[140].....	56
Figure 2-23 NO _x and CO relationship with combustion temperature [142]	58
Figure 2-24 NO _x , smoke, CO and UHC emission with power setting[138]	58
Figure 2-25 TERA philosophy and framework for aircraft optimisation [144]	59
Figure 2-26 Cash flow activities	64
Figure 3-1 Time-based ambient temperature (rainy Season)[26].....	70

Figure 3-2 Time-based ambient temperature (dry season)[26]	71
Figure 3-3 Time-based ambient temperature (Hot season)[26]	71
Figure 3-4 Methodology for pipeline module.....	72
Figure 3-5 The pipe discharge & GC inlet pressure at the various CS.....	76
Figure 3-6 Relationship between gas flows, Exit pressure and gas temperature in the pipeline	77
Figure 3-7 Effects of varying exit pressure and gas temperature on gas flow in a pipeline	78
Figure 3-8 Effects of constant exit pressure and gas temperature on gas flow in a pipeline	78
Figure 3-9 Gas compressor PR at varying ambient temperature	80
Figure 3-10 Steps in achieving the gas compressor power using Aspen-Hysis	81
Figure 3-11 Nelson-Obert compressibility chart [163]	82
Figure 3-12 Polytropic efficiency & Head coefficient vs flow coefficient	83
Figure 3-13 Gas compressor power at varying ambient temperature (RS)	88
Figure 3-14 Gas compressor power in a varying ambient condition (DS)	88
Figure 3-15 Gas compressor power at varying ambient condition (HS)	89
Figure 3-16 Effect of ambient temperature on GC power (WS)	90
Figure 3-17 Effects of ambient temperature on GC power (DS)	90
Figure 3-18 Effects of ambient temperature on GC power (HS)	90
Figure 3-19 Variation in inlet pressure with GC power (WS).....	91
Figure 3-20 Variation in inlet pressure with GC power (DS).....	91
Figure 3-21 Variation in inlet pressure with GC power (HS).....	91
Figure 3-22 Variation in inlet pressure with constant discharge pressure	92
Figure 3-23 Gas compressor power for three seasons of the year	93
Figure 3-24 Pressure ratio-volume flow curve	94
Figure 3-25 Polytropic efficiency-volume flow curve	94
Figure 3-26 Discharge temperature-volume flow curve	95
Figure 3-27 Effects of gas molecular weight on the polytropic head	96
Figure 3-28 Effects of varying gas temperature on the polytropic head	97
Figure 3-29 Effects of varying gas temperature on GC power at a constant volume flow	98

Figure 3-30 TERA framework for pipeline natural gas investment	99
Figure 4-1 DP power of the selected engines	107
Figure 4-2 Schematic diagram of the IGTA 2-spool engine	110
Figure 4-3 IGTA Compressor map (PR vs CM)	112
Figure 4-4 Effect of ambient temperature and degradation on shaft power	114
Figure 4-5 Effect of ambient temperature and degradation on thermal Efficiency	115
Figure 4-6 Effect of ambient temperature and degradation on fuel flow.....	115
Figure 4-7 Effects of Altitude on shaft power	117
Figure 4-8 Effect of Altitude on thermal efficiency.....	117
Figure 4-9 Effects of Altitude on fuel flow.....	118
Figure 4-10 Effects of compressor degradation on performance parameters	119
Figure 4-11 Effects of turbine degradation on performance parameters	120
Figure 4-12 Effect of "compressor and turbine" degradation on performance parameters	120
Figure 4-13 Schematic diagram of IGTB 3-spool engine	121
Figure 4-14 IGTB LP Compressor map (PR vs CMF).....	123
Figure 4-15 IGTB HP Compressor map (PR vs CMF)	123
Figure 4-16 Effects of Compressor degradation on engine parameters.....	125
Figure 4-17 Effects of turbine degradation on engine parameters	125
Figure 4-18 Effects of both "compressor and turbine" degradation on engine parameters	125
Figure 4-19 Schematic diagram of IGTD 3-spool.....	126
Figure 4-20 IGTD LP compressor map	128
Figure 4-21 IGTD HP Compressor map.....	128
Figure 5-1 Methodology to determine the number of GT at various CS	133
Figure 5-2 Constant GT power with increasing degradation with time	135
Figure 5-3 Implanted level of degradation for the winter season.....	139
Figure 5-4 Implanted level of degradation for dry season	139
Figure 5-5 Implanted level of degradation for the hot season	139
Figure 5-6 TET Profile for the pessimistic scenario without TET limit	144
Figure 5-7 TET Profile map for optimistic scenario (Rainy season)	146

Figure 5-8 TET Profile map for medium scenario (Rainy season)	147
Figure 5-9 TET Profile map for pessimistic scenario (Rainy Season)	147
Figure 5-10 TET Profile map for optimistic scenario (dry season)	148
Figure 5-11 TET Profile map for medium scenario (Dry season)	148
Figure 5-12 TET Profile map for pessimistic scenario (dry season)	149
Figure 5-13 TET Profile map for optimistic scenario (Hot season)	150
Figure 5-14 TET Profile map for medium scenario (Hot season)	150
Figure 5-15 TET Profile map for pessimistic scenario (Hot season)	150
Figure 5-16 Number of GTs used for project life at different scenarios	154
Figure 5-17 Number of GTs used for 18 CS at various scenarios	154
Figure 5-18 Number of GTs used at various scenarios for project life	155
Figure 5-19 Number of different mixed fleets used at various scenarios	155
Figure 5-20 Emission calculation method	158
Figure 5-21 Annual CO ₂ emission for the winter season	163
Figure 5-22 Annual NO _x emission for the winter season	163
Figure 5-23 The effect of lower power settings on TET	164
Figure 5-24 Annual CO ₂ for all CS and seasons of project life	165
Figure 5-25 Annual NO _x for all CS and seasons of project life	166
Figure 5-26 Annual CO ₂ emission for each CS throughout the project life	167
Figure 5-27 Annual NO _x emission for each CS throughout the project life	167
Figure 5-28 Integrated CO ₂ generated for project life	168
Figure 5-29 Integrated NO _x for Project life	169
Figure 5-30 Annual emission tax for the winter season	170
Figure 5-31 Annual emission tax for dry season	170
Figure 5-32 Annual emission tax for the hot season	170
Figure 5-33 Annual cost of CO₂ emission for the project life	171
Figure 5-34 Annual cost of CO ₂ emission for each CS for the project life	171
Figure 5-35 Total costs of CO ₂ emission for the project life	172
Figure 6-1 Economic Model for pipeline investment	178
Figure 6-2 Calculation of the capital expenditure (CAPEX)	179

Figure 6-3 Price of gas turbine[64].....	182
Figure 6-4 Method of calculating Fuel and O&M costs	183
Figure 6-5 Cost of GT engines used for each compressor station	187
Figure 6-6 Capital cost of GT engines at various scenarios for 18 CS.....	187
Figure 6-7 Capital cost of GC in each compressor station	188
Figure 6-8 Cost of various pipeline accessories	189
Figure 6-9 Capital cost of all pipeline components for 18 CS.....	190
Figure 6-10 Total CAPEX for the pipeline project-Baseline case	190
Figure 6-11 Fuel cost for the rainy season.....	192
Figure 6-12 Fuel cost for dry season	192
Figure 6-13 Fuel cost for the hot season	193
Figure 6-14 Annual fuel cost at different scenarios	193
Figure 6-15 Fuel cost for each CS in rainy season for 30 years	195
Figure 6-16 Fuel cost for each CS in the dry season for 30 years	195
Figure 6-17 Fuel cost for each CS in the hot season for 30 years	195
Figure 6-18 Total fuel cost for each CS in 30 years	196
Figure 6-19 Integrated fuel cost for each scenario.....	196
Figure 6-20 Effects of degradation on the engine creep life (rainy season)	198
Figure 6-21 Effects of degradation on engine creep life (dry season)	198
Figure 6-22 Effects of degradation on engine creep life (hot season)	198
Figure 6-23 Variable costs for the rainy season.....	201
Figure 6-24 Major maintenance cost for the rainy season	201
Figure 6-25 O&M costs for the rainy season.....	201
Figure 6-26 Annual O&M cost for dry season	202
Figure 6-27 Annual O&M costs for the hot season	202
Figure 6-28 Annual O&M cost for the project's life-Baseline Case.....	203
Figure 6-29 O&M costs for each CS in the rainy season	205
Figure 6-30 O&M costs for each CS in the dry season	205
Figure 6-31 O&M costs for each CS in the hot season	205
Figure 6-32 O&M costs for each scenario at the various CS for 30 years.....	207

Figure 6-33 Integrated O&M cost for each scenario-Baseline case	207
Figure 6-34 Life cycle cost (LCC) for pipeline investment.....	209
Figure 6-35 Percentage of the baseline project cost at the clean condition	209
Figure 6-36 Integrated LCC for the pipeline investment.....	210
Figure 6-37 Annual escalated fuel cost for project life	211
Figure 6-38 Total escalated fuel cost for the life	211
Figure 6-39 Annual escalated O&M cost for the project life	212
Figure 6-40 Total escalated O&M cost for the project life	212
Figure 6-41 Escalation rate on the life-cycle cost of the project.	213
Figure 6-42 Annual loan repayment on project	214
Figure 6-43 Annual net cash flow on the project.....	215
Figure 6-44 Methods to calculate the engine divestment.....	216
Figure 6-45 Divestment sale due to overhaul	217
Figure 6-46 Total benefits from engine divestment.....	218
Figure 6-47 Method to evaluate the NPV	219
Figure 6-48 The influence of degradation and DR on NPV	220
Figure 6-49 Methodology of leasing calculation	221
Figure 6-50 Engine leasing and return from service	223
Figure 6-51 Revenue from Engine leasing and year 30 divestment.....	224
Figure 6-52 Total revenue from engine leasing and divestment at year 30.....	224
Figure 6-53 The influence of engine leasing on NPV	225
Figure 7-1 Methodology for the re-scheduled overhaul.....	229
Figure 7-2 Implanted level of degradation for the rainy season	231
Figure 7-3 Implanted level of degradation for dry season	231
Figure 7-4 Implanted level of degradation for the hot season	232
Figure 7-5 Time-based opt TET profile for the baseline condition -Rainy season	233
Figure 7-6 Time-based TET profile after ROH- Rainy season	233
Figure 7-7 GT investment for the re-schedule overhaul.....	234
Figure 7-8 Number of engines used in the pessimistic scenario	234
Figure 7-9 Number of GT used for each CS at different scenarios	235

Figure 7-10 GT used for ROH in comparison with the baseline	235
Figure 7-11 Number of each GT used in the mixed fleets for ROH.....	236
Figure 7-12 Cost of GT in comparison with the baseline	237
Figure 7-13 Total CAPEX of the project-ROH.....	237
Figure 7-14 Annual fuel cost for the project life-ROH.....	238
Figure 7-15 Total fuel cost for project life-ROH.....	239
Figure 7-16 Annual O&M cost for the life-ROH.....	240
Figure 7-17 Total O&M costs for ROH in comparison with the baseline	240
Figure 7-18 Annual CO ₂ emission for the project life.....	241
Figure 7-19 Total emission generated for the ROH throughout the project life	242
Figure 7-20 Annual emission tax for ROH throughout the project life	242
Figure 7-21 Total costs of CO ₂ emission for ROH throughout the project life.....	243
Figure 7-22 Life cycle cost-ROH.....	244
Figure 7-23 Annual benefits from divestment	245
Figure 7-24 Benefits from engines divestment –ROH.....	245
Figure 7-25 The Influence of engine divestment on NPV for ROH.....	246
Figure 7-26 Revenue from leasing –ROH.....	247
Figure 7-27 The influence of engine leasing on NPV for ROH.....	248
Figure 7-28 Methodology for compressor washing	250
Figure 7-29 Massflow recovery through CW.....	251
Figure 7-30 TET reduction through CW	252
Figure 7-31 Fuelflow reduction through CW	253
Figure 7-32 Improvement in the PR after CW	253
Figure 7-33 Annual litres of DH ₂ O and CONC used for the washing period.....	255
Figure 7-34 Total litres of DH ₂ O and CONC used for project life	255
Figure 7-35 Total costs of washing	256
Figure 7-36 Annual fuel cost after compressor washing (rainy season).....	257
Figure 7-37 Annual fuel cost after compressor washing (dry season).....	257
Figure 7-38 Annual fuel cost after compressor washing (hot season).....	257
Figure 7-39 Annual fuel cost before compressor washing	258

Figure 7-40 Annual fuel cost after compressor washing	258
Figure 7-41 Total fuel cost after compressor washing with the baseline	259
Figure 7-42 Total O&M cost after compressor washing with the baseline case ...	260
Figure 7-43 Annual CO ₂ emission after compressor washing	261
Figure 7-44 Annual CO ₂ cost after compressor washing	262
Figure 7-45 Annual NO _x emission after compressor washing	262
Figure 7-46 Total CO ₂ emission before and after compressor washing	263
Figure 7-47 Total Cost of CO ₂ emission before and after compressor washing ..	263
Figure 7-48 Total NO _x emission before and after compressor washing.....	264
Figure 7-49 Method for NPV calculation in compressor washing	265
Figure 7-50 Total capital cost of the compressor station components	267
Figure 7-51 NPV before compressor washing	267
Figure 7-52 NPV after compressor washing	267

No table of figures entries found.

LIST OF TABLES

Table 2-1 Summarised Gas flow equation [17]	27
Table 3-1 TSGP design data	69
Table 3-2 Distance and Elevation between pipe segment[18]	74
Table 3-3 First Compressor station design point[161].....	75
Table 3-4 Aspen Hysis and calculated result	85
Table 4-1 Design point parameters of IGTA engine at ISA condition	111
Table 4-2 Performance validation of IGTA.....	111
Table 4-3 Design point parameters of the IGTB model at ISA condition	121
Table 4-4 IGTB Model Validation at DP	122
Table 4-5 Design point parameters of IGTD engine model at ISA condition	126
Table 4-6 Performance validation of IGTD.....	127
Table 5-1 Gas turbine engines attached to each compressor station	142
Table 5-2 Number of GTs used for different season and scenarios for CS-2.....	152
Table 6-1 Pipeline data[6].....	180
Table 6-2 Impact of ER on LCC.....	213
Table 7-1 Economic assumptions for compressor washing	254
Table 7-2 Cost of washing fluid.....	256
Table 7-3 Component of Compressor Station 18.....	266
Table 7-4 Benefits of compressor washing on the NPV	268

LIST OF EQUATIONS

(2-1).....	23
(2-2).....	24
(2-3).....	26
(2-4).....	34
(2-5).....	35
(2-6).....	38
(2-7).....	38
(2-8).....	39
(2-9)	57
(2-10).....	63
(3-1).....	72
(3-2)	72
(3-3)	73
(3-4)	73
(3-5)	73
(3-6)	73
(3-7).....	79
(3-8).....	81
(3-9).....	81
(3-10).....	82
(3-11).....	82
(3-12).....	82
(3-13).....	83
(3-14).....	83
(3-15).....	83
(3-16)	83
(3-17)	84
(3-18).....	84
(3-19).....	84

(3-20).....	84
(3-21).....	85
(3-22).....	86
(3-23).....	86
(3-24).....	86
(3-25).....	86
(3-26).....	86
(3-27).....	86
(3-28).....	87
(3-29).....	87
(4-1)	108
(4-2)	108
(4-3)	109
(5-1).....	135
(5-2).....	136
(5-3).....	136
(5-4).....	159
(5-5).....	160
(5-6).....	160
(5-7).....	160
(5-8).....	160
(5-9).....	160
(5-10).....	161
(5-11).....	161
(6-1).....	180
(6-2).....	181
(6-3)	181
(6-4)	181
(6-5).....	182
(6-6).....	184

(6-7).....	184
(6-8).....	184
(6-9)	185
(6-10)	185
(6-11)	186
(6-12).....	213
6-13	214
(6-14).....	216
(6-15).....	216
(6-16).....	219
(6-17).....	222

LIST OF ABBREVIATIONS

AS	All season	
Bcm	Billion cubic meter	
CO ₂	Carbon dioxide	
C _o	Capital Cost	
CONC	Concentrate (washing fluid)	
CS	Compressor station	
C_p	Specific heat at constant pressure	
C_v	Specific heat at constant volume	
C_f	Net Cash Flow	
CT	Contingencies	
DS	Dry season	
E	Pipeline Efficiency	%
ECM	Engineering construction and management	
E&P	Environmental and permitting	
ER	Escalation rate	
Fac	Fuel air ratio	
γ	Specific Ratio	
G	Gas gravity	
GT	Gas Turbine	
GHGs	Green house gases	
Hp	Polytropic Head	KJ/Kg
HPC	High Pressure Compressor	
HPT	High Pressure Turbine	
HS	Hot season	
L	Length	Km
LCC	Lifecycle Cost	
LPT	Low Pressure Turbine	
Mcost	Maintenance cost	
η_p	Polytropic Efficiency	%
NPV	Net Present Value	
$\dot{m}_c$	Corrected Mass Flow rate	kg/s
MAOP	Maximum Allowable Operating Pressure	
MLV	Main line valve	
MW	Molecular Weight	
M _{air}	Mass flow of air	
NO _x	Nitrogen oxide	
P	Pressure	

PIPC	Pipeline initial project cost	
PL/AC	Pipeline and accessories	
P1	Inlet Pressure	Bar
P2	Outlet Pressure	Bar
Pc	Critical Pressure	Bar
PR	Pressure ratio	
Pr	Reduced Pressure	Bar
P_{ref}	Reference Inlet Pressure (101.325kpa)	
PR_{DP}	Design point pressure ratio-new map	
$PR_{DP Map}$	Design point pressure ratio-new map	
Q	Volume flow rate	m ³ /day
r	Discount rate	
R	Gas Constant	
RS	Rainy season	
R _{ref}	Reference Gas Constant (287.058J/Kg.K)	
SC.1 OPT	Scenario 1 operating point	
SC.2 OPT	Scenario 2 operating point	
SC.3 OPT	Scenario 3 operating point	
SCADA	Supervisory control and data acquisition	
ROH	Reschedule overhaul	
ROI	Return on investment	
ROW	Right of ways	
Tamb	Ambinet temperature	°C
t	Pipeline thickness	mm
T	Absolute Temperature	K
T1	Inlet Temperature	K
T2	Outlet Temperature	K
Tc	Critical Temperature	K
TET	Turbine Entry Temperatures	K
T _f	Gas Temperature	K
Tr	Reduced Temperature	
T_{ref}	Reference inlet temperature (288.15 ⁰ K)	
V	Specific Volume	
VA	Flow Velocity at Point A	m/s
VB	Flow Velocity at Point B	m/s
W	Standard Flow	m ³ /d
WAC_{DP}	mass flow for new map	m ³ /d
$WAC_{DP Map}$	mass flow for OEM map	m ³ /d
Z	Compressibility Factor	
η_{DP}	Design point efficiency for new map	
$\eta_{DP Map}$	Design point efficiency for OEM map	

1 INTRODUCTION

1.1 Background of Study

The rise in world population and the increase in industrialisation has led to an increase in demand for energy to be used in various sectors of life. The population of the world is projected to reach 9.8 billion by the year 2050, which is about a 28% increase from the current global population[1]. The present growth in industrialisation in addition to this world population increase and the projected upsurge in the standard of living have been predicted to have a significant consequence on the world's energy demand. It is obvious that part of this demand for energy will be met from the use of natural gas. Again, the yearning desire globally to reduce or eliminate global warming associated with the release of greenhouse gases has been a great concern to both the government and industries. This emission has constrained the use of fossil fuel to satisfy the increasing demand for energy. In the way of exploring for clean energy, natural gas was found to be a less polluting energy resource to coal and oil. It is known that more than 24% of the primary global energy supply is provided by natural gas[3]; [4]. From research, it is projected that world consumption of gas will double by 2030[5]. Based on these facts, natural gas has become an essential fuel for heating, electrical generation and mechanical drive due to its domestic abundance, low environmental emission, and high energy content.

Figure 1-1 shows the natural gas demand and supply of some of the projected European countries between 2005 and 2030. If no medium projects are put in place, the difference between the natural gas demand and supply within Europe is most likely to reach up 36 to 113 Bcm by the year 2020. Nevertheless, Trans-Saharan gas

pipeline (TSGP) is an ongoing planned gas supply projects which, aimed at filling this future demand

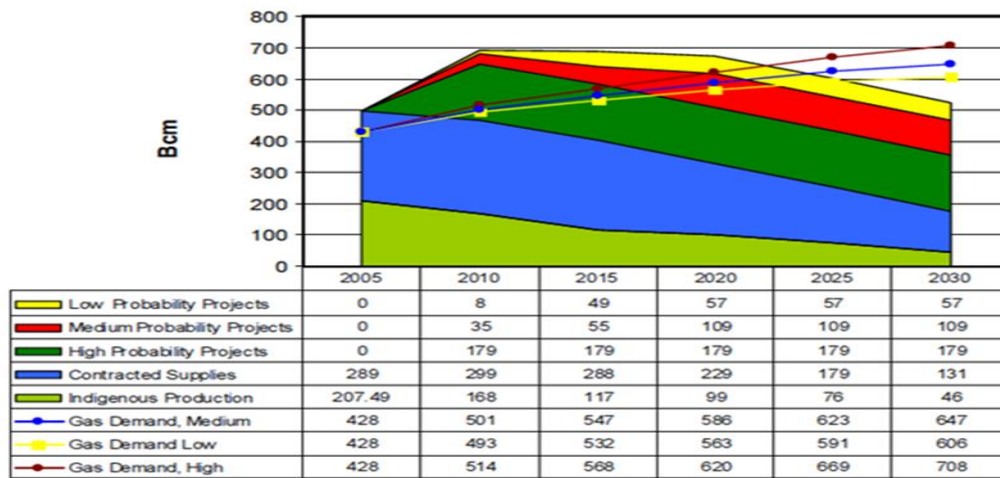


Figure 1-1 Natural Gas Demand and Supply for EU [6]

Transportation is required for active movement of this gas from the treatment plant to consumer. Presently, the easiest way of moving this gas through a long-distance is by the use of pipeline network systems. Gas transmission network comprises of large numbers of pipes, regulators, and valves with injection and delivery points and compressor stations [3]; [7]. A pipeline carrying natural gas which may span several kilometres needs compressor stations which are placed at a strategic point along the pipeline. This help to minimise pressure loss along the pipelines by boosting the reducing gas pressure. These compression systems must be supervised to ensure a high level of operational reliability. The compression system accommodates gas compressor which may be either of the reciprocating or centrifugal types. This gas compressor is solely responsible for the pressure boost as the gas flow to each of the station along the pipeline. Most of the gas compression system makes use of centrifugal compressor driven by either electric motor or gas turbine. [8]

Centrifugal compressor (GC) is a fluid handling mechanical device capable of transferring energy to the fluid medium. The fluid is delivered in a large volume at high delivery pressure.

It has much application, which ranges from process industries, aircraft and to household appliances such as refrigerators and air conditioners. Depending on a

specific application and how pressure energy is imparted to the air, Figure 1-2 illustrates the gas transmission process along a pipeline.

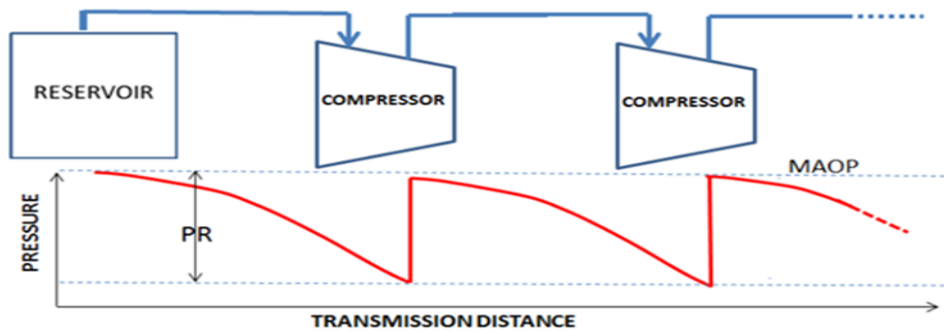


Figure 1-2 Layout of Pipeline with Compressor stations[9]

The centrifugal compressor must have a reliable and efficient prime mover to have effective and steady gas transmission from one compressor station to another. Gas turbine (GT) engine becomes the favoured prime mover along the natural gas pipeline because of the remote location of the compressor that is impoverished of electricity. Also, GT serviced by natural gas is reliable and efficient in terms of the energy source as it uses the gas from the pipeline[10]; [11]; [12] Compressor station for a gas transmission includes GT, GC, pipeline and all necessary accessories.

The major issue with gas turbine operation besides its advantages is the effect of the changes in ambient temperature. A typical example is in the dry and hot region like Nigeria, where the ambient condition is considered in this research. Also, all prime mover, including GT engines degrades with time. The degradation, in addition to ambient temperature, affects the overall performance of the engines[13]; [14]; [15]. These significantly affect the engine performance and causes changes in the component maps and reduces the efficiencies. Also, due to the economic benefits of the compressor station, the engines are always allowed to run continuously for an extended period [14]. Application of gas turbine, as part of this project, requires investigation in terms of the cycle architecture and its performance analysis.

Economically, this is a capital-intensive project because the equipment, such as GT and GC operates under diverse local conditions, including all the pipeline accessories. The costs of these components form an integral part of the pipeline project. The operation of these machines in the compression station involves the interaction of several components that leads to degradation over time, as earlier

mentioned. The degradation can occur in the form of fouling, wear and tears etc. Also, different types of emissions are generated when the gas turbine operates. The techno-economic success of any compression station operation is hinged on the involvement of the compressor[16]. The reason is that the pressure loss minimisation and GT selection depends on the capacity of the gas compressor.

1.1.1 Identification of Research Gap and Contribution to Knowledge

The techno-economic assessment of the natural gas pipeline transportation system involves many variables. The variables include the various compressor station machinery/accessories, the ambient operating conditions and the economic viability of the project. A tool or technique that captures the entire system's evaluation is needed. Therefore, this research employs the techno-economic and environmental risk assessment (TERA) framework, which is the core of this research

Relative to previous studies, this research has proposed a methodology which;

- (i) accounts for the effects of flow capacity and isentropic efficiency degradation of the turbomachinery component of the GT on the performance and economic viability of the pipeline project.

Much research has not been explored in applying the TERA tool on natural gas pipeline transportation system. The only two known work on TERA application on natural gas pipeline transportation as at the time of this research was the research work of Nasir[17] and Tukur[18]. Both authors assumed that the gas turbine engine operation was clean throughout the project life. To author's best knowledge, previous TERA studies on a natural gas pipeline as stated above has not considered this aspect. Therefore gap in knowledge exists which this research tries to cover.

- (ii) account for the impacts of degradation on the cost of CO₂ to form an integral part of the life cycle cost of the pipeline project due to increased fuel consumption.

To the author's best knowledge, no previous work has looked into the influence of degradation on the cost of CO₂ generated from a natural gas pipeline application.

- (iii) captured the effects of variation in ambient condition (time-based temperature and altitude) and desired volume flow on gas compressor power evaluation and pipeline performance

To author's best knowledge, previous work on gas compressor and pipeline such as reported in Campbell[19], Kurz et al. [20], Lars et al. [21] and Menon [22] have not considered this aspect which this research tries to cover

- (iv) applied compressor washing for gas turbine NO_x and CO₂ emission reduction.

Also, to author's best knowledge, previous work on compressor washing such as reported in Musa [23], Leusden et al. [24] and Abdelrazik et al. [25] have not considered this aspect. Therefore gap in knowledge exists which this research tries to cover.

1.2 Motivation and Problem Statement

The Trans-Saharan Gas Pipeline (TSGP) is a conceivable natural gas pipeline project by the Nigerian government which can aid the economic development of the country. This project is expected to transport natural gas from Nigeria to Europe through Niger, Algeria to Spain[26]. The TSGP forms the basis of this thesis. The motivation for the present study is hinged on the problem this research aims to solve. The motivation includes but not limited to i). The challenge of the increase in the demand for natural gas all over the globe. ii). The need and urgency to resume the abandoned TSGP project by the federal republic of Nigerian. iii). To explore the opportunities in natural gas pipeline business with particular attention of evaluating the economic implication of running the GT engines in a degraded mode, and iv) to implement the pipeline group at the centre for power and propulsion in Cranfield University amongst others.

The high demand in the world today for energy specifically those of Europe and America as a result of economic growth and the concern for environmental protection, geared up the need to have an environmentally fuel-friendly. Since natural gas has been earmarked to serve this need, it becomes imperative to have a well-arranged booster stations and equipment's selection to enhance a better technical and economic evaluation of the natural gas transmission system. Adequate selection

of the prime mover and the various compressors to be used is a thing of concern in this project. More importantly, keeping the engine in operation without downtime amid a degradable mechanism before and after the overhaul is a thing of concern also. In practical term, degradation of the turbomachinery is a crucial ingredient to be a concern with both in performance and the economic implication in terms of cost and benefits. To meet up these challenges and to have uninterrupted gas delivery, the following conditions have to be attended to:

- ❖ A pipeline system with compressor stations at a strategic position has to be in place
- ❖ The required number and size of GT engines that can meet the power requirement of GC in each CS especially at high ambient temperature and GT degradation must be in place
- ❖ The total capital cost of transporting the required volume of gas has to be accommodated when compared with the benefits of investment.
- ❖ To have an efficient maintenance plan for the project facilities, especially on the GTs for continuous gas delivery.
- ❖ Operating and integrated logistics system cost has to be minimum

As the research progresses and to address the conditions outlined above, specific logical and hypothetical questions that this research will answer need to be reasoned with. The questions are itemised below:

- Considering the impacts of varying operational condition at different seasons of the year and also with engine degradation, what size and number of GT engines needed at each compressor stations to meet up the power requirement of the GCs at each station especially at the pessimistic condition (worst case scenario) of the engine degradation?
- How does engine degradation affect operation and maintenance cost (O&M), fuel, life cycle cost (LCC) and the Net present value (NPV) of the project?
- How will the baseline case be technically improved upon to have a lower LCC and high NPV
- Which maintenance action technique could be adopted on the engine in order to keep a constant gas transport without any downtime due to degradation?

1.3 Aim of Research

The aim of this research is to investigate the impact of gas turbine degradation on the TERA of natural gas pipeline investment.

1.4 Research Objectives

The objectives are

- Evaluation of gas compressor (GC) power across 18 compression station based on Trans-Saharan Gas Pipeline (TSGP) report data and the pipeline network in a non-isothermal condition
- Selection, modelling & performance evaluation at design point (DP), off-design (OD) and degraded conditions of the GTs based on the GC power requirement
- To evaluate the influence of GT degradation for a given pipeline application, taking into account the time-based ambient temperature of the project site. The three degraded scenarios (reduction in flow capacity and isentropic efficiency) and their ranges include:
 - ❖ Optimistic scenario----(0.25 -3.15%)
 - ❖ Medium scenario----- (1.25- 4.15%)
 - ❖ Pessimistic scenario---(2.25-5.51%)
- To quantify the total NO_x and CO₂ emissions generated throughout the project life
- To evaluate the economic viability of investing in the natural gas pipeline project using NPV approach: The NPV appraisal method include
 - ❖ NPV with engine divestment
 - ❖ NPV with engine leasing
- To evaluate the impact of GT "re-scheduled-overhaul" on
 - ❖ the engine performance with respect to fuel consumption and TET,
 - ❖ engine creep life
 - ❖ CO₂ emission
 - ❖ the project life cycle cost also accounting for CO₂ emission tax
 - ❖ economic benefit of the project relative to the investment cost

- To evaluate the impacts of compressor washing on
 - ❖ GT performance with respect to TET, mass flow, PR and fuel flow
 - ❖ NO_x and CO₂ Emission
 - ❖ the economic enhancement of the pipeline project accounting for the washing cost and the overall benefit using the NPV method.
- To evaluate the impact of GT “re-scheduled-overhaul” on the Techno-economic of the pipeline network.
- To evaluate the impact of compressor washing on GT performance, economic benefit and emission reduction on pipeline investment

(The non-isothermal condition means varying the ambient temperature)

1.5 Research Contribution

To address the need of this research for the Nigerian Government and by extension to enhance efficient operation in the pipeline industry the following has been contributed to the knowledge

1.5.1 The results on the impact of engine degradation on the techno-economic assessment of pipeline investment

Many successes have been achieved in the process of this research. These results are a novel contribution to knowledge. These are itemised below:

- For the first time research to show not just how degradation affects engine performance but how it influenced the number of engine usage on pipeline application at a controlled TET
- Varying yearly O&M and fuel cost of GT for pipeline application in a time-based system has been investigated amid engine degradation
- The result showed that GT Rescheduled Overhaul has an influence on the techno-economics performance of natural gas pipeline investment by reducing the NGT, fuel flow and CO₂ emission cost
- The combined effect of timed-based ambient temperature and degradation on GT emissions operating on natural gas pipeline transportation is

investigated. In this regard, the influence of degraded GT part-load operation on reducing NOx emission was investigated

- A study of the effect of degraded GT emissions operating in a timed-based ambient temperature on the economics of natural gas pipeline transportation has been conducted
- Compressor washing reduces the TET, fuel flow, improves the economics of NG pipeline transportation network and reduces GT emission and cost at fixed shaft power operation.
- The economic investigation of the influence of engine divestment and leasing on redundant fleets of engines after overhaul shows that engine leasing has better NPV than engine divestment.
- A well-detailed impact of GTs degradation on LCC and NPV of natural gas pipeline investment has also been investigated for the first time.

The results here can be used as estimates of expected benefits and relevant information on investing in natural gas pipeline transportation business.

1.5.2 Development of a model and methodology to evaluate the economic implication of using a degraded gas turbine in a time-based ambient temperature for natural gas pipeline investment

Having done a detailed review of literature on the techno-economics assessment of a natural gas pipeline, the research gap as identified in public domain is the lack of literature to show the economic impact of engine degradation for the entire project life since the engine must degrade with time. This research has successfully developed a model to evaluate the O&M cost, fuel cost and the benefits of investing into natural gas pipeline taken into account the impact of timed-based ambient temperature and engine degradation in each season of the year, throughout the project life.

This developed model was used to evaluate the economics of investing in a gas pipeline using the data from the TSGP report of Nigeria and can be applied to similar work anywhere in the world. The robustness of the economic model proposed by this research makes it useful as a guide for investing in the pipeline business.

1.5.3 Development of an emission model and methodology to quantify gas turbine emission and cost of CO₂ in a timed-based system

The production of emission from GT operation has geared up the need to quantify the emission that will be generated from the operation of these GTs on the pipeline network. Exploring the literature, empirical equations on the considered emission which are duly referenced in the relevant section of this work were used to develop a model to quantify these emissions for the entire project life. The model takes into account the impact of GT degradation and time based-ambient temperature at different scenarios in each season of the year. Also, the model was used to evaluate CO₂ tax for all the 18 CS at different scenarios throughout the project life.

The developed emission model will help investors and relevant government agencies to evaluate the precise operational amount of emission from GT operation, taking into account the potential influence of timed-based ambient temperature and engine degradation during operation.

1.6 Scope of Research and Limitation

To streamline the research and to ensure the accomplishment of the research within the time frame as required, it is essential to define some boundaries to support the aim and objectives as stated above. Similarly, some limitations of the research are outlined. The boundaries are thus stated:

- ✓ The gas turbine operation is mainly for mechanical drive not for power generation hence alternator's application becomes inconsequential
- ✓ There are several mechanisms accountable for gas turbine degradation; nevertheless, only fouling will be considered in this research. The reason is that fouling is very common in the area where the engine will be operated.
- ✓ Among the three crucial turbomachinery components of the gas turbine, only Combustor degradation will not be looked into. Hence, degradation of both "compressor, compressor turbine and power Turbine" as one type of degradation will be considered for this research.
- ✓ Whenever the word " degradation" is used anywhere in this work, it refers to only fouling.

- ✓ The word “winter” and “rainy” season means the same season as it may be used interchangeably at different occasions. The reason is that the region (Nigeria) where this research is focussed on observed the rainy season as their winter period.
- ✓ Among all engine emissions, only NO_x and CO₂ are being investigated. Also, the CO₂ cost only was used in economic analysis.
- ✓ Empirical emission model was chosen because most of the successful model equation today is derived from empirical analysis. Also, the empirical equation has been seen as been accurate in some investigative study.
- ✓ Engine divestment and leasing approach was applied to address the issue of redundant fleets after an engine overhaul
- ✓ Only online-compressor washing is analysed in this research. The reason is that downtime is not an encourageable practice on natural gas transportation.
- ✓ Only the NPV as an appraisal method shall be considered to evaluate the economic benefits of the pipeline project.

Some of the research limitations are thus stated:

- ✓ Modelling uncertainties and their implications were not incorporated into the GT modelling process.
- ✓ None-inclusion of residence time in emission evaluation.

Conclusively, The TSGP is seen as a strategic project because when constructed, will generate substantial revenue to the economy and open opportunities to Nigerian gas sector. The developed economic model and methodology will provide useful decision making for the Nigerian government for investing in the TSGP. The achieved results will also help in the financial budget of the Nigerian government towards investing in the TSGP and by extension, serve as useful information for pipeline investors.

1.7 Publications and Conferences during PhD

1.7.1 Publications

- Aziaka D.S, Tukur N, Pilidis P (2019). Techno-Economic Evaluation of Pipeline Compression System: Evaluation of Gas Compressor Power in a Natural Gas Pipeline Network-Part 1: International Journal of Mechanical Engineering and Technology (IJMET). 10 (11), 2019, pp 102-118. Article ID: IJMET_10_11_010
http://iaeme.com/MasterAdmin/UploadFolder/IJMET_10_11_010-2/IJMET_10_11_010-2.pdf
- D **Aziaka D.S**, Tukur N, Pilidis P. (2019). Techno-Economic Evaluation of Pipeline Compression System: Gas Compressor Operation Evaluation on Natural Gas Pipeline Network-Part 2. International Journal Of Mechanical Engineering and Technology (IJMET). 10 (11), 2019, pp 119-132. Article ID: IJMET_10_11_011
http://iaeme.com/MasterAdmin/UploadFolder/IJMET_10_11_011/IJMET_10_11_011.pdf
- **Aziaka D.S**, Tukur N, Pilidis P. (2019) Techno-Economic Evaluation of Pipeline Compression System: Economic Evaluation of the Natural Gas Pipeline Compression System-Part 3: International Journal Of Mechanical Engineering and Technology (IJMET). 10 (11), 2019, pp 133-154. Article ID: IJMET_10_11_012
http://iaeme.com/MasterAdmin/UploadFolder/IJMET_10_11_012/IJMET_10_11_012.pdf

1.7.2 Conferences

- **M. Obhuo; D.S. Aziaka, & P. Pilidis (2018)** “ The Delaying Influence of Degradation on the Divestment of Gas Turbines for Associated Gas Utilisation.: Presented at the 22nd International Conference on Modern Gas Turbine Systems, London (13th & 14th February 2020), (ICMGTS 2020)
- **D.S. Aziaka, U.; Nasir. A.; Igie U.; & P. Pilidis:** Gas turbine Emission Reduction through Compressor Washing on Natural gas Pipeline Application: Presented at the 3rd International Conference & Expo On Aerospace and Unmanned Aerial Systems (Aero-Events-2020) in Philadelphia, USA. (February 17-18, 2020).

- **D.S. Aziaka, U. Igie, & P. Pilidis (2018)** ‘The Influence of Time-based Gas turbine Degradation on Life Cycle Cost of Pipeline Investment’: Poster Presentation at the 2nd Annual Cranfield Aerospace Research Students Conference ---2018
- **D.S. Aziaka, U. Igie, & P. Pilidis** “ The Influence of Rescheduled-Overhaul and Degradation of Gas turbine in Determining the Number of Gas turbine Usage in Natural Gas Pipeline Transportation Network: presented at the 10th international conference and EXPO on Oil and Gas, London. September 2019
- **D.S Aziaka, A.Le-Ol, T.Tamuno & P.Pilidis:** The Impacts of Component Degradation on the Overall Performance of Gas turbine used on Natural Gas Pipeline Network: Presented at Center for African Development Studies, University of Ibadan, Ibadan, Oyo State, Nigeria. March 2019

1.8 Thesis Structure

The thesis comprises of eight chapters. The description of each of these chapters summarises the contents of what is involves in those chapters. The chapters are found to be interdependent on one another. These chapters are briefly discussed below

Chapter 1: This is an introductory chapter that explained the context of the research and relevant sections that are vital to this research. The background of the research that emphasised the real heart of this research is discussed. The aim and objectives of the research are outlined alongside the boundaries of this research. The problem statement which outlined some hypothetical questions this research must answer, followed by the research contribution to knowledge and relevant conferences attended, including publications achieved during the research period are all discussed in this chapter.

Chapter 2 presents the literature reviews of related studies for an integrated pipeline system. The reviews covered natural gas reserves by countries, and the mode of transportation, pipeline design consideration, its accessories, and compressor station. Prime mover selection for gas transportation and relevant maintenance action plan are also reviewed. The TSGP project, TERA Application and progress beyond state-of-the-art are reviewed in this chapter. It also highlights some of the gaps the research has identified

Chapter 3 presents the methodology and tools adopted in this research. The pipeline performance and gas compressor power evaluation are presented. Several operational conditions of the pipeline and gas compressor are discussed. The obtained results are presented and discussed.

The developed TERA framework for the different modules considered in this research is presented. All the modules that this research aims to consider are discussed. TURBOMATCH software is also discussed.

Chapter 4 presents the gas turbine selection, modelling and performance evaluation. The significance of this chapter is that the developed model was validated at their design point, which is the baseline model. The off-design simulation using different operating condition were done to validate the model at OD condition. It also explains how the performance parameters deviate from their DP due to degradation

Chapter 5: This chapter is divided into two sections. Section one detailed the influence of time-based ambient temperature and degradation in determining the number of engine usage in each compressor station. Section two presents the development of the emission model using some empirical model equation. The model was developed in MATLAB. The various emissions under consideration and the corresponding cost of CO₂ emission based on the influence of the time-based ambient temperature and degradation were also analysed and presented

Chapter 6 presents the development of the economic model of the natural gas pipeline network in MATLAB. The different scenarios under investigation (clean, optimistic, medium, and pessimistic) were contained in the developed model. The results obtained from this model served as the research baseline case. The composition of the compressor station was identified and analysed in the developed model. The developed method to evaluate the capital expenditure of the project was explained. In this chapter also, the economic decisions were made based on the data received from the GT, GC, pipeline and its accessories. The developed model evaluated the fuel and maintenance cost in a time-based system in three seasons of the year for thirty years of project life. The life cycle cost, in addition to sensitivity analysis in terms of economic inflation, was presented. NPV as an appraisal method to evaluate the economic viability of the project in addition to engine divestment and leasing was analysed in this chapter.

Chapter 7 considers the maintenance action adopted on the GT engines for the pipeline operation. This chapter is divided into two sections. Section one deals with the impacts of a re-schedule overhaul, on the techno-economics of the natural gas pipeline system. Section two deals with the influence of compressor washing on GT performance, economic improvement and emission reduction for a natural gas pipeline application

Chapter 8 provides a summary of the entire research already conducted. It also recommends areas of further improvement in line with this field of research.

1.9 Summary of Chapter

In this chapter. The background and overview of research which encapsulates all relevant areas of natural gas transportation system have been explained and justified. Also, the research gap and needs that will be addressed have been described, including the scope of this research. In addition to the needs or problem statement. The technical and hypothetical questions this research tends to address are clarified. Furthermore, the main reason for undertaking this research, which is the main aim is clearly defined. The different objectives and tasks that will foster the achievement of the main aim and answer the hypothetical questions of this research have been defined and highlighted. Several contributions, publications and conferences attended on the bases of this research and in line with the research gap are also outlined. All chapters of this research are hereby summarised and presented.

2 LITERATURE REVIEW

2.1 Background

To address the theme of this research, various component areas of literature that form an integral part of this research has been addressed. The theme of this research function with many bodies and the collective entity is a phenomenon of interest. These areas of different components form the basis of this research. This include:

- (i) Natural gas and means of transportation
- (ii) Gas compressors
- (iii) Gas turbine engines
- (iv) Engine maintenance action
- (v) TERA framework

These five aspects/areas of the literature have been reviewed to provide a theoretical and analytical framework for the investigation concerning the specific logical and hypothetical question of this research. Understanding the various components will enable a clearer view of the focus of this research.

2.1.1 Natural Gas Reserve and Productions by Countries

It is known that some countries that have the highest consumption rate of natural gas do not have enough reserve like Europe, North America etc. [27]. It is also clear that 80% of the world proven natural gas reserve is found in ten countries with Russia topping the list with one-fourth ($\frac{1}{4}$) of world's total gas reserves followed by Iran and Qatar in the Middle East. It is worth noting that Nigeria, where this research is focussed, is among the ten countries in the world with the highest natural gas reserve as represented in Figure 2-1 [28]. It therefore implies that for this need to be met, natural gas must be transported from place or country with large reserve to a region where the gas reserve is limited.

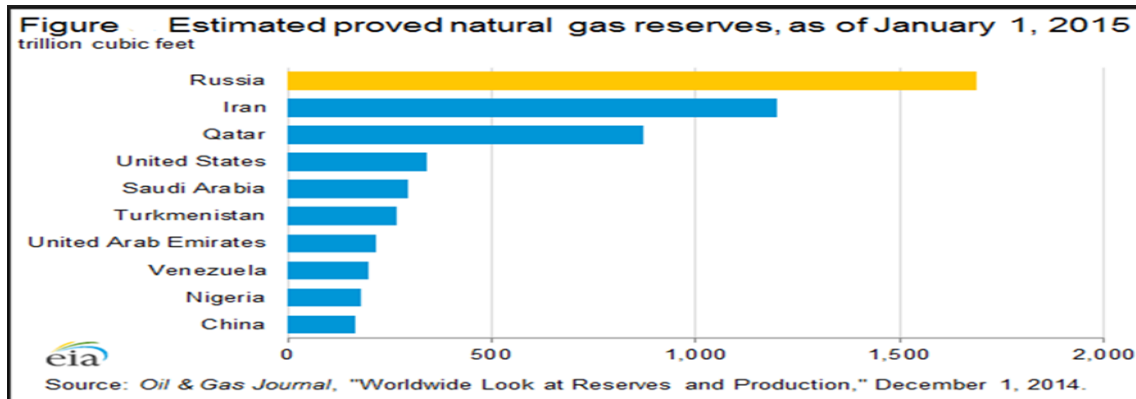


Figure 2-1 Natural gas reserve by countries[28]

2.2 Natural Gas Transportation System

Several countries of the world and industries make use of natural gas in both heating and cooking their food at home. It is also useful as fuel for cars and power generation due to its numerous advantages. Such advantages include i) It burns cleaner than other fossil fuels. Hence, it is an environmentally friendly gas ii) It is easier to save and store when compared to other fossil fuels, iii) It is less expensive when compared to other fossil fuels. The challenge in natural gas is in transportation and storage as a result of its low density.

There exist a long distance to cover between the production and end-users. For this reason, a cost-effective means of transporting this gas becomes vital. Having this means of transportation will tie the gap between the producer and the end-user. Some methods employed for gas transportation was analysed by Rajnauth et al. [29] and was reviewed by Thomas and Dawe[30]. These include Pipeline Natural Gas (PNG), Liquefied Natural Gas (LNG), and Gas to wire (GTW), Gas to liquid (GTL), Natural Gas Hydrate, and Compressed Natural Gas (CNG). However, only pipeline natural gas shall be discussed.

2.2.1 Natural Gas Pipeline Transportation System

All modern nations rely almost exclusively on pipelines to transport gas, water, crude oil, sewage and products that are refined from a refinery to other cities over several kilometres [31]. Transportation of natural gas through a pipeline has proved efficient over the years. It has also enhanced the availability of natural gas in regions where natural gas is not produced, especially a long-distance region.

To have efficient transportation of natural gas from production to consumers, pipeline net network, compressor stations, city gate station and storage facilities have to be in place. The heart of the transmission system of natural gas is the compressor station. The reason is that it enables the gas to flow at a continuous rate and pressure; hence, it is also called a booster station. Trunk or gathering, transmission or transportation and distribution pipelines are the three types of gas transmission system for onshore or offshore pipelines. However, many factors such as gas reserve, distance to the consumer, time frame to monetise the gas, available infrastructure, investment and gas processing affects the types of a gas transportation system to be adopted [29]. The demand for transporting natural gas over a long distance through pipelines is on the increase as investigated by Rajnauth et al.[29]. However, Pipeline transportation is more efficient and suitable for short-distance less than 3000km. Transportation efficiency for various gas transportation methods is represented in Figure 2-2 [29]. It shows that pipeline transportation method has the highest efficiency for a pipeline length of less than 100km. It can also be used for a distance less than 5000km beside gas hydrate and LNG.

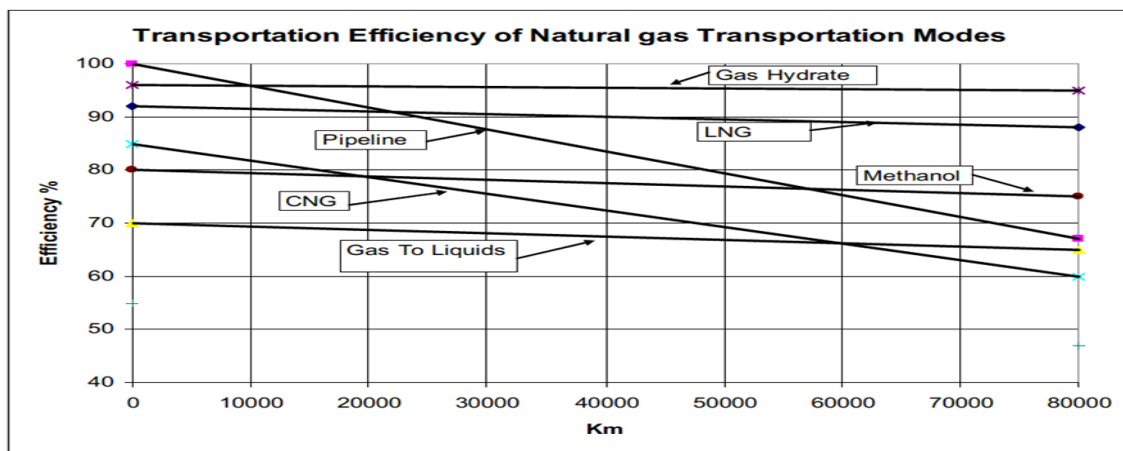


Figure 2-2 Transportation efficiency for various transportation mode [29]

Pipelines primarily serve as a means of moving the natural gas from the production site to the consumers [29]. Usually, the construction of pipelines could either be underground, above the ground or subsea in the case of offshore arrangement and they are constructed in a mode that the gas reaches its predetermined point in a fast and effective way [32]. Wang and Economides [33] presented that pipeline accounts for about 70% of global natural gas transport.

The mechanical integrity of the pipes are firm and are generally constructed of carbon steel and with various sizes ranging from 2 inches (0.051m) to 56 inches (1.4m) in diameter depending on the type of pipeline network and its application [29]

The pipeline system which conveys the gas from gas fields either onshore or offshore facilities through gathering systems to residential, industrial, commercial and utility companies, could be inter or intrastate. The pipeline system has the inlet point, which could either be offshore or onshore compressor station. Also, the outlet point, which could also be another platform or an onshore receiving station. Another type of pipeline system can be the one from gas treatment plant which passes through scrubbers, liquid removal tank, compressor stations, valves, intercooler, and the city gate into the distribution line and finally to the consumers. A typical pipeline layout is shown in Figure 2-3, which are classified into three categories:

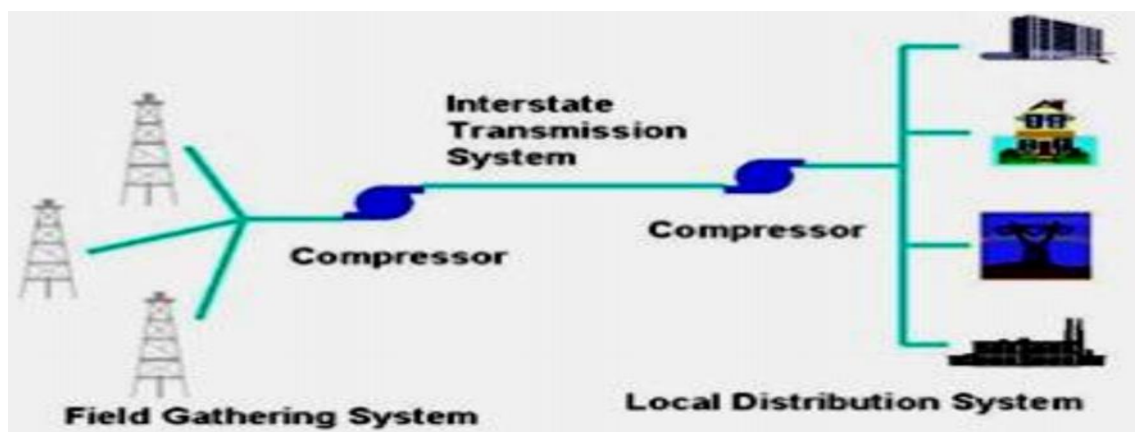


Figure 2-3 Pipeline layout[32]

The three set classifications are:

- **Gas gathering pipeline:** These are short distance with small diameter pipeline used to convey natural gas from the production well to another facility for further refinement or send to the transmission line.
- **Gas transmission line:** These are the natural gas pipeline used for long-distance with a big diameter for gas transportation within or outside the country. These types of pipeline network need compressor station placed at a convenient position along the pipeline to avoid pressure loss
- **Gas distribution pipeline:** These are the set of pipeline used for delivering natural gas to consumers at low pressure. The difference between the

distribution pipeline and the transmission line is seen in their diameter, simplicity of design due to absent of valves, compressors or nozzle and their operation at low and medium pressure.[32]

Rajnauth *et al.*, [29] explained that the available natural gas volume could determine the gas transportation mode to be implemented. Every efficient pipeline network has different types of equipment included in the network which are gathered together to move the gas from one point to another. Lun and Xie [34] noted that gas flow along the pipeline encounters pressure and energy losses due to the friction between the gas and the inner walls of gas ducts. They presented that a known factor that causes the pressure and energy losses include the heat transfer between gas and the environment. They further explained that pressure drop along the pipeline could be restored by installing a compressor station on the pipeline network. It was concluded that in the sixties (60s), the pressure of natural gas pipelines was only 2.5 mPa before it rose to 6 mPa, but currently, the pressure of the pipeline is about 10mPa. The need to restore pressure drop along a pipeline is a concern to researchers in this field of study. This concern centres on the way to deliver the gas with a specified pressure without dropping.

Having reviewed the importance of pipeline flow, this research considers only single-phase flow compressible flow that transport natural gas through the pipeline. Also, the impacts of the pressure and gas temperature on the gas flow rate shall be looked into.

2.3 Pipeline Design Consideration

Gas pipeline design requires an in-depth knowledge of head loss brought about by fluid friction, wall friction, bends, and necessary pipe parameters such as length, diameters, topography and the amount of gas being transported. Consideration of pressure loss along the pipeline, especially when the pipe length is very long is very crucial. Nasir [17] explained that changes in pressure are always present if the distance of the pipe is far. Therefore, it becomes essential to apply the concepts of compressible fluid flow.

The throughput (flow rate) of the system depends on the pipe length, diameter, gas properties, initial gas pressure and temperature and the pressure drop due to friction.

In this case, commonly used model equations will be considered. Montoya [35] presented that optimisation of the system topology (geometry), size and operational conditions to minimise investment and maintenance cost will be required when designing a new gas pipeline transmission system or the expansion of any existing one. The crucial steps in the pipeline system design comprise the establishment of critical pipeline performance objectives and critical engineering design parameters such as:

- The required volume of gas per unit time (throughput)
- Source and terminating point of gas movement
- The topography of the pipeline route
- Thickness, diameter and required yield strengths
- Hydraulic calculations to determine losses
- Number of the compressor station and the distance between them or station
- The power requirement of the compressor station power

Menon [22] made some simplification by assuming a uniform gas temperature with no heat transfer between the gas and the surrounding soil in a buried pipeline in order to espouse pipeline model equations for a manual calculation. However, in reality, maintaining the gas temperature constant is virtually impossible. Subero *et al.*, [36] investigated that pipeline for gas transportation system produced a better return on investment(**ROI**) when compared with other means of transportation. Watanabe [32] analysed in his work that gas transportation system is affected by distance to the consumer. Hence, the need to have a well-designed compressor station along the pipeline. The ability to choose the correct model for pipeline simulation depends on the network structure and network complexity. Ouyang and Aziz. [37] investigated that oversimplification by assumptions, approximations and addition of inaccurate friction factor correlation in some equations of flow presented in literature has led to nontrivial errors in natural gas pipeline calculation with respect to flowrate and pipeline hydraulic evaluation.

2.3.1 Analysis of Gas Flow

The total energy of the gas at various points consists of energy due to pressure, velocity, and position or elevation above an established datum[22]. As the gas flows through the pipeline, Bernoulli's equation is applied to connect the components of

the energy of the flowing fluid to form energy conservation. Equation (2-1) shows the Bernoulli's equation considering two points as indicated in Figure 2-4

$$Z_A \frac{P_A}{\gamma} + \frac{V_A^2}{2g} + H_p = Z_B + \frac{P_B}{\gamma} + \frac{V_B^2}{2g} + h_f \quad (2-1)$$

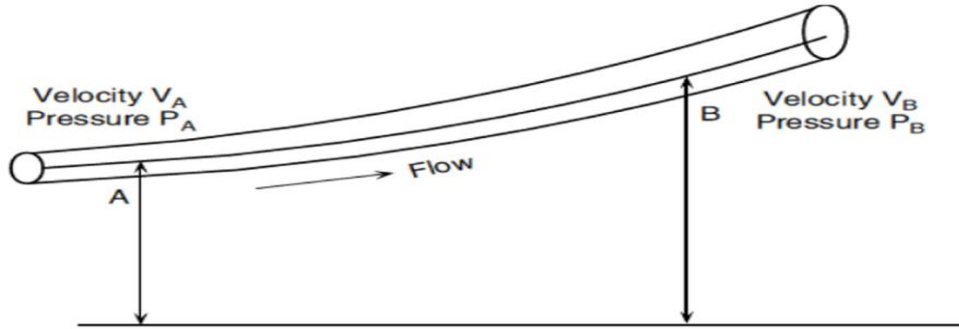


Figure 2-4 Energy of flowing fluid[38]

H_p represents the equivalent head added to the fluid by a compressor at A, h_f shows the total frictional pressure loss between points A and B while Z_A and Z_B represent the elevation at A and B

The basic equation for flow has been developed for many years now. Considering the energy Equation (2-1) and in addition to gas laws, the performance of a pipeline conveying gas can be predicted. Adequate design of a gas transmission pipeline requires methods for predicting pressure drop for a given flow rate or predicting flow rate for a specified pressure drop in combination with installed compression power and energy requirement. Conversely, there is a need for practical methods to relate the flow of gas through a pipeline to the properties of both the pipeline and gas and to the operating conditions, such as temperature and pressure. For a single-phase, dry gas pipeline, the isothermal steady-state pressure drop calculation approach is commonly utilised and are the primary link in the gas delivery system[39]; [40]; [41]. However, the present study used non-uniform gas temperature with respect to time of the day for the pipeline investigation.

There are several commonly used model equations to predict the performance of a pipeline transporting gas. These include

- General Flow equation
- Colebrook-White equation

- Modified Colebrook-White equation
- AGA equation
- Weymouth equation

These equations show the relationship among the gas properties, such as compressibility factor, gravity, flow rate, pipe length, diameter, and pressures along the pipeline[22]; [42]. The general flow equation shall be discussed in relating to Weighmouth equation

2.3.1.1 General flow equation

The basic equation for connecting the pressure drop with flow rate is the general flow equation for the single-phase steady-state isothermal flow in a gas pipeline. It is also regarded as the fundamental flow equation, for a steady-state isothermal flow in a gas pipeline.

Figure 2-5 represents a typical pipeline showing a steady flow. In analysing the flow in the pipeline, Figure 2-5 in S.I unit is given concerning pipe diameter, temperatures, pressures, gas properties and flow rate in Equation (2-2) [48]

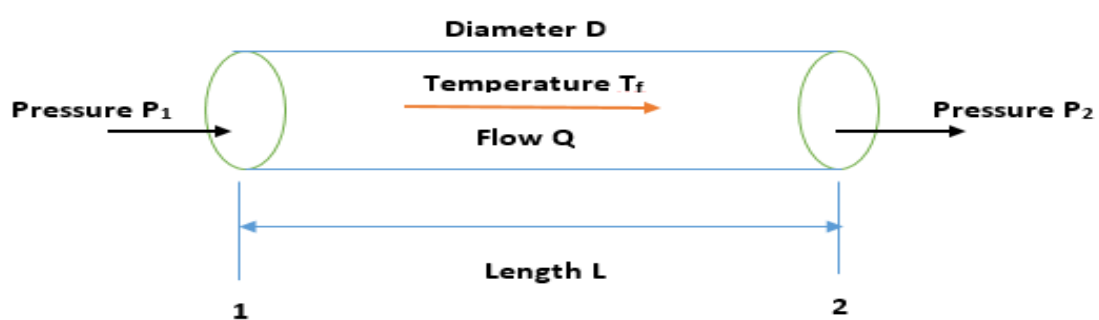
$$Q = 1.1494 \times 10^{-3} \left(\frac{T_b}{P_b} \right) \left[\left[\frac{(P_1^2 - P_2^2)}{f G T_f L Z} \right] \right]^{0.5} D^{2.5} \quad (2-2)$$


Figure 2-5 Steady flow in gas pipeline

Where,

Q = gas flow rate, measured at standard conditions, m³/day

f=friction factor, dimensionless

Pb=base pressure, kPa

T_b=base temperature, K (273

P₁=upstream pressure, kPa

P₂=downstream pressure, kPa

G=gas gravity (air=1.00)

T_f=average gas flowing temperature, K (273+°C)

L=pipe segment length (horizontal pipe), km

Z=gas compressibility factor at the flowing temperature, dimensionless

D=pipe inside diameter, mm

From Figure 2-5, point 1 to 2 on the pipe segment T_f is assumed to be constant that is, Isothermal condition. In looking into the general flow Equation (2-2), it becomes clear that the pipe segment of length L, Gas flow rate Q, at standard condition, and diameter, D depends on many factors. Q depends on gas properties represented by the gravity G, and the compressibility factor Z increase in the gas gravity (heavier gas), resulting in a decrease in the flow rate. Likewise, as the compressibility factor Z rises, the flow rate will decrease.

It is evident here that the influence of inside diameter and pipe length is apparent. As the pipe segment length increases for a given pressure P₁ and P₂, it results to decrease in the flow rate. Conversely, as the diameter becomes bigger, the flow rate increases also. The term $P_1^2 - P_2^2$ denotes the driving force that affects the flow rate from the upstream end to the downstream end. Keeping the upstream pressure P₁ constant as the downstream pressure P₂ increases, the flow rate will increase. When there is friction between the gas and pipe walls, pressure drop (P₁-P₂) is bound to occur from the upstream point 1 to downstream point 2. The friction factor f is a function of the internal condition of the pipe as well as the type of flow (laminar or turbulent). Equation (2-2) was established with the assumption that the gas pipeline is horizontal; this is commonly not entirely so. Therefore, Equation (2-2) is not suitable for this research because:

- (i) It was developed for Isothermal condition
- (ii) It was established with the assumption that the gas pipeline is horizontal; hence, it does not take pipeline elevation into consideration.

2.3.1.2 Weymouth Equation

This equation is useful for a large diameter pipe with high pressure and high flow rate. For a given value of gas gravity, inlet and outlet pressures, compressibility and length of pipe, the Weymouth equation directly calculate the flow rate through a pipeline. The formula is represented in Equation (2-3)

$$Q = 3.7435 * 10^{-3} E \left(\frac{T_b}{P_b} \right) \left[\frac{P_1^2 - e^s P_2^2}{G T_f L_e Z} \right]^{0.5} D^{2.667} \quad (2-3)$$

Where Q-the gas mass flow (m³/day), E- pipeline efficiency (%) a decimal value less than or equal to 1.0. (P₁ & P₂),- Inlet and outlet pressure of the pipeline (kPa), e^s, -elevation adjustment parameters L_e- equivalent length of pipe segment involving slope(Km), and other parameters as defined in Equation (2-2). The Weymouth equation was adopted for the pipeline analysis in this research. The reason is that the pipeline under investigation falls into the class of large diameter pipeline, high flow rate and high pressure. Again, Weymouth equation compensates for the pipeline elevation and uses pipeline efficiency factor to avoid the iterative process as seen in other flow equation like Colebrook-White equation that uses trial-and-error approach and ends up an iterative process[42]. Also, the Weymouth equation is being utilised in Aspen-Hysis, been the software for GC power evaluation.

Table 2-1 briefly summarises the application of the various flow equation and their corresponding application.

Table 2-1 Summarised Gas flow equation [17]

Equation	Application
General Flow	Fundamental flow equation using friction or transmission factor; used with Colebrook-White friction factor or AGA transmission factor or AGA transmission factor
Colebrook-White	Friction factor calculated for pipe roughness and Reynolds number; popular equation for general gas transmission
Modified Colebrook-White	Modified equation based on U. S. Bureau of Mines experiments; gives higher pressure drop compared to original Colebrook equation
AGA	Transmission factor calculated for partially turbulent and fully turbulent flow considering roughness, bend index, and Reynolds number
Panhandle A Panhandle B	Panhandle equations do not consider pipe roughness; instead, an efficiency factor is used; less conservative than Colebrook or AGA
Weymouth	Does not consider pipe roughness; uses an efficiency factor, used for high-pressure gas flow

Menon [42] presented that in a long gas pipeline, the temperature of the gas ultimately is equal to the surrounding soil temperature. This assumption was also

used by Tukur[18] in his research work. Therefore, the present study assumed that the gas temperature is the same as the ambient temperature.

2.4 Pipeline Components/Accessories

The natural gas pipeline has several components or accessories to enable smooth gas transportation. These components range from electronics to mechanical devices, pipeline management and contingency system. All for accurate and smooth gas transportation investment. Interestingly, these components contribute to the initial capital cost of the natural gas pipeline business. These components include:

2.4.1 Meter Stations and Regulators (MSR)

Meter station are installed along the pipeline system to measure the gas flow rate and to monitor the pipeline system performance, especially at areas where the transfer of custody takes place. The metering method may differ concerning the accuracy required, either for performance or gas sales [42]. The cost estimate for meter stations are generally estimated as a fixed price also with labour and materials per site while that of the regulating stations are also estimated as a lump sum for each station and these are added to the cost of the pipeline.

2.4.2 Mainline Valve Stations

The mainline valve equipment cost at an indicated interval along the pipelines can be estimated as a lump sum value that includes, mainline valves and operator blowdown valves and piping. Menon [42]. presented that the mainline valve is installed every 20 miles on the pipeline, and the cost of it be added to the pipeline capital cost.

2.4.3 Supervisory Control and Data Acquisition (SCADA) and Telecommunication System

The term SCADA and telecommunication system is an important electronic system used on a pipeline carrying natural gas to remotely, operate, monitor and control a natural gas pipeline network from a central control centre. Menon [42] analysed the cost of SCADA and presented that it ranges from \$2 million to \$5 million or more depending on the length of the pipeline, mainline valve station and a number of a compressor station. However, this may not be the exact cost of SCADA for an

integrated pipeline system which consists of both prime movers, the driven, pipeline and all operational accessories. Menon [42] further stated that sometimes, SCADA equipment cost is assessed as a percentage of the initial total project cost. The later on SCADA cost estimation was adopted in this research.

2.4.4 Environmental and Permitting Cost

These are the costs that are related with the pipeline amendments, compressor stations, and valve including meter stations to guarantee that these facilities do not contaminate the atmosphere, rivers and streams, or destroy the ecosystems including the flora and fauna, fish and game, and endangered species. Permitting costs has to do with those costs related with changes needed to compression equipment, pipeline alignment such that toxic emissions from pipeline facilities do not endanger the environment, humans, including plant and animal life. On a gas pipeline project, the environmental and permitting costs may range between 10 and 15% of the initial project costs.

2.4.5 Right of Way (ROW) Acquisition

The ROW for a pipeline is gotten from private parties, federal agencies, state and local government for a fee. This fee might be a lump sum of money paid during the acquisition period with additional annual fees to be paid for a specified period. Menon [42] presented that most gas pipelines project, the initial costs of ROW is in the range of 6 to 10% of the initial project costs.

2.4.6 Engineering and Construction Management (ECM)

ECM is related to the design and preparation of drawings for the entire pipeline system, which includes, pipeline, compressor stations, and other facilities. These components will take account of both preliminary and detailed engineering design costs, which includes the development of specifications, purchase documents, manuals, equipment inspection, and other costs associated with materials and equipment acquisition for the project. The ECM costs range from 15 to 20% of the initial pipeline project cost.[42]

2.5 Trans-Saharan Gas Pipeline(TSGP) Project

The Tran-Saharan Gas Pipeline (**TSGP**) project was proposed to be built through the partnership between the Nigerian National Petroleum Corporation (**NNPC**) and Algeria's Sonatrach. The purpose of this project is to convey natural gas from Nigeria to the consumer market in Europe through Niger and Algeria. The proposed route of the **TSGP** would commence from the Niger Delta region in South-South Geopolitical zone of Nigeria where most of the natural gas reserves of Nigeria are located. After that, it would then pass through the Northern part of Nigeria running through Zinder town to Algeria and be connected to the Trans-Mediterranean. It is interesting to note that Nigeria is among the first ten countries in the world with highest gas reserve with an unexploited natural gas of about 5.153Tcm. It is anticipated that the planned TSGP will help enhance the gas deficit in northern Nigeria[26].

The project will ensure closer co-operation and integration between neighbouring countries, and will eventually permit for the diversification and marketing of Nigeria's natural gas export route[6].



Figure 2-6 Trans-Saharan Gas Pipeline route[26]

The total pipeline distance to cover the estimated area was measured to be 4,180km. The pipe thickness is 56 inches with a constant output compressor station pressure of 95bar. TSGP was also estimated to have an annual capacity of 30billion cubic litres of natural gas.

2.6 Compressor Stations

The pipeline carries natural gas under pressure from the area of supply to consumers point and is expected to work continuously. Pressure loss occurs as the gas moves in the pipeline. In order to overcome this pressure loss and to have a consistent gas delivery at the desired rate, the compressor station needs to be strategically located along the pipeline. The planned location of the pipeline will help boost the pressure along the pipeline by providing the required gas pressure. The compressor stations are constructed at an interval of 50 to 100 miles along the gas pipeline and consume about 3 to 5% of the total natural gas transported [43]; [44]. This help to minimise pressure loss along the pipelines by boosting the reducing gas pressure. However, Tukur [45] presented that the effective operation and location of the compressor station depends on the MAOP, topography and flow rate.

Compressor stations (CS) are gas pressure pipelines reloading point, and it accommodates suitable sized gas compressors for the transportation of gases. Flows and pressures are related in any given application that employs gas compressors as well as some other factors. This influences the compressors stations arrangement, as well as the type of equipment, used[46]. The impact of distance between compressor stations on availability, fuel consumption at compressor stations along a pipeline and the gas flow rate was carried out by Kurz & Lubomirsky [16]; [46]. It was concluded in their work that the more the compressor stations installed along the pipeline, the lesser the resistance offered by the pipeline, and vice versa. However, designing the required GC with appropriate prime mover can help in minimising the pressure loss without multiplying numbers of CS.

There are different types of equipment that forms an integral part of the compressor station. This equipment includes but not limited to compressor units with its driver, cooling facilities, scrubber/filters, emergency shutdown system, onsite computerised flow control and dispatch system and SCADA [44]. These equipment uphold the operational integrity of the compressor stations. Other components of compression station include mufflers fuel; lube oil system, backup generators, and gas system. All these equipment have various responsibilities for the smooth operation of the compressor station. For instance, Figure 2-7 shows the schematic of the major

hardware of the compressor station indicating the driver, driven and the accessories drive.

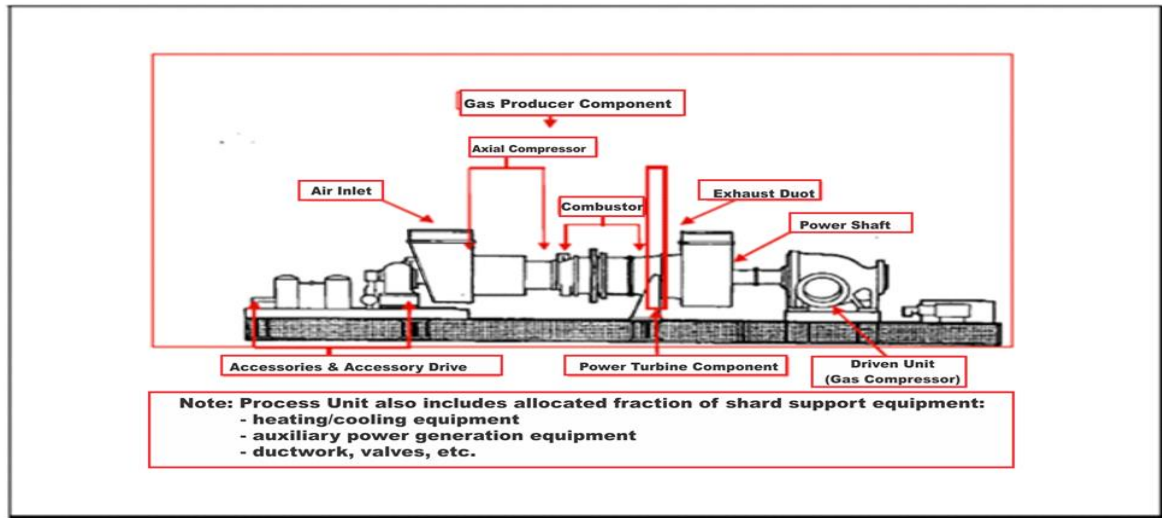


Figure 2-7 Schematics of GT compressor station[47]

Figure 2-8 showing the streamline of gas entering the compression at point C, first of all, passed through the scrubbers and filters at point D. the primary function of the scrubber and filter is to remove both solid and liquid contaminations and permits clean gas to pass through the smaller piping assigned each compressor located point E. The gas volume is re-pressurised by the compressor units to gas cooling facilities at point H. The temperature and pressure increase as a result of gas compression. In order to protect the inner coating of the pipeline and to increase the efficiency of transmission, the gas is allowed to cool at the cooler unit before returning to the mainline pipeline.

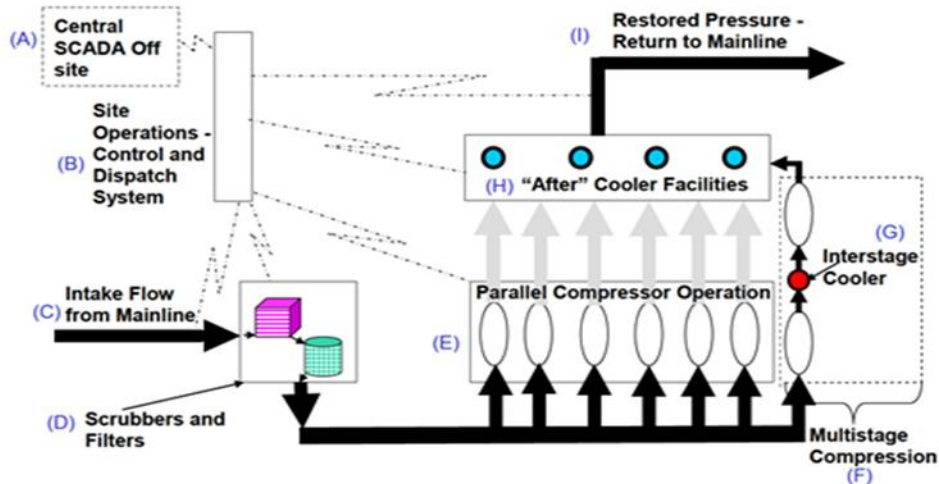


Figure 2-8 Generalised compressor station schematic[44]

Figure 2-9 also shows another generalised schematic diagram of a compressor station where the gas passes through all relevant section such as valves, scrubber, etc and finally leaves the cooling system into the main pipeline for gas transportation. Figure 2-9 equally shows another schematic diagram where an anti-surge controller is incorporated in the compressor station

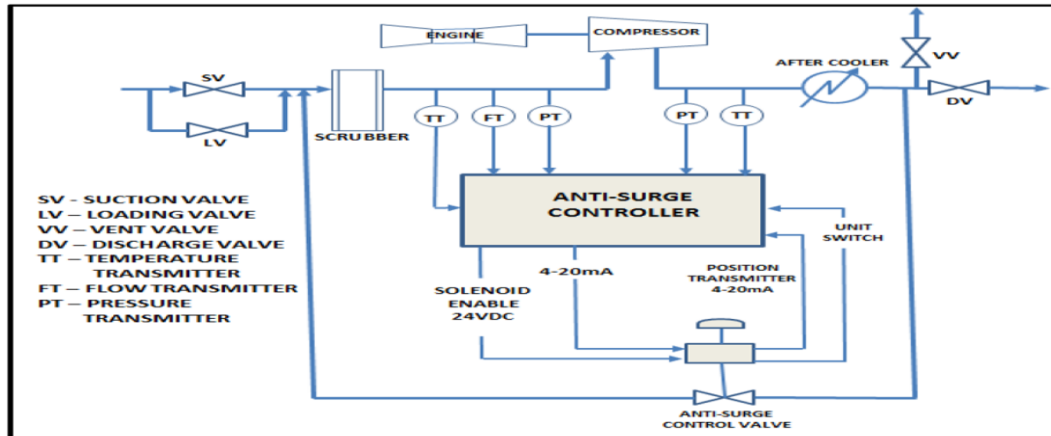


Figure 2-9 Schematic diagram of a Compressor Station [48]

2.6.1 Arrangement of Compressor unit on Natural Gas Pipeline

Arrangement of the compressor on a long-distance pipeline is very important in natural gas transportation on a pipeline system. Santos [49] presented that the arrangement could be either a series or parallel connection. These are two possible configurations acceptable in the oil and gas industry. Santos [49] and Kurz *et al.*, [50] addressed the advantages and disadvantages of adopting a particular type of arrangement, whether series or parallel arrangement with a conclusion that was not comprehensive. These configurations usually are executed based on economics and simulation of failure analysis[49]. It is mostly assumed that a parallel arrangement of compressor yields a good flow if one unit fails. Nevertheless, Ohanian and Kurz [51] analysed two identical compressors in series and presented that series configuration produces a better flow compression than that of the parallel arrangement. The series arrangement was disputably admitted since the relationship between flow in pipeline and pressure ratio at the compressor station is dictated by pipeline hydraulic (pressure). In parallel compressors configuration, if one compressor fails, it forces the remaining compressors to operate at or near choke with very low-efficiency

due to flow limitation range[20]; [52]. Figure 2-10 shows the parallel arrangement of compressors. Furthermore, the compression ratios of a parallel arrangement of compressor units are usually the same; the better thing here is that it can handle a massive volume of gas. Unlike the series configuration, the temperature of the gas discharged from the parallel arrangement will not be high since it does not undergo multiple compression stages. In parallel configuration also, the discharge gas temperature of the individual compressor is the same as that of a single compressor with the same compression ratio.

The compressor operates within the limit of the temperatures not exceeding that of the pipe coating material. Gas temperature not exceeding 60/65°C is generally required of the pipe coating materials [42]. There is a considerable increase in discharge gas temperature with increase in compression ratio. In order to actualise an elevated compression ratio, it is very vital to have multi-compression stages, and this will ensure that each stage of compression does not exceed the acceptable range of compression ratio. Generally, if some number of compressors are installed in series to get requisite compression ratio r , each of the compressor stages will operate at a compression ratio of

$$r = (r_i)^{\frac{1}{N_c}} \quad (2-4)$$

Where r , stand for compression ratio, r_i – Overall compression ratio; N_c – number of compressors in series. Reduction in power requirement is achieved by using identical compressors in series to deliver the overall pressure.

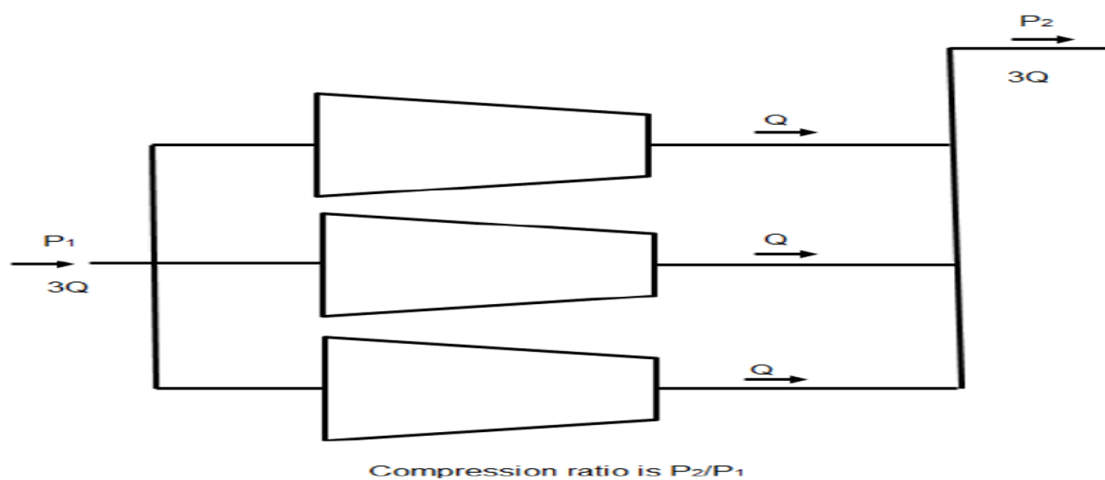


Figure 2-10 Parallel Configuration of Compressors[10]

- **Series Arrangement**

In a series arrangement, the operation is usually with high efficiency as the operation is far from the choke region. In series arrangement also, the failure of one compressor does not affect the performance of the other as the remaining compressor operates with better compression and efficiency to supplement the failed one when compared with the parallel configuration[51]. Figure 2-11 shows a series connection of three compressors. In this arrangement, each unit compression has an equal amount of gas but may have different pressure ratio. It is during stages of compression that the overall pressure ratio is achieved.

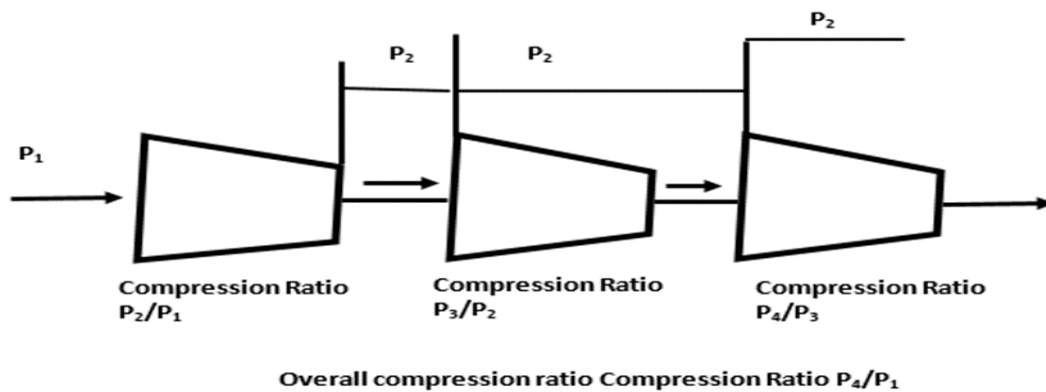


Figure 2-11 Compressors in Series[10]

It is interesting to note that in some cases, intercoolers are incorporated between two stages, this is because of the compressed gas as represented in Equation (2-5)

$$T_2 = T_1 \left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} \quad (2-5)$$

In a multi-stage gas compression system, the intercoolers serve two functions:

- It reduces the temperature of the gas at the inlet to the next stage. If the temperature were not reduced between stages, the compressed gas would exceed the temperature limits of the compressor equipment, resulting in mechanical damage.

- ii. To allow the condensation and separation of any heavy gas components and water vapour in the gas stream. Gas routed through an intercooler partially condenses to a liquid that is then separated by gravity. This reduces the workload on

Since the throughput of the gas reduces with flow temperature, elevated temperature of the gas is not desirable. During the early stage of pipeline design, gas compressor arrangement and its frequency along the pipeline is always a concern[16]. As earlier stated, gas compressor and prime mover which have a simple control system is an integral part of the high-pressure natural gas transmission system. The centrality of pipeline performance is hinged on the loading requirement in term of station frequency with consideration on the performance of their layout arrangement. Lubomirsky et al. [53], presented in their work that in each compressor station, the number of compressors mounted along a pipeline network has a substantial impact on, fuel consumption, the capacity of the network and availability.

Conclusively, In deducing the answer and conclusion from the questions that which configuration is better than the other, several researchers like Kurz,[14], dosSantos[110] and Santos[125] analyses the pros and cons of series and parallel configuration in a compressor station. Their analysis was centred on the economic analysis of the two configurations. They presented that the series arrangement of the compressors enhanced a better result than the parallel arrangement when there is no concern for standby units. However, parallel units give better results than the series configuration when there are the need and a condition for a standby unit. It is also reported that parallel units are better when more flexibility is desirable and can move a large volume of gas. Based on the above reasons and the nature of the TSGP project, the parallel configuration was adopted in this research.

There are various classes of compressors used in the industry. which is represented in Figure 2-12. They are the rotary, positive displacement and dynamic compressors. All types of gas compressors belong to these classes of compressors. Most compression systems make use of the centrifugal compressor, which belongs to the dynamic class than others, and the centrifugal compressor was adopted in this research for the CS. However, the flow required to be compressed, the density of the gas in conjunction with the total head and the duty which has to be performed will

determine the choice of the compressor to be employed for the compression process[56].

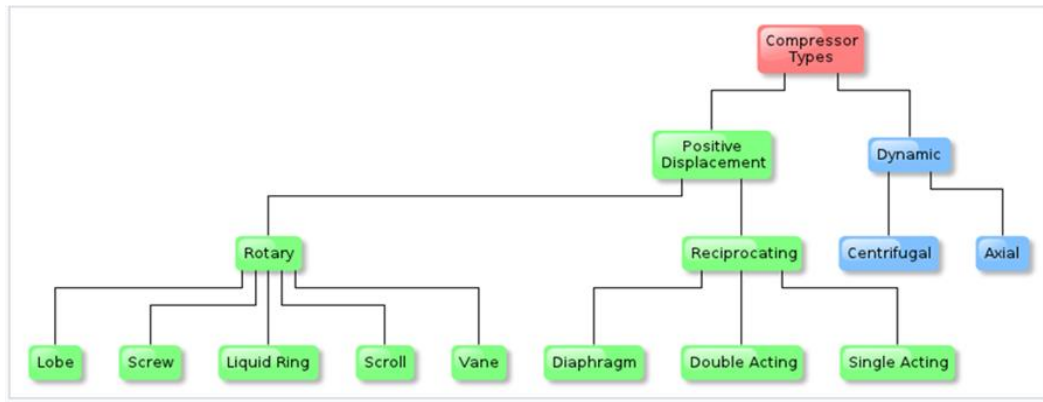


Figure 2-12 Types of compressors [57]

When comparing the centrifugal compressor with other types of gas compressor, the centrifugal compressor has a wider operating range with good operating pressure ratio (PR). In facts, Boyce [58] affirmed that centrifugal compressors are better used in gas pipeline transportation and process plant because of their extensive operating range and an appropriate PR as represented in Figure 2-13

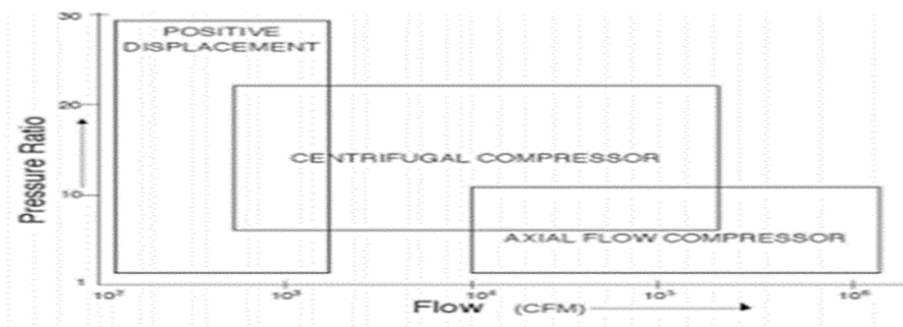


Figure 2-13 Range of application of different types of compressor[58]

When considering the performance of the GC in a CS, Bloch [59] developed a curve to deduce the head coefficient and the polytropic efficiency as a function of inlet volume flow or flow coefficient. A value of flow coefficient within a range of 0.01 to 0.14 is usually assumed to estimate the value of polytropic efficiency and head coefficient [60]

2.6.2 Thermodynamics of Gas Compression

Through mechanical work, the natural gas is moved from a suction pressure to higher discharge pressure. There are three ideal processes used to describe the movement of gas in a system. This includes isothermal, isentropic and polytropic compression[61]. When the temperature is kept constant during the compression process, then the process is regarded as isothermal compression. If no addition or subtraction of heat from the system, then the process is regarded as an isentropic or adiabatic reversible process. Also, it is a process in which the entropy of the system remains the same throughout the compression process. However, from a practical point of view, this is not feasible. Again, any reversible process on any open or closed system of gas which involves both heat and work transfer, such that a specified combination of properties was continuously maintained throughout the process is used to describe the polytropic process. In summary, the polytropic process is a non-adiabatic reversible isentropic cycle [62]

Although for Isothermal process, there is a constant temperature, however, the pressure and volume of the compressed gas vary from inlet to outlet as represented in the ideal gas equation (2-6)

$$P_1 V_1 = P_2 V_2 \quad (2-6)$$

Where P_1 and P_2 are the respective inlet and outlet pressure while V_1 and V_2 are inlet and outlet volume, respectively. In the case of isentropic compression, the discharge temperature is determined by the pressure ratio as represented in equation (2-7)

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1} \right)^{\frac{\gamma-1}{\gamma}} \quad (2-7)$$

Where T_1 and T_2 depicts the inlet and outlet temperature conditions respectively and γ stand for the ratio of specific heats $\frac{C_P}{C_V}$.

Polytropic compression is represented in equation (2-8) which is related to adiabatic compression of the discharge temperature. This process satisfied the real process of compression.

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \quad (2-8)$$

Where n represents the polytropic factor

2.6.3 Compressor Station's Economics

The economics of compressor station evaluation depends on some vital constituents like capital cost, emission tax, operating cost, and fuel cost, which result in the LCC. The types of gas compressor and the driver has a significant impact on the overall cost and fuel consumption[20]. Kurz et al. [63] established that gas compressor operating range and efficiency in addition to its drive are known to be the essential performance indices for the evaluation of the economic assessment of the compressor station. Fuel consumption during the compression stage to move gas from suction pressure to discharge pressure is hinged on the efficiency while the operating range has to do with the range of possible operating conditions in terms of flow and head at an acceptable efficiency within the deliverable power of the driver[52]. Higher revenue with lower fuel consumption on the compressor stations results from the higher operating range with higher efficiency. The economics and environmental aspect of the compressor station also comprises of the cost of emission consequence of methane and the exhaust of the turbine driver. Kurz et al. [52] presented that, methane is usually measured about 20 times potent greenhouse gas as CO₂. Hence, 1Kg of methane correspond to 20kg of CO₂. They also explained that the station availability to operate the required hours per year and the entire life of the project play a significant role in the compressor station economic evaluation. The cost related to availability analysis is the failure of the compressor's station to operate for some specific period and as a consequence, not earning money. The loss of income can be as a result of reduction in pipeline throughput, transportation tariffs, higher operating cost or consequences of not meeting up contractual agreement.

In order to achieve the cost of GT engines for any capacity, Gas turbine world handbook [64] presented the trend for a simple cycle gas turbine. The trend shows the relationship between the price (\$/kW) and shaft power (MW). According to the handbook, it was an intention to make the analysis a helpful guide for project planning

and cost estimation. In facts, based on this report, Person Brinckherhoff[65] expressed that using this relation correctly and for a basic application, its accuracy can reach $\pm 10\%$. Also, in order to achieve the cost of the pipeline accessories, Menon [22], stated that the costs of the pipeline accessories are a function of the pipeline initial project cost (PIPC). In this research, the PIPC costs include the costs of pipelines, GCs and GTs.

The O&M costs usually consist of two components, such as fixed costs and variable costs[66]; [67]. However, this research shall include a major maintenance cost (TET based costs) as the third components of O&M cost. As used by Obhuo [68] in his PhD work to analyse the maintenance costs of GT engines. Meanwhile, the fuel cost was not added with the O&M costs. It was considered independently. It is important to state that the fuel cost and O&M cost is attributed to only the GT operation.

Many practical conditions where the knowledge of fuel consumption on the pipeline network is important was seen in the work of Sekirnjak[69], Wu et al[70] and Krishnaswami[71] where they modelled the fuel consumption of centrifugal compressors to apply in a pipeline model. Again, Santos et al. [72] analysed the issue of fuel consumption from the perspective of economic investigation of the pipeline network. All the above researchers confirm the fact that an accurate fuel consumption model plays a significant role in simulating natural gas pipeline performance. The same viewpoint was verified by the research of Kurz and Ohanian [73]. In fact, Fabio et al [74] developed a model to estimate fuel consumption in a compressor station using the life data at the pipeline control room. Their model was based on a two-shaft engine and utilised the correlation that exists between fuel flow rate and air inlet temperature and the speed of the axial compressor of the turbine. Again, kurz and Bruns [75][17] showed in their work that fuel cost constitute more than 50% of GT operational life cost. In this research, a thermodynamic model was developed to quantify the fuel generated to assess the cost.

2.7 Prime Mover Option for Compressor Station

There are many prime movers or drivers used in the pipeline system. These include; steam turbine, motor, gas turbine and reciprocating engines.[76]. Wash and Fletcher[77] explained that the selections of these drivers depend on some consideration which includes; fuel or energy cost, life cycle cost, reliability, excellent

thermal efficiency, availability, low weight, and realistic part power torque. Kurz, R., Mokhatab[78] also explained that in terms of installed cost, the gas engine driver is not competitive at the power level demanded by high-pressure pipeline transmission network. Smalley et al. [11] finalised that gas turbine engines are widely used in gas pipeline transmission since they are suitable for driving centrifugal compressors, especially in terms of reliability and availability. Also, the gas turbine operating range matches that of the centrifugal compressor, which is between 50 -105% of the compressor rated speed [79]. To affirm the suitability of GT engine with centrifugal compressor, Horowitz[80] investigated the operating comparison between steam turbine, gas turbine and electric motor. He confirmed that gas turbine has a wide operating range as compared to other drivers which make it suitable to drive centrifugal compressors

Gas turbine and electric motor are the two main competitors in the prime movers' application for pipeline compression today[10]. The considering factor which is economic-based in selecting between the prime mover is the available energy to power the installation. The condition that needs the building of long-distance of electric line to the site of the project must involve a thorough economic analysis. This will enable the project owner to either use the electric motor or not. From the reasons above, an electric motor cannot be a feasible choice for a system characterised by electric outages. Taking into account of all the factors mentioned above and based on the work of Horowitz[80], the gas turbine engine fit into the criteria for this research. Therefore, the present study will consider only the gas turbine engine for the pipeline compression of natural gas.

2.8 Gas Turbine

A gas turbine engine is indisputably one of the most significant intervention in the 20th century and has improved the wellbeing of the society ranging from industries to homes in the world today. Because of the gas turbine usefulness and a further improvement in its application, gas turbine engines are utilised in aircraft propulsion, off-shore application and land-based [81]; [82]. In this research, it is used for mechanical drive application. When used in a pipeline for compressor drive there are certain conditions such as the power requirement of the compress that must be met.

Lubomirsky et al [53], presented that gas turbine drivers are selected based on the maximum power requirement at the highest ambient temperature. A margin will then be applied to provide for the entire pipeline operation and variations. Usually, gas turbine engines are selected based on ambient locations. The condition of selection is very significant because the efficiency, power output, and the fuel consumption at the designed point will vary based on the operating condition. Therefore, it is vital to select a gas turbine that will provide the power requirement of the gas compressor. The present study utilised a time-based ambient temperature variation for the GT operation at each compressor station. This temperature corresponded to the time of day.

In the gas turbine operation, the critical turbomachinery component is the high-pressure turbine (HPT).[83]. The HPT is critical because it operates at high shaft speed with a correspondingly high temperature. This operation increases stress acting on the turbine blades and disc. Clara [84] explained that the hot section is prone to creep deformation, mechanical and thermal fatigue when operated at elevated temperature. This occurrence can lead to a reduction in life expectancy rate and component failure [85]. Gas turbine blades failure affects other components of the engine, which can lead to complete failure of the engine if any preventive measure is not taken. Therefore, careful operation, proper evaluation of engine life and application of life extension technologies for safely extracting maximum usage of the engines can help reduce unsafe incidents and early failure in the engine operations[86].

2.8.1 Gas Turbine Degradation

Wear and tear that are generally associated with all prime movers working overtime are equally applied to a gas turbine operation. The gas turbine, as a modular design engine functions with the aid of the co-operation between different components of the system. The combination of the various turbomachinery components makes up the entire gas turbine system instead of the individual component. Treating the gas turbine as a system reveals the effects of degradation on the match of the components. The entire system contributes awareness to the problem of GT degradation. It is difficult to ascertain the exact amount of performance degradation in the field.

Kurz [87] investigated the mechanism and causes of degradation and its impact on GT performance in natural gas transportation. Their main aim was to look at the factors that influence the degradation and how component degradation can affect the general performance of the engine. They concluded that the individual GT turbomachinery components affect the overall performance of the GT. Kurz and Bruns [88] also analysed some mechanism of degradation and adopted a method to simulate the effects of engine and driven equipment. It goes with a relatively simple set of equations that describe the engine behaviour and a number of linear deviation factor that can easily be obtained from the maps or test data. Igie et al. [89] used machine-generated data to quantify the level of degradation that is mostly related to compressor fouling. They proposed the concept of using handles and filters which demonstrate the level of insight to the data and forms the basis of the outcome. The work further revealed that the engine deteriorates during operation at a varying degree.

Meher-homji and Bromley[90] classified performance deterioration into recoverable, non-recoverable and permanent degradation as represented in Figure 2-14. The aftermath effect of this degradation affects the industry in terms of profit maximisation in gas turbine operation.

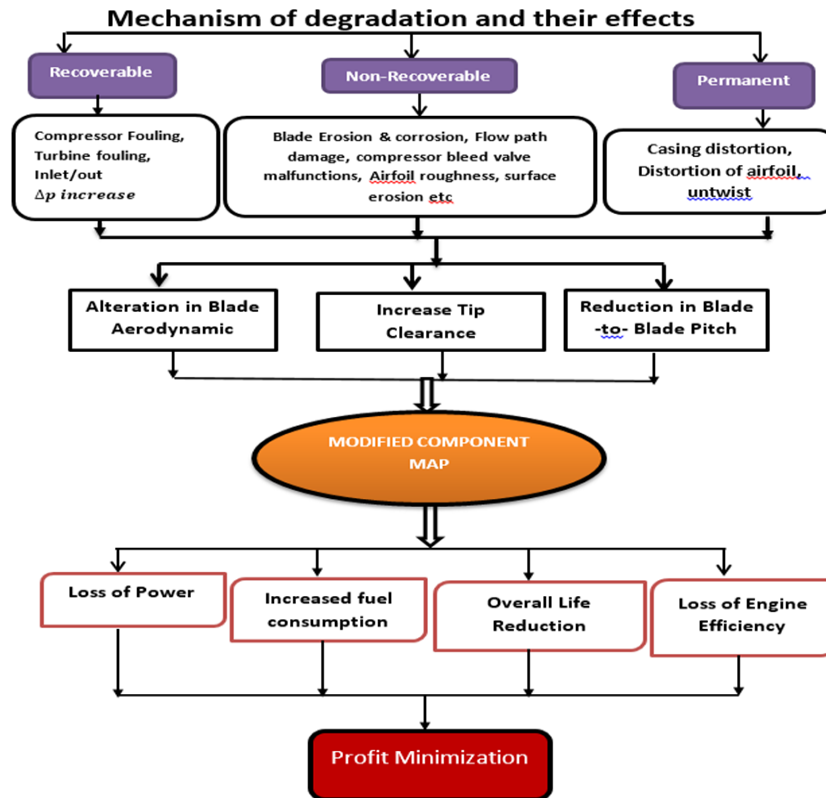


Figure 2-14 Mechanism of Degradation and their effects

2.8.1.1 Fouling and Dirts Deposit

Fouling is seen as the reduction in component efficiency and flow capacity caused by adherence of particular contaminants to the gas turbine engine airfoil and annulus surface[91]. When fouling occur, there will be material built up, which changes the airfoil shape, the inlet angle airfoil is changed, surface roughness increases, and reduces airfoil throat opening. The resultant effect is to reduce component performance.

The particles that cause fouling are usually between 0.1µm to 10µm and degrade the engine by changing the geometry of the airfoil [92]; [93]; [94]; [95]; [96]. It progresses to a worse condition as the operating time continues if no maintenance action is applied. However, most of the compressor fouling are recoverable, and these particles are removed by suitable compressor washing

Noticeably, this fouling which can be eradicated by washing occurs both in the compressor and turbine. The most common one is the compressor fouling, which has a severe effect on engine performance. Usually, during operation, 70-85% of all

GT engine performance loss is credited to compressor fouling because the contaminated air first passes through the compressor [97]. While those that caused the turbine fouling passed to the turbine with the inlet air, cooling air, fuel, fuel additives, and water/steam used for NO_x control[91]. Meher-Homji and Bromley [98]presented that fouling result in a possible increase in blade natural frequency due to an increase in the mass of blade caused by particle buildup. This leads to an imbalance that can be accompanied by vibrations. Also, fouling causes the blockage of the turbine stator blade cooling passages since the compressor bleed air is utilised. This is usually caused by small foulants less than 5µm[98]. A typical fouled blade is shown in Figure 2-15 and the work of Meher-hormji and Bromley [98]

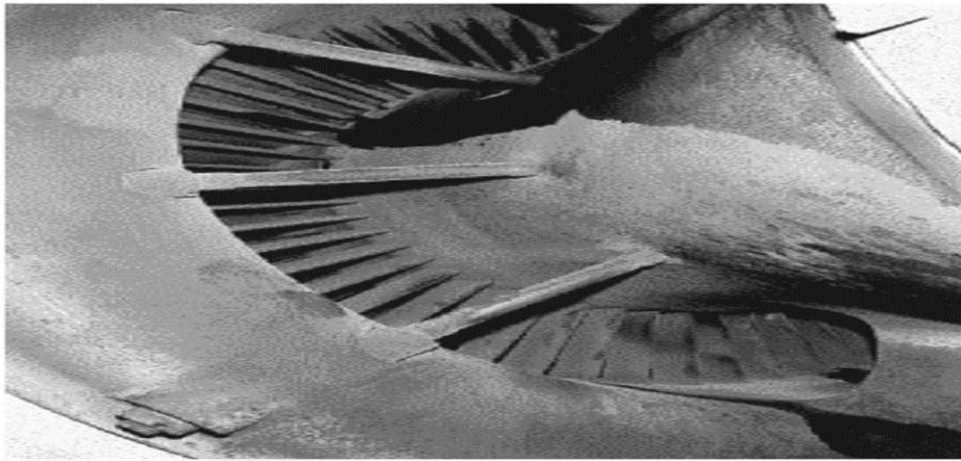


Figure 2-15 Degraded GT air inlet Bellmouth and blading[98]

When the first stage of the compressor is fouled, it reduces the amount of airflow that enters the engine, which subsequently affects later stages performance. The fouled stage has a different operating point from the clean one [99]. The range of the gas turbine power output loss due to fouling is from 2% under favourable conditions, 15-20% under adversarial condition[100].

2.8.2 Degradation of Turbine Blades

Generally, blade material and vane in gas turbine applications are subject to degradation during service by the combined effects of the condition of the environment, mechanical stresses and temperature[101]

Figure 2-16 shows the macroscopic features of a damaged in the as-received condition. It is observed here that the airfoil was severely distorted and that fracture

occurred along the section of the blade normal to the direction of maximum tensile stress. Meanwhile, Part “a” shows the view of the airfoil showing the macroscopic distortion while part “b” indicates the view of the lower section of the fracture surface signifying the cooling air channels. Generally, corrosion, High-temperature oxidation, Erosion, foreign object damage, and abrasive are mechanisms that cause degradation in GT engines. [102].

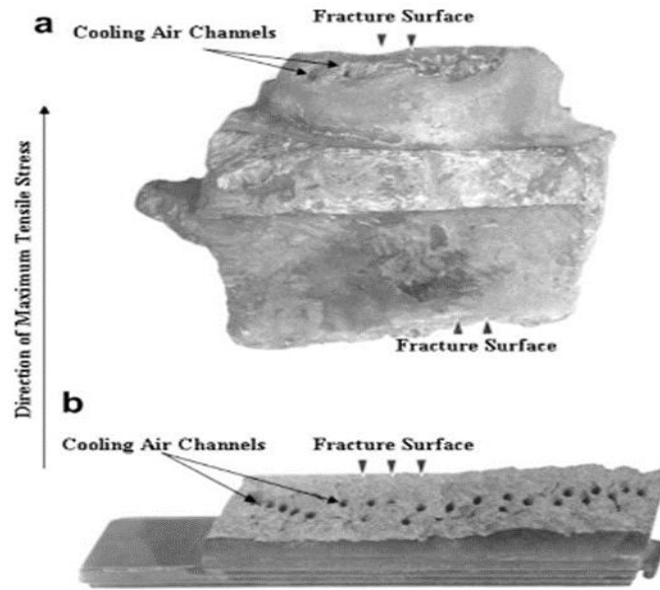


Figure 2-16 Photographs showing the fractured blade[103]

From the review of the previous study, different mechanisms of GT degradation has been identified. However, the present study considers fouling as the mechanism of degradation to be considered in this research. Usually, both the compressor and turbine component experience fouling[104]. The most common one is the compressor fouling, which has a more severe effect on engine performance.

Furthermore, Fouling and erosion have been studied to show their effects on thermal efficiency and the power of the engine. It involves the accumulation of dust, which reduces the tip clearance and increases the surface roughness of the blade[88]. Those that cause the turbine fouling passed to the turbine with the inlet air, cooling air, fuel, fuel additives, and water/steam used for NO_x control[91]. Fouling decrease the compressor isentropic efficiency and the flow capacity[105].

2.8.3 Gas turbine Degradation and Evaluation Methods

Fault and degradation in GT is an ancient and modern problem, and the quest to evaluate and solve this problem has been a concern to researchers. In the early time, to better identify a fault in the engine during operation, Aker [106] used a trend plot to map the deviation of engine parameters from the manufacturer's specified baseline. However, this type of study generally did not offer sufficient information to specify what composed the fault. Also, Urban [107] recommended a linear based method where the changes in the performance characteristic of the engine component were associated with the measurement parameters changes using vital thermodynamic performance equation specific to a particular engine. Observing the drawback in the use of trend line plotting and linear analysis approaches, Saravanamuttoo [108] adopted a nonlinear approach hinged on cycle deck analysis to be used with trend line analysis. Saravanamuttoo et al. [107]; [108] analysis show changes in critical parameters of the engine through a relationship with changes in measurement parameters. Notably, Saravanamuttoo[108] adopted the use of engine models as a predictive instead of being a sole diagnostics tool using models of faults constructed as combinations of efficiency and flow capacity changes. The same method was adopted by Lukachko & Waitz [109] where cycle deck calculation with generic component maps was utilised, and the degradation of the engine was modelled through changes in component efficiencies and flow capacity. In the quest to mitigate the lack of rigorous connection with actual deterioration circumstances, current efforts have geared on developing models of particular fault that can be verified, primarily for the case of compressor fouling. Although Lukachko & Waitz [109] provided only a representative analysis of the impacts of components degradation on NO_x emission,

Garcia and Brun [110] Developed a computational pipeline model that was used to simulate a two-loop pipeline system of a distance 218 miles of rippling terrain. The aim of the investigation was to maximise gas flow while energy usage is minimised. The author analysed five separate cases that centre on the number of units and compression power and concluded that among the five cases, two were optimum. In their analysis, they were silenced on how the machines degradation could affect their decision.

Similar approach but different application was used in this research where a thermodynamic model was created using TURBOMATCH. “Compressor and Turbine” were degraded as a single type of degradation were some percentage of reduction in the flow capacity and component efficiency of the turbomachinery (compressor and turbine) were implanted in the engine model for each year throughout the project life (thirty years). Its overall effect on the engine is presented and interestingly, how this overall effect could be used to investigate the techno-economic of the natural gas transportation at a controlled TET is investigated.

2.9 Engine Maintenance Action: Compressor Washing

GT engine is expected to get rid of any interference that impedes on the performance to achieve the desired output performance at constant engine operation. Hence, the need for compressor washing as one of the maintenance action for GT engines. Compressor cleaning was adopted to wash away the deposited particles to recover the performance loss due to fouling as shown in Figure 2-17

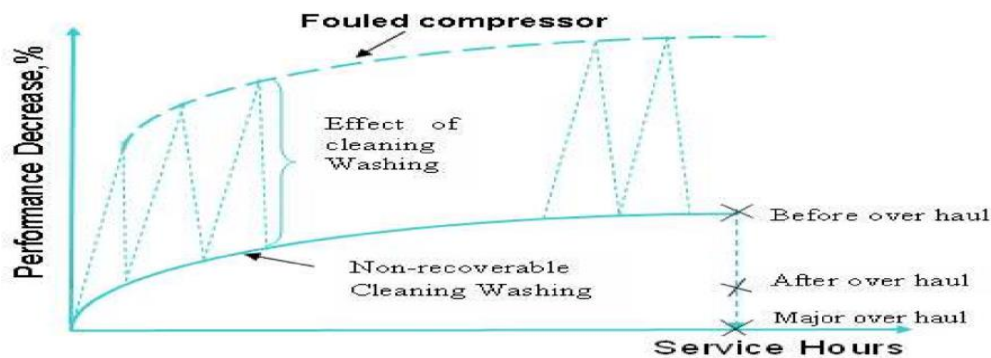


Figure 2-17 Effects of compressor washing [111]

Over the years, different methods of washing have been adopted to clean the degraded compressor due to fouling with their respective pros and cons. These methods include:

2.9.1 Manual Procedure

Manual method is one of the approaches of compressor washing that has proved potent. The engine has to be disassembled and cleaned manually with brushes and detergent. The method is effectively executed by cleaning the IGV by hand, which bring benefits to the industry, but the limitation here is that it involves lots of labour, downtime, and cooling of the engine, which is rarely needed, especially for oil and

gas application[112]. However, Abdelrazik and Cheny [24] and Luesden et al[25] further analysed that the level of power recovered during manual methods cannot be achieved with any other methods. The manual cleaning procedure is shown in Figure 2-18



Figure 2-18 Manual cleaning of the compressor [113]

2.9.2 Abrasive of Grit-blasting methods

An abrasive method known as a grit-blasting method is used in the industry today. In this method, the particles were get rid of by injecting rice, charcoal, nutshells, or synthetic resin particles into the air-steam of the operating engine. Since high-speed air was needed to move the particles in order to achieve the necessary impact, the process has to be undertaken during engine operation[114]. The disadvantages for this approach is that the last stages remain contaminated, and is necessary to remove the contamination so that the seal and the internal passage of the secondary air system of the engine can remain uncontaminated. It means that the removal of oily deposit was not entirely satisfactorily done[115]. Brittain [116], concluded that despite the demerits, the methods were widely used for an aero engine. However, Godon [117] presented that this method was discouraged, as the new turbine blades in the late 70s/early 80s came with a series of holes for cooling. Also, Boyce and Gonzalez [118] also explained that some of the abrasive also tend to shatter and get into the seals, bearing, and lubrication system

2.9.3 Off-line and On-line Compressor Washing

A better and effective method to clean the deposits than the abrasive cleaning methods is the liquid injection in the form of demineralised water with or without the

mixing of detergents[119]. This method has proven to be the leading applied method. In this method, two processes are involved: **on-line** and **off-line** application.

Off-line compressor washing some time called soak or crank wash involves a shutdown of the engine and executed unfired, with the engine been turned by the starter motor[120]. The gas turbine rotational speed is on the order of 20%[121]; [122]; [123] to a limit of 30%[25] of the usual operational speed, resulting in a reduced flow of air into the engine. For a more massive engine, the downtime increases[95]

Even if it is a preferred method now for the aero engine, early concerns take in to account the dispersive characteristics of the injected fluid concerning the component to be washed[121]; [123]. Liquid solvents that chemically attack not only the aluminium contaminants but also the parts of the engine that made up the same materials, causing chemical erosion should be taken into consideration while applying washing fluid. This method (crank washes) is very efficient, and the recovery of lost power is close to the first level or the level reached after overhaul[119]. Nonetheless, in term of techno-economic of the system, the downtime results in loss of revenue for the business owner.

On-line washing: Further to the idea of implementing compressor washing using liquid injection, on-line washing, also called fired washing, is embraced. Since on-line washing is performed fired, generally under full load with no or little reduction of the unit's capacity and speed, it is considered to be the most advanced method [124]; [125]

Online washing could best be defined officially in the instruction sheet of the Association of Power and Heat Generating Utilities (APHGU)[126]. Although the benefits are not as efficient as the off-line method, it is recommended that the frequency of daily to once in a month and durations should be between 5 and 30 min. This process necessitates a permanently installed washing device (which indicates both off-line and on-line processes) as shown in Figure 2-19. It also accommodates the opportunity of a more frequent washing arrangement when the engine is in operation. Demineralised water, surfactants, and solvent are most customarily utilised when cleaning with fluids for the injection from a spray ring upstream of the compressor[123]. Some cleaning agent can be seen in the work of Lambert et al.

[127]. Meanwhile, the OEM provides the washing device and can be installed or replaced by a retrofit washing device from a supplier.

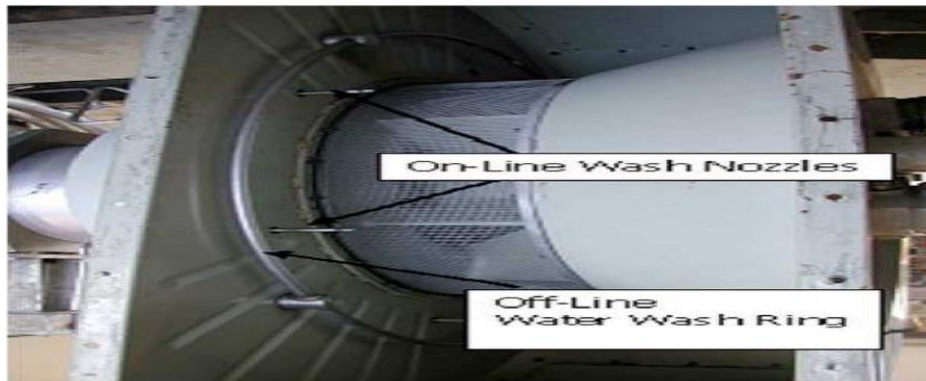


Figure 2-19 Description of online and off-line washing device[128]

It is worthy of note that online washing has substantial economic benefit than the off-line (crank) washing when considering the downtime loss associated with the off-line method on natural gas application. The reason is that the engine does not need to shut down before applying the method. For this reason, it can conveniently be applied in the oils and gas industry where the operation is not allowed to shut down due to the economic benefits associated with oil and gas business. However, online washing can be counterproductive as some contaminant can move downstream of the compressor, causing more fouling to the engine. Also, an excessive volume of water can result in erosion, thereby causing more harm to the engine. However, frequent online washing with detergent is allowed to achieve the washing away of any portion of insoluble compounds at any given time[98]; [129]

In order to investigate how much power that could be recovered after compressor washing, Igie et al. [130] demonstrated the blade aerodynamic effects of fouling through experimental compressor cascade test and also implemented on-line compressor washing. The implemented on-line compressor washing was to ascertain the level of recovery of lost power. In their analysis, they applied demineralised water for 10minutes, 0.2% liquid-to-air ratio. 30% recovery of the lost power due to washing was achieved from the simulation of the degraded and washed engine. Again, Leusden et al. [24] investigated both on-line and off-line compressor washing on the performance of a power plant and implemented 80% recovery of lost power. The same 80% recovery of lost power was adopted by Musa [23] who

investigated the economic benefits of applying both on-line and off-line methods and concluded that both methods yield a better ROI than a single method. Further reading on the application of compressor washing for lost power recovery can be viewed from the work of Aretakis et al.[131]. Figure 2-20 shows a water cleaning device. However, Kacprzyński et al. [132] concluded that whenever compressor performance drops by 3-4%, offline crank wash should be executed

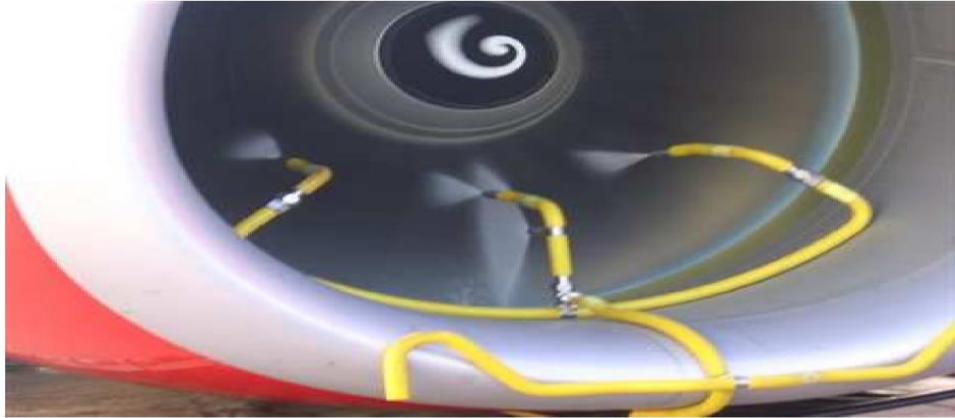


Figure 2-20 Water cleaner example [133]

It is interesting to conclude here that for better performance loss recovery on gas turbine operation, both on-line and off-line compressor washing method should be indispensably complimentary, especially for power production as shown in Figure 2-21. Nevertheless, the gas turbine for the mechanical drive in the oil and gas needs online compressor washing to help continue production. It is on this note that the present study considers fixing the shaft power of the GT engines irrespective of the condition of operation.

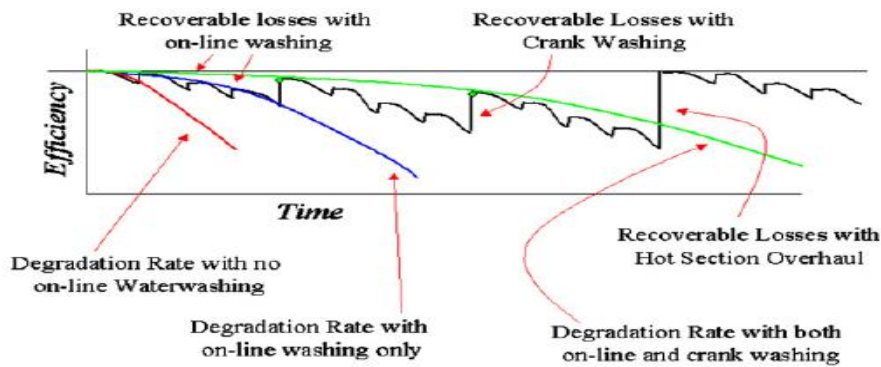


Figure 2-21 Types of Degradation Rate[132]

From the review of literature, most of the application of gas turbine (GT) compressor washing is mostly seen in power generation. Power recovery has been the focus of applying the compressor washing over the years.

Application of compressor washing for a mechanical drive on pipeline application is scarcely seen in the literature, especially an operation when the engine power is fixed throughout the operation. Applying the methods as reviewed above, the present study considers only on-line compressor washing as one of the maintenance action adopted. On-line was adopted to avoid economic loss due to downtime. The uniqueness of adapting online compressor washing to the present research is such that, instead of power recovery after compressor washing, what is recovered is the TET, fuel flow and the improvement in the thermal efficiency as the engine operate on constant power throughout the operation. The present study also intends to investigate the impacts of compressor washing on emission reduction.

2.10 Gas turbine (GT) Engine Overhaul

A gas-fired turbine in natural gas transmission service is much smaller than a “typical” unit employ in the power sector. The unit size in gas transmission service makes the engine amenable to an overhauled approach that may not be possible for other application[47]. To enhance a reliable and safe service, the gas turbine engine, like all mechanical equipment, need a planned maintenance program that is dependable with the operating environment. Such plans generally consist of frequent inspections, routine servicing, and component replacement. Engine overhaul is performed as a result of routine inspections that revealed operational or mechanical deficiencies such as high equipment vibration levels, abnormal oil temperatures, or visible damage and the deterioration of subcomponents. Anyone of these inspection results could trigger routine maintenance activities that range from minor repairs to overhauling [75].

The compressor is vulnerable to dust and particles which generally affect GT operation. Also, compressor turbine and power turbine portions of the gas-fired turbine are exposed to high-temperature combustion gases, along with vibration and thermal cycling, which result to mechanical stresses, thermal stresses, and corrosion in these sections of the turbine. Hence the need to employ engine overhaul periodically to maintain the engine from breakdown.

Beal [47] explained that inspection, disassembly, repair and or replacement of subcomponents, assembly, and test of the gas generator and power turbine component is an integral part of the overhaul process.

NSPS [134] presented that most GT engines are designed for 20,000 to 40,000 operating hours between overhaul. Also, most of the OEM such as GE, Solar, and Rolls Royce jointly represent about 87% of GT in the natural gas pipeline industry, and all three manufacturers recommend scheduled GT engine overhauls at approximately 30,000 hours of service[134]. However, the time interval varies with make, model and engine condition of operation which may extend or reduce to longer or shorter service intervals.

Given the above explanation, NSPS [134] finalised that most GT engines in natural gas pipeline transportation network are subject to overhaul at an interval of every three to five years of operation. In some cases, overhaul can take place more often, subject to operational performance and the results of inspections. Based on the above explanation, the present study implemented three years interval at baseline case. Again, since the time interval varies with engine condition of operation, the present study also implemented two years overhaul interval different from the baseline cases as one of the maintenance action for the project to compare their effects on the gas pipeline investment. According to Wulf [135], GT engines returns to operation with some level of degradation after each maintenance cycle. The returned percentage of degradation increases as the operation continues with time. This implies that as the engine continues operation after each segment of the overhaul, the unrecovered degradation increases with additional maintenance cycles[42]. However, the specific value of degradation that returns to operation after each maintenance cycle was not mentioned by the two authors work. This phenomenon was adopted in this research and also some specific values of degradation that returns to service after each maintenance cycle was assumed in this research. To understand the costs associated with this approach of engine overhaul, EPA [136] assembled eighteen compressor stations on five gas pipeline system which compared the costs for different maintenance activities to the cost to replace the entire combustion turbine process unit and concluded that overhaul costs increases between 3% to 17% of the process unit cost. Based on this report, the present study assumed that the initial cost of the overhaul (maintenance cost) was

10% less when compared with the cost of the rest maintenance schedule. Meanwhile, a specific assumption in term of performance has to be made to compensate for the effects of increasing degradation after each maintenance cycle.

2.11 Gas Turbine Emission

Carbon monoxide (CO), Oxides of nitrogen (NO_x), and other combustion byproducts of GT engines have been acknowledged as dangerous atmospheric pollutants to both human being and the environment. Nasir et al.,[137] presented that emissions are among the fundamental challenges in the development of the gas turbine to meet environmental regulations.

Carbon dioxide (CO₂) and water vapour (H₂O) has been recognised as the primary products of complete combustion of air and hydrocarbon fuels. Even though these two by-products are not categorised as pollutants, there are environmental and health implications that are associated with this by-product. In addition to these by-products, Lefebvre [138] presented that nitrogen oxide (NO_x), carbon monoxide (CO), sulphur dioxide (SO₂), unburned hydrocarbons (UHC) particulate matter (PM₁₀), volatile organic compounds (VOCs), soot and smoke are generated at varying levels subject to fuel type, engine operating and combustion conditions. All these gases are produced at the exit of the GT through the exhaust gas of the engine and could contaminate the environment when produced outside the acceptable quantity. Usually, high efficiency with corresponding high pollution is a product of high combustion temperature of GT operation. [139].

Due to the impacts of emission on the environment and human health, countries have set strict limits for emissions in new and existing power plants. For instance, the power plants in the United States are said to be the largest contributor to GHG emissions with values accounting for nearly 40% of the country's total emissions with CO₂ emissions accounted for 82% among all the pollutants from engine operation as represented in [140].

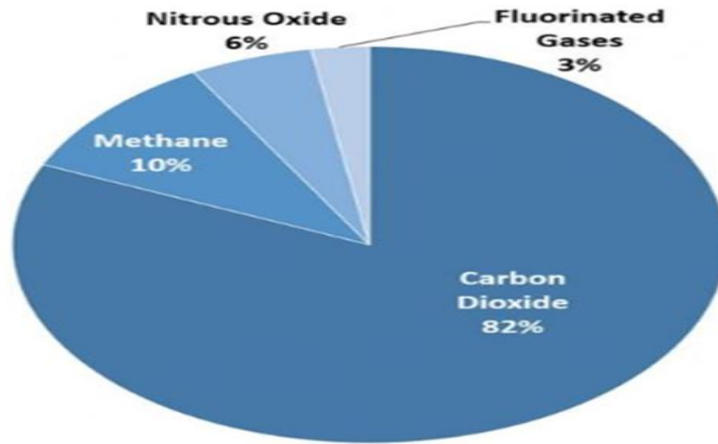


Figure 2-22 Source of greenhouse gas emission[140].

Likewise, in the United Kingdom, power stations have been seen to be a major contributor to SO_2 , NO_x , PM_{10} , VOCs, and CO. BEIS, UK[141] reported that the provisional GHGs emissions figures rely on provisional estimates of CO_2 emissions. Statistically, based on new technology in engine operation, it was observed that emission production in 2018 was 2.4% lower than the 2017 figure. The report further affirmed that Carbon dioxide (CO_2) is the main greenhouse gas, accounting for 81% of total UK greenhouse gas emissions. Many developing countries, including Nigeria, are making frantic effort to curb this emission challenge. However, the concern to quantify these emissions during operation on the proposed gas pipeline is the main focus of this chapter.

2.11.1 Correlation and empirical modelling of emission

Many empirical and semi-empirical models are currently used for the correlation of experimental data on pollutant emissions in terms of all the relevant parameters. These include design features, combustor dimensions, and operating conditions, as well as fuel spray characteristics and fuel type. Although the focus of this research is not on combustor design, this empirical model has been seen to play an essential role in the development and design of low emission combustor. They also help in permitting more accurate correlation of emission for anyone combustor than can be achieved by more comprehensive numerical models. Furthermore, the chemical reaction governing the formation of some pollutants such as UHC, smoke, etc., are highly complex. The majority of the empirical models developed so far had been

confined to CO and NO_x[138]. Based on this viewpoint, the development of the empirical expression of CO and NO_x has been achieved by many researchers.

Rokke et al., [142], investigated five different gas-fired industrial engines operating between the power range of 1.5MW to 34MW at 35-100% load and derived an empirical equation to evaluate NO_x emission which was also acknowledged and used by Lefebvre[138]. In the derived equation as represented in Equation(2-9) combustion temperature was conspicuously absent. They further, established that the influence of combustion temperature on NO_x emission is acknowledged by the inclusion of fuel/air ratio (Fa). This empirical equation was adopted in this research and used in the developed model to evaluate NO_x since the GT engines under consideration fall within the class and power range of the experimental engines.

$$NO_x = 1.46 \left(\frac{P_2}{P_1} \right)^{1.42} * M_{air}^{0.72} * Fa^{0.72} \quad (\text{gNO}_x/\text{kgfuel}) \quad (2-9)$$

It is worthy of note that some of the numerical methods are derivable from empirical analysis; hence, the use of the empirical method has been found efficacious in many engineering application, including emission prediction. Also, the general chemical equation for the complete combustion of a hydrocarbon fuel of the alkane group was used in the developed model to calculate the CO₂ emission[17]; [143].

2.11.2 Considered Emission for the Present Study and the Correlation between the Emission

CO₂, H₂O, SO₂, NO_x, CO, PM₁₀, and UHC, VOCs, soot, and smoke are the fundamental emission leaving the exhaust of a typical gas turbine engines as earlier mentioned. This research, however, focused primarily on the emissions of NO_x and CO₂ only. These emissions were selected based on current research focus and the proficiency of the intended emission model to be developed to quantify these emissions on pipeline application. Presently, CO₂ and NO_x are the significant pollutants (greenhouse gases) receiving research focus due to their contributory effects on global warming. Based on this reason, CO₂ is regarded as a pollutant. NO_x is also a precursor of photochemical smog and contributes to acidification and PM₁₀ emissions. Figure 2-23 showed the relationship between CO and NO_x emission with respect to combustion temperature. When the gas turbine combustion temperature is low, it results in slow oxidation. This makes the formation of NO_x and CO have an inverse relationship. The emission of thermal NO_x increases as the

combustion temperature increases while CO emission decreases with increasing combustion temperature. This relationship implies that the concentration of CO increases at a lower power setting and become lower as the power increases. Accordingly, NO_x emission attains maximum when the power setting is high and reduces with power reduction [138] as represented in Figure 2-24 [138]. These emissions are related to combustor inlet temperature, and pressure, primary zone equivalent ratio, condition of operation of the engine, residence time and also vary with the type of combustor.

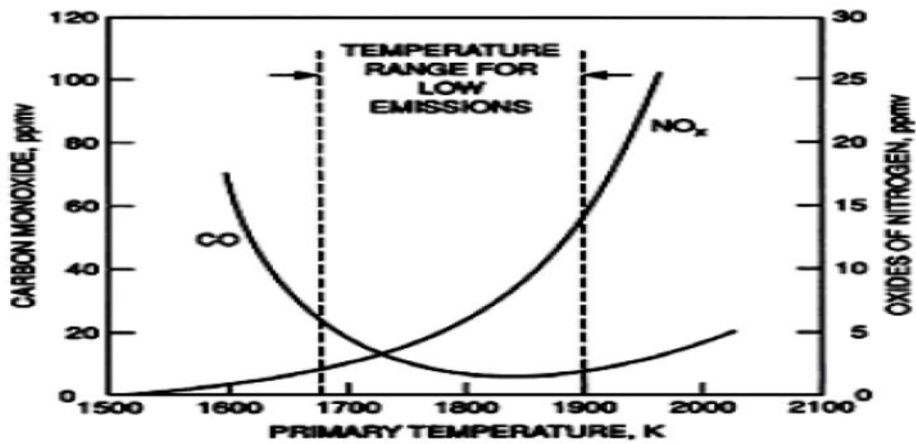


Figure 2-23 NO_x and CO relationship with combustion temperature [142]

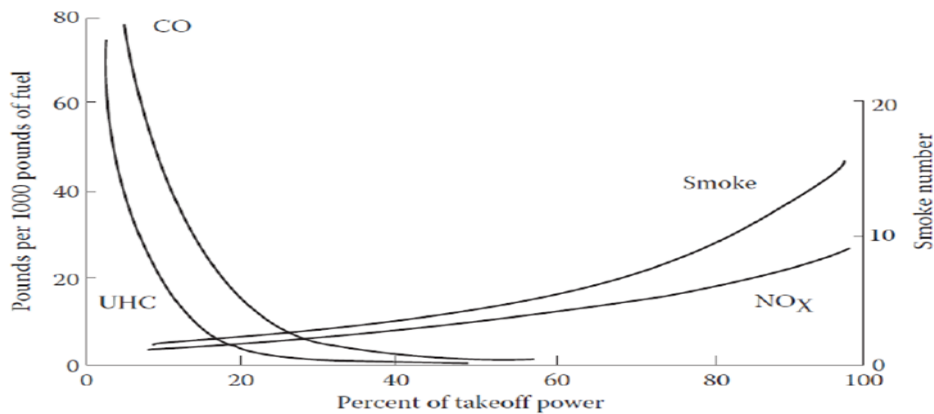


Figure 2-24 NO_x, smoke, CO and UHC emission with power setting[138]

Conclusively, the present study makes use of the developed thermodynamic model of the investigative engines, and the performance result from the model was connected to the developed emission model to calculate the annual emissions from the operation of the engines.

2.12 TERA Tool

The awareness, development and use of Techno-economic and Environmental Risk Analysis (TERA) tool were revealed by Ogaji et al. [144] at Cranfield University in the united kingdom. TERA was implemented as a result of research work carried out in areas of multi-disciplinary optimisation and management of power plant; and the environmental impact on the plant at both design stage and in operation. In their work, the TERA framework was employed in aero-engine management and modelling by integrating a commercial optimiser tool called iSIGHT, a product of Engineous Software Ltd. Its philosophy as illustrated in aircraft optimisation is shown in Figure Figure 2-25 [144]

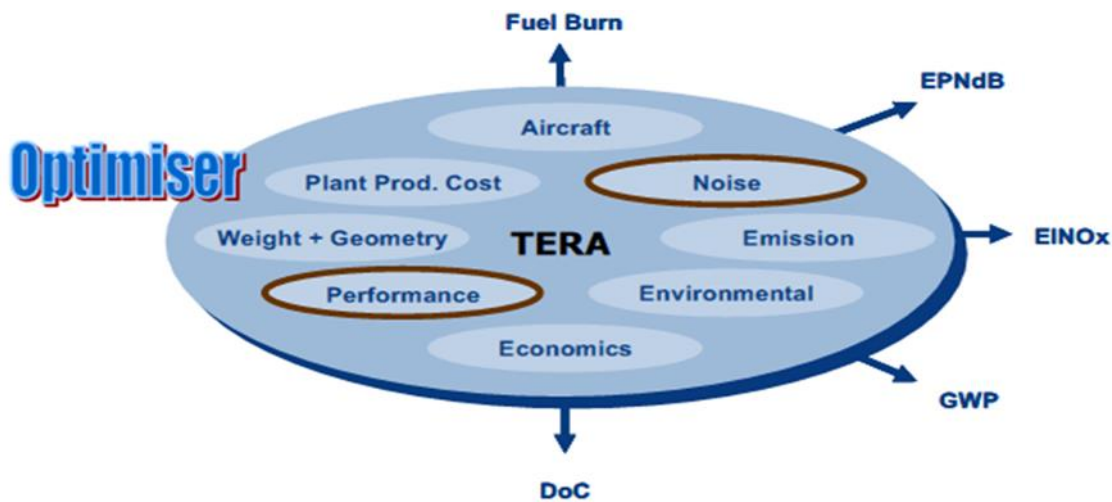


Figure 2-25 TERA philosophy and framework for aircraft optimisation [144]

TERA is an investment decision tool that looks into the excellent available option for engine performance, the impact of the environment and its associated emissions, cost and the engine availability and reliability[145].

The multi-discipline modules of TERA framework focus on performance modelling, cost, noise, the environmental impact of gas turbine and aircraft weight. In its holistic form, TERA modules comprise performance, economics, environmental, emission, weight, noise, and lifing modules. Each of these modules solves different problems, and the result is condensed into the TERA framework. The performance module occupies the central point of this tool. The reason is that it explains the thermodynamics of the component parameters and simulating the design point, off-design point, and performance degradation of the gas turbine engines. It implies that

result from these modules is fed by another module, which includes the emission, economics, risk and environmental. These are all analysed from the performance module. In fact, according to Ogaji et al. [144], TERA is a veritable tool to select and rank the excellent options for investment and to identify any risk involve and efficiently manage the resources allocation.

2.12.1 TERA Application and progress beyond state-of-the-art

Di Lorenzo et al.[146] applied the TERA concept for the European project for aeronautics research as part of the effort to meet the advisory council for aeronautics research in Europe. The aim was to investigate advanced aircraft engine concept for an environmentally friendly aero engine and their economic implication. Quintero et al. [147] applied TERA concept to illustrate the trade-off between the operating cost and the environmental impacts of future aero engines for short-range aircraft. All researchers found TERA a veritable tool for effective project implementation in terms of techno-economic analysis.

TERA has been explored both for marine and industrial application[146]. Gayraud [148]; [149]; [150] investigated industrial gas turbine for power application and used TERA to establish the basis for a decision support system for combined cycle systems. Maccapani et al. [151] detailed their investigations into TERA where they created a TERA based tool to help the choice of suitable driver for LNG. They concluded that among the four GT engines considered in the analysis, the one with the highest power rating serves as the best solution while the lifing and risk issue depends on the firing temperature. However, the cost of operating the engine on degraded mode with time was not considered. Nkoi[152] applied the TERA technique for GT engines for a petrochemical industry for 25 years of operation. In all the TERA application above, the annual engine degradation was not considered. EL-Suleiman [153] explored the use of TERA on CO₂ compression power equipment where he developed and implemented variflow to an existing GT simulation tool in order to improve the off-design performance. He analysed the economic viability of the project using the NPV approach with the aid of a developed model for 25 years. Again, engine degradation was not also considered in his analysis as he considered a project life of 25 years. However, GT engines degrade with time for real engine operation.

Relative to previous studies on the application of TERA, much research has not been explored in applying the TERA tool on natural gas pipeline transportation system. The only two known work on TERA application on natural gas pipeline transportation as at the time of this research was the research work of Nasir[17] and Tukur[18].

Nasir[17] examined the application of GT and electric motor as a prime mover on the techno-economics assessment of the natural gas pipeline under the influence of ambient temperature. The investigation presented that for an interstate pipeline, the use of electric motor is not the best option in areas where electricity supply is not available. The study further analysed the economic capability of using the gas turbine as a prime mover for 20 years. However, for the annual operation of the GT engine for project life, the influence of degradation of the engines on the project was not considered in his analysis. The study assumed that the operation of the engines was clean throughout the project life. Also, the ambient temperature used in his studies was not time-based.

Tukur[18] also investigated the application of TERA on a natural gas pipeline. In his analysis, he took account of the compression station (CS) location, power matching based on the ambient temperature variation between the GT and GC while optimising the life cycle cost. In his analysis, he assumed that maintenance and fuel cost in each year for the entire project life (30 years) remain the same. Which implies that he assumed the engine operation was clean throughout project life (30 years). Also, in his TERA application, he did not consider the cost associated with the production of CO₂ emission, which is an integral part of the TERA framework due to its environmental effects and also an important cost element in determining the life cycle cost of the pipeline project

These studies omitted the fact that all turbomachinery equipment especially gas turbine engine degrades with time and this degradation has significant effects on not only the engine performance but also on the entire project investment and benefits. Also, an in-depth analysis of the fuel and maintenance cost based on the combined effects of time-based ambient temperature and degradation in each season of the year was not considered by these researchers. In fact, those authors used only the effect of ambient temperature, which was not time-based on the GT for their analysis

and assumed that the engines remain clean for the entire project life. Therefore gap in knowledge exists which this research tries to cover.

Furthermore, to explore the entire natural gas transportation system in terms of pipeline performance, Menon [22] analysed the flow in a gas pipeline using pipeline model equation and assumed a uniform gas temperature with no heat transfer between the gas and the surrounding soil in a buried pipeline. Also, Rajnauth *et al.* [29] and Watanabe [32] analysed pipeline flow using the same concept. However, in reality, maintaining the gas temperature constant (isothermal) is practically impossible for gas flow in a pipeline system. Hence the need to investigate the non-isothermal condition in a time-based system which the present study tries to cover this limitation.

Again, the performance analysis of the centrifugal compressor can be seen in the work of Campbel[19], Kurz et al. [20] and Lars et al. [21] where the effects of wet gas and mechanical performance of the centrifugal compressor were evaluated including the method of evaluating the efficiency. However, the research work was silence on the evaluation of centrifugal compressor power in a time-based non-isothermal condition, taking into account the combined effects of pipeline elevation, mechanical losses, and the desired volumetric flow of the compressor. Also, the evaluation of gas compressor performance, taking into account the synchronisation of the pipeline and gas compressor performance is uncommon in the literature, which this study tries to cover.

Summarily, it implies that a new TERA framework that will be adapted to natural pipeline investment will be developed in this research taking into account, the various compressor station machinery/accessories, the ambient operating conditions, engine divestment, leasing and the economic viability of the project.

2.13 Economic Appraisal Methods

Economic appraisal or evaluation is the comparative analysis of two or many interventions in terms of their cost and consequences. It also serves as the assessment of a project based on their given input and the generated outcome. The most fundamental concept of this evaluation is the analysis of the costs and benefits of all the available options in the project [154]. The technique is indeed a vital process

for accomplishing and satisfying requirements for decision accountability. In summary, it is a systematic process for investigating alternative uses of resources, focusing on assessment of needs, options, funding, objectives, costs, risks, benefits, affordability and other factors relevant to decisions.

Natural gas pipeline investment needs an appraisal method to evaluate the viability of the project. The following are some of the different appraisal methods

- (i) Net Present Value (NPV)
- (ii) Internal Rate of Return (IRR)
- (iii) Discounted Payback Period(DPP)

2.13.1 Net Present Value (NPV)

The basic economic principle on investment is that a dollar today is worth more than a dollar tomorrow. The reason is that a dollar today can be invested to start earning interest immediately. The phenomenon is referred to as "time value of money"[155]. It means that the value of money for many years ahead can be brought to the present, and this is referred to as the present value (PV) of money, taking into account the discount rate. Because of this, the NPV of a project is the difference between the cash inflow and cash outflow happening for some time of the project. The cash inflow is referred to as the project benefits, while the cash outflow is referred to as cost cash flow streams. The cost stream comprises of the capital cost, annual maintenance and operation costs. The cash inflow or benefits are the cash received from the sales of goods or services rendered. The difference between the inflow stream and the outflow stream is the "cash flow" which is sometimes regarded as the "net benefit stream as shown in Figure 2-26." The NPV calculation is done by dividing the expected yearly income (net cash flow) of a project by a term which is equal to 1 plus the discount rate raised to a power equal to the project's year minus the investment cost. This is represented in Equation (2-10). The calculated method implies that the NPV takes into account the time value of money[156]; [157]; [158].

$$NPV = \sum_{t=1}^n \frac{C_t}{(1+r)^t} - C_o \quad (2-10)$$

Where C_t -the net cash-flow for the project year under consideration. r - the discount rate. C_o is the initial cost of the project. While t and n is the year considered in the

project. Generally, the higher the NPV, the higher the benefits and the more feasible the project becomes[159]. However, there are three possibilities form NPV analysis, these are

- $NPV > 0$ Economically viable for a specified condition of the project
- $NPV = 0$ Economically feasible but the ROI equals the discount rate
- $NPV < 0$ Investment not viable economically for a specified condition

NPV is mostly utilised in many commercial and industrial sectors and is regarded as more complete techniques in evaluating economic investment viability of the project. It is worthy of note that the cash flow in NPV analysis is discounted based on the following two main reasons;

- (i) To adjust for the risk of an investment opportunity
- (ii) To account for the time value of money

Based on this, the NPV was adopted as the appraisal method in this research for exploring the economic viability of investing in Trans-Saharan gas pipeline for natural gas transportation. Furthermore, the present study also introduces additional terms such as “divestment cost” and “leasing cost” in calculating the NPV. The addition of these terms will compensate for the redundant fleets of the engine after overhaul. It should be noted that the salvage value of the engine is known to be an estimated resale value of the engine at the end of its useful life[160]. It implies that the salvage value will be the resale value at a time the owner intends to sell it for economic reason. The approach was considered in this research since there must be some redundant fleets of engines after overhaul.

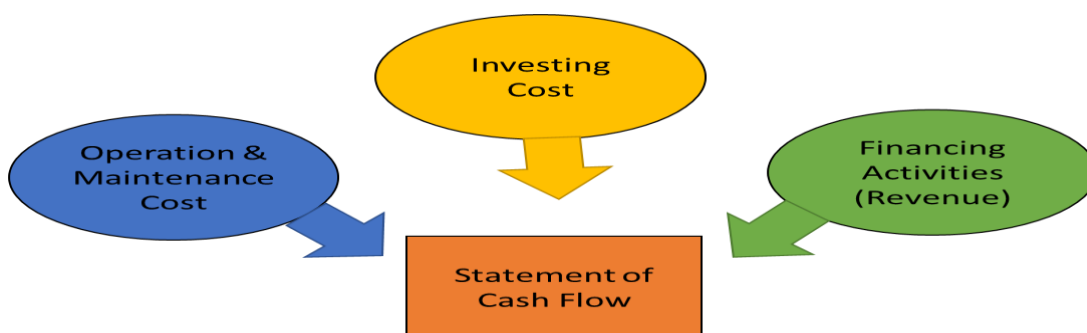


Figure 2-26 Cash flow activities

2.14 Summary of Chapter

The review shows that among all the means of natural gas transportation, pipeline accounts for about 70% of global natural gas transportation[16] and the present research considers only single-phase flow compressible flow that transports natural gas in a pipeline. Among the various pipeline flow equation, Weighmouth equation was adopted in this research. It was reviewed that pipeline elevation, and the capacity of the equipment should be a contributory factor to be considered when locating the compressor station on the pipeline network. Among the various gas compressors used in the oil and gas industry, the centrifugal compressor was chosen because of its versatility, compactness, a high degree of reliability and availability and are also useful owing to their high efficiency, good economy, and low maintenance [108]. It was concluded that among all the degradation mechanism, GT engines are vulnerable to fouling and fouling was adopted as the engine degradation to be considered in this research. It was also discovered that the failure of many researchers in implementing engine degradation in the economic analysis of GT engines for compressor station operation become a gap in this research. The implementation of the annual degradation of engine operation in a broader range is to investigate how this practice could affect the entire techno-economics of the gas transportation system. The review also presented various methods of compressor washing. However, the on-line method was adopted in this research to avoid economic loss due to downtime.

Finally, different economic appraisal techniques were outlined to ascertain the economic viability of a natural gas transportation network and any investment project. The NPV was adopted in this research as the appraisal method. The principle of TERA was presented as the technique that will take into account all the multi-disciplinary variables that are involved in the present research.

3 RESEARCH METHODOLOGY

3.1 Overview

This chapter presents the relevant approaches and tools used for the pipeline transportation network. Such an approach and tools include an overview of the compressor and pipeline modules, which is one of the components of this research. Also, the application of techno-economic and environmental risk assessment (TERA) developed towards this research is also explained. The TERA framework in this research consists of several modules which comprise of pipeline/gas compressor, performance, lifing, emission, and economic module. All these modules are connected to evaluate both the technical and economic viability of the natural gas pipeline network. The performance module plays a significant role in the use of TERA tool. Therefore, TURBOMATCH, which is very significant in analysing the performance of gas turbine engines, is also discussed in this chapter. Considering the complexity (multi-system) of this research, and the discussion of the various approaches and tools adopted for this research in this chapter, it is important to state that, the analysis of the subsequent chapters shall elaborate and substantiate the methods adopted. The evaluation will give a clearer view of the obtained result from each of the relevant section of this research.

3.2 Pipeline and Gas Compressor (GC) Module

3.2.1 Introduction

This module presents the investigation of the proposed 4,180km TSGP to analyse the pipeline performance and to evaluate the power requirement of the gas compressor across the entire 18 compression station. The analysis involves both isothermal and non-isothermal condition of the pipeline and the gas compressor in a time-based system.

The objective of the first section of this module was to evaluate the pipeline outlet pressure since a fixed inlet pressure was given and to authenticate the correlation between the performance parameters of gas compressor and pipeline. The investigation further looked into the effect of varying gas temperature with their corresponding pipeline exit pressure on the rate of throughput delivery at constant pipe diameter and how these parameters relate together on the overall performance

of the pipeline network and gas compressor (centrifugal compressor). Also, the latter part of the module evaluates the power and performance characteristics of centrifugal compressors using the available thermodynamic gas equations. The effect of the suction pressure of the GCs on the power requirement was also investigated.

Before the gas compressor power can be evaluated, the various discharge pressure of the pipeline after the first compressor station needs to be calculated. The reason is that the GC needs an inlet pressure from the pipeline. This inlet pressure which is the pipeline discharge pressure is unknown and need to be calculated.

Achieving the pipe exit pressure in each of the compressor stations help to evaluate the PR and subsequently, the power of the gas compressor across the entire compression station. The analysis was done for three seasons of the year under a time-based ambient temperature.

It is worthy of note that pipeline analysis becomes very important in calculating the GC power requirement after the first CS. The reason is that it was assumed in this research that the outlet condition of the pipeline becomes the inlet condition of the gas compressor and vice versa. Utilising this phenomenal assumption will authenticate the correlation between the GC and pipeline. With this assumption in mind, since the inlet pressure of the GC apart from the first compressor station was unknown, the discharge pressure of the pipeline after calculation becomes the inlet of the GC. Also, the ambient temperature becomes the inlet temperature of the pipeline [18]. So, as the time of day changes, the ambient temperature changes also, hence the “non-isothermal condition”. On this basis, the pipeline and the gas compressor was analysed. With the help of the available data from the TSGP, Aspen-hysis/Microsoft-excel was used for the evaluation of the gas compressor power while the pipeline model was developed in MATLAB

Gas component and properties obtained from the Trans-Saharan Gas Pipeline (TSGP) report were used for the evaluation of gas compressor and pipeline analysis. The equations involve the use of available gas compressor inlet parameter and gas property calculation for the evaluation of the required power by the centrifugal compressor.

However, Tukur[18] explores the available TSGP data and evaluated the elevation and distances between the compression stations. Based on this analysis, the present study further analysed the TSGP and investigated the discharge pressure of the pipe and the corresponding gas compressor power in all the CS.

3.3 Pipeline performance Calculation

Table 3-1 shows the necessary data given from the TSGP report. From the report, as stated on the table, all GCs to be installed in each compression station along the pipeline are expected to have a pressure ratio (PR) between 1.4 and 1.66.

Table 3-1 TSGP design data

Parameters	Values	Unit
Total Pipeline distance	4180	km
Number of CS	18	
Pipe diameter	1.4224	m
Delivery of gas/year	30billion	m ³ /year
PR of first CS	1.4	
MAOP	95	bar

Since the ambient temperature varies according to the time of the day, so also the various parameters that will be calculated vary in each season of the years.

There are several commonly used model equations to predict the performance of a pipeline transporting gas, as presented in chapter 2. These equations show the relationship between the pipe and the gas properties, such as compressibility factor, gravity, flow rate, pipe length, diameter and pressures along the pipeline[22]; [42]. Weymouth equation was adopted for this analysis because the pipeline under investigation falls into the class of large diameter pipeline, high flow rate and high pressure. Again, Weymouth equation compensates for the pipeline elevation and uses pipeline efficiency factor to avoid the iterative process as seen in other flow equation like Colebrook-White equation that uses trial-and-error approach and ends up in iterative process[42]. Also, the Weymouth equation is being utilised in Aspen-Hysis, been the software for GC power evaluation.

3.3.1 Methodology

3.3.2 Local Condition

The average ambient temperature profile with respect to time of day of the area where the natural gas pipeline and the engines are operated is shown in Figure 3-1.to Figure 3-3. The temperature profile shows the average temperature profile for the three seasons of the year, (rainy, dry and hot seasons) across the 18 compression stations. Since the pipeline has a different compressor station, each station has its different ambient temperature that varies from each other and from season to season. It is worthy of note that the rainy season was the season with lower temperature with respect to time of day, the dry season's temperature was intermediary between the rainy and hot while the hot season was the season with the highest temperature

Based on these ambient conditions, the pipeline and gas compressor analysis were carried out. It is interesting to note that three hours interval was chosen to ascertain the ambient temperature, and these temperatures are the cumulative three-hourly-temperature with respect to time of day for each month. This research uses the average temperature of each season with the assumption of four months for each season. In all the seasons, it was observed that at 15:00 time of day, the ambient temperature was the highest while at 6:00, the ambient temperature recorded the lowest value.

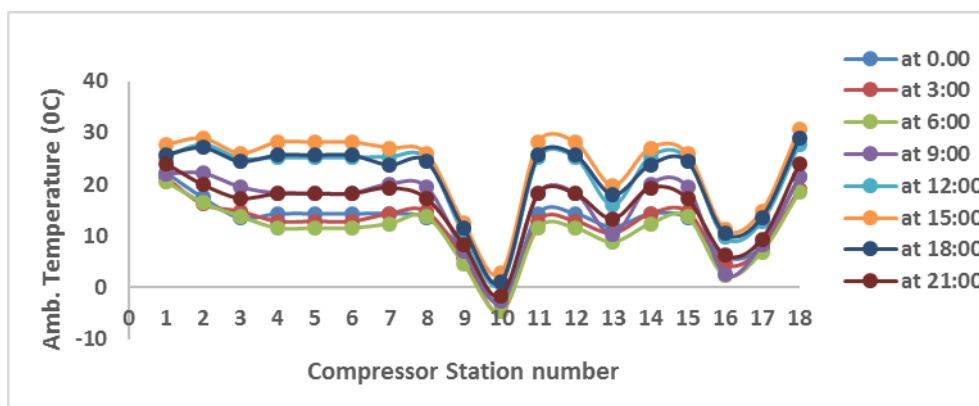


Figure 3-1 Time-based ambient temperature (rainy Season)[26]

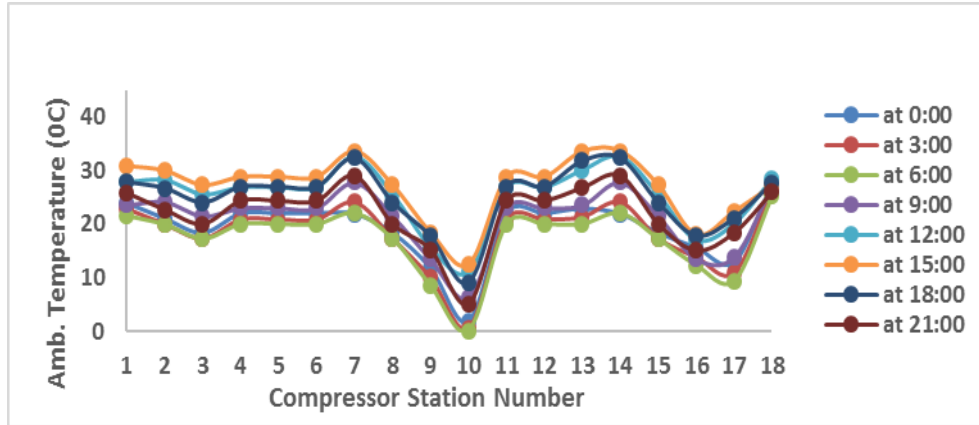


Figure 3-2 Time-based ambient temperature (dry season)[26]

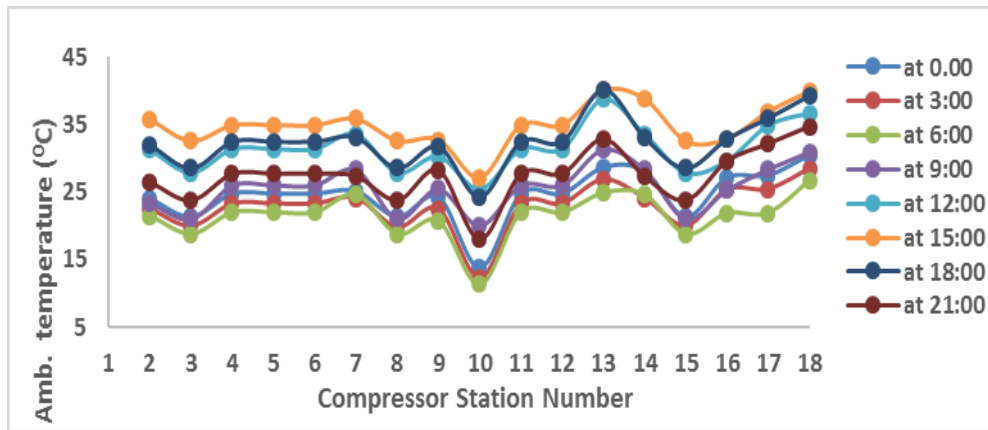


Figure 3-3 Time-based ambient temperature (Hot season)[26]

3.3.3 Pipeline analysis

Figure 3-4 shows the methodology and calculation method adopted for pipeline performance. The gas component and properties obtained from TSGP aided the evaluation of the pipeline and gas compressor analysis. The vital gas properties used were the molecular weight (MW_g), gas gravity (G) and gas temperature. These helped in evaluating the relevant parameters for the pipeline analysis. The basic parameters required for the pipeline module as indicated in Equation (3-1) are the gas mass flow (Q), pipe efficiency (E), base temperature and pressure (T_b & P_b), Inlet and outlet pressure of the pipeline (P_1 & P_2), elevation adjustment parameters (e^s), gas gravity (G), gas temperature (T_f), length of the pipeline (L_e), compressibility factor (Z) and the pipe diameter (D)[22]; [60]. Figure 3-4 also represents these parameters.

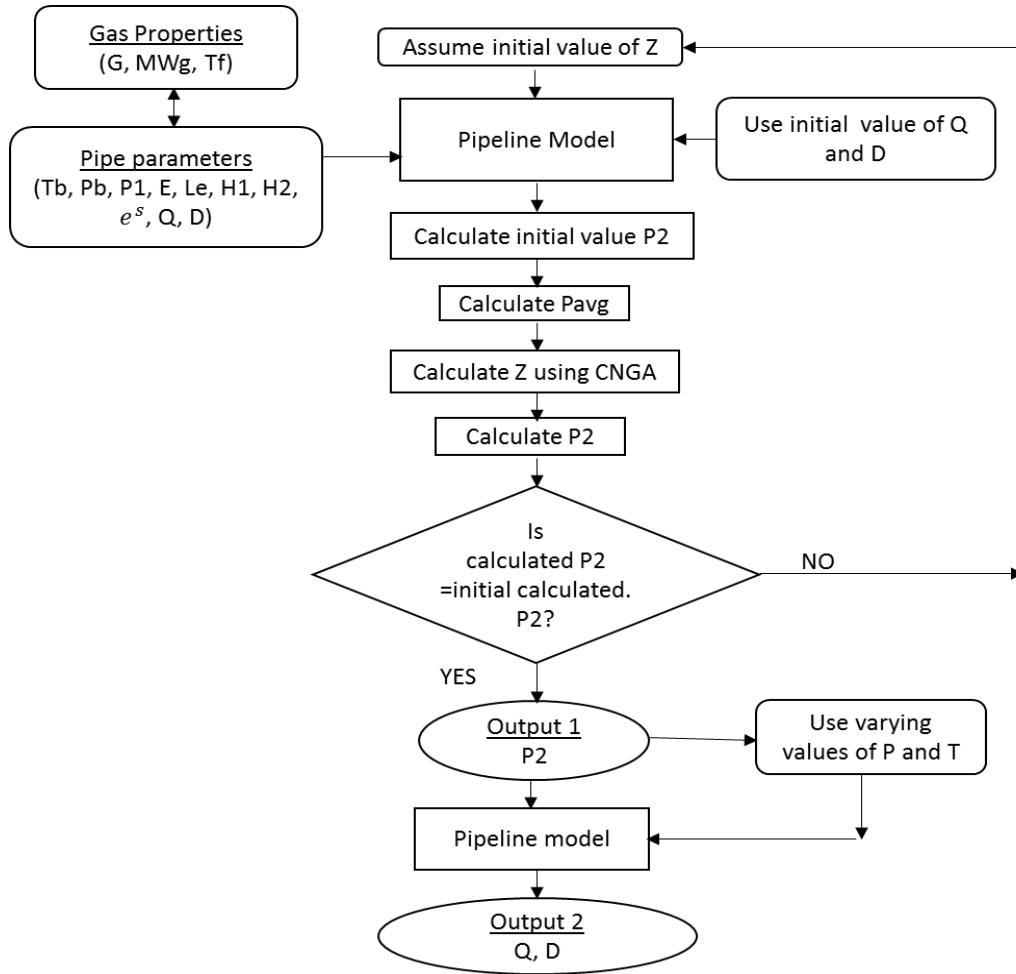


Figure 3-4 Methodology for pipeline module

Having known the discharge pressure of the gas compressor which is fixed at 95 bar which also is the inlet of the pipeline, the various distances (L) between each pipeline segment and elevation, then the pipeline discharge pressure can be calculated by rearranging equation (3-1). Therefore, the pipe outlet pressure (P_2) is made the subject of the formulae as represented in the derived Equation (3-2).

$$Q = 3.7435 * 10^{-3} E \left(\frac{T_b}{P_b} \right) \left[\frac{P_1^2 - e^s P_2^2}{G T_f L_e Z} \right]^{0.5} D^{2.667} \quad (3-1)$$

$$P_2 = \left\{ P_1^2 - \frac{\left[G T_f L_e Z \left(\frac{Q * P_b}{3.7435 * 10^{-3} E * T_b * D^{2.667}} \right)^2 \right]}{e^s} \right\}^{0.5} \quad (3-2)$$

Where, elevation adjustment parameter (s) is given in equation **(3-3)** This depends on gas gravity (G), gas temperature (T_f), elevation (H_1 and H_2) upstream and downstream elevations respectively, e is the base of the logarithm ($e = 2.718$)

$$s = 0.0684G \frac{H_2 - H_1}{T_f Z} \quad (3-3)$$

However, in some cases, to achieve a more realistic value of the pipeline discharge pressure if there are many unknown parameters on the pipeline module, sequence of iteration is done in the system. In Equation **(3-2)** P_2 and Z are unknown and we need to get Z so that P_2 can be evaluated. Because of the above expression, a flow compressibility factor (Z) of 0.95 was assumed for the initial iteration for the calculation of P_2 . The already known inlet pressure and the calculated discharged pressure for the initial value of Z is used to calculate the average pressure using equations**(3-4)**. This is necessary because equation **(3-4)** is generally accepted to be a more precise method for calculating the average pressure of the gas in a particular pipeline segment [42]. With this initial result, a new compressibility factor can be calculated using the California Natural gas association method (CNGA) as shown in Equation**(3-5)**. The result will be compared with the preliminary used value until the convergence level. In this case, the discharge pressure will achieve its confident value. This is illustrated in Figure 3-4.

$$P_{avg} = \frac{2}{3} \left(P_1 + P_2 - \frac{P_1 * P_2}{P_1 + P_2} \right) \quad (3-4)$$

$$Z = \frac{1}{\left[1 + \frac{344400 P_{avg} x 10^{1.785G}}{T_f^{3.825}} \right]} \quad (3-5)$$

Equation **(3-4)** is valid when the average gas pressure (P_{avg}), is more than 6.9 bar. For pressure less than 6.9 bar, Z is approximately equal to 1.00. Hence, equation **(3-4)** is valid for this analysis because P_{avg} is above 6.9bar. Specific gas gravity (G) tells how heavy the gas (Molecular weight, M_{wg}) is in comparison to air (M_{air}) at a certain temperature and pressure. This was calculated thus:

$$G = \frac{M_{wg}}{M_{air}} = \frac{M_{wg}}{28.95} \quad (3-6)$$

Generally, according to Figure 3-4, the P_2 at the various CS was achieved with the initial given value of the gas volume flow (Q) and pipe diameter (D). This is referred to as output 1. After that, varying values of T_2 with their corresponding values of P_2 were used to evaluate the different Q and D . This is also referred to as output 2. The model was explored under the isothermal and non-isothermal condition

Table 3-2. shows the pipeline route indicating the elevation with respect to the eighteen compressor station and the various distances between each compression station which sum up to a total distance of 4,180 km pipeline as analysed and presented by Tukur [18]. These values were used for the various distances and elevation between each pipe segment in calculating the pipe exit pressure.

Table 3-2 Distance and Elevation between pipe segment[18]

CS Number	Distance (km)	Elevation (m)
1	240	100
2	240	200
3	240	600
4	240	402
5	240	400
6	240	400
7	230	480
8	250	600
9	230	800
10	230	1050
11	230	400
12	230	400
13	230	560
14	230	480
15	230	600
16	230	900
17	230	700
18	190	240

Having known both the inlet (P_2) and discharge (P_1) pressures of the pipeline which becomes the outlet (P_2) and inlet (P_1) pressures of the gas compressor respectively, the pressure ratio (PR) of the gas compressor can be calculated.

Table 3-3. shows the data of the pipe segment for the first compressor station at design point obtained from TSGP. All compression stations analysis was done based on the first station. The First station was assumed to operate at ISA condition.

Table 3-3 First Compressor station design point[161]

Parameters	Values	Unit
Pipeline Inlet Pressure P1	95	bar
Pipe Discharge pressure P2	67.85	bar
Pipe nominal diameter	1.4224	m
Pipe internal diameter	1.3812	m
Pipe outer diameter	1.4224	m
Wall thickness	0.0206	m
Gas gravity (G)	0.6048	-
Base temperature	288.15	K
Gas temperature	288.15	K
Base pressure Pb	101	bar
MAOP	95	barg
Elevation	100	m
Equivalent length Le	242.1	km
Efficiency E	91.8	%
Mass flow Q	82000000	m ³ /day

3.4 Result and discussion

Figure 3-5 shows the pipeline discharge pressure across the 18 compression stations. This is the discharge pressure entering the GC in each of the compression stations according to the time of day. From Figure 3-5, the pipeline discharge pressure at 15:00 time of day was the lowest while 6:00 time of day had the highest discharge pressure across the entire 18 CS. It was also observed that at 15:00 time of day the inlet (ambient) temperature across the entire CS was higher while at 6:00 the ambient temperature recorded the lowest value as shown in Figure 3-1 to Figure 3-3. The discharge pressure of the pipeline at each of the compressor station was assumed to be the inlet pressure of the gas compressor as earlier stated. This shows the correlation between the pipeline and the gas compressor during the operation. It is important to state also that the gas compressor discharge pressure became the pipe inlet pressure. This pressure was fixed at 9500kPa throughout the entire CS.

It is worth noting that only the hot season was used for this evaluation because the hot season gives the highest ambient temperature for the discharge pressure evaluation. It was also assumed that no deterioration was found with the pipeline under consideration.

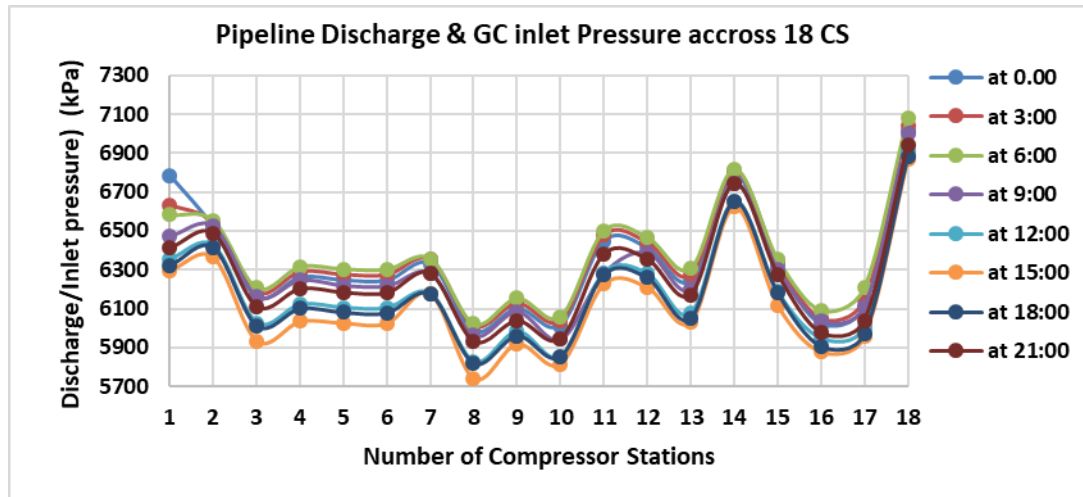


Figure 3-5 The pipe discharge & GC inlet pressure at the various CS

3.4.1 Effects of Pipe Exit Pressure on Volume of Gas flow at different gas Temperature

Although the amount of gas to be transported through the pipeline yearly was given in the TSGP data, with the aid of the developed model, Figure 3-6 shows the relationship between the gas flow (Q), discharge pressure (P_2) and gas temperature (T_f) in the pipeline system. It revealed the variability of the pipe discharge pressure with respect to the volume of gas flow at a different gas temperature in the pipeline all at constant pipe diameter. From the result; the flow of gas expected at a particular pipe exit pressure can be predicted. For instance, from the curve, it was observed that an increase in the exit pressure of the pipeline resulted in a corresponding increase in the amount of gas flow. It was further revealed that for every 1% increase in exit pressure resulted in a 1.8% increase in the volume of gas flow. Again, an increase in gas temperature results in a corresponding decrease in both pipe exit pressure and the volume of gas to be transported into another CS, hence the need to have many booster stations along the pipeline. However, the non-linearity of the curve, as shown in Figure 3-6, indicated that the gas temperature changes with respect to time of day.

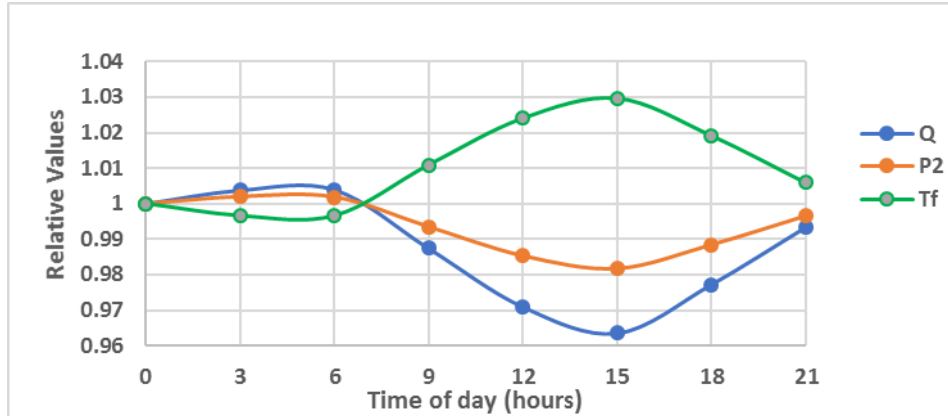


Figure 3-6 Relationship between gas flows, Exit pressure and gas temperature in the pipeline

Figure 3-7 and Figure 3-8 shows the relationship between the volume of gas flows both at different pipe exit pressure with a corresponding gas temperature and at constant pipe exit pressure and gas temperature, respectively. Figure 3-7 represents the non-isothermal condition of the gas flow in a pipeline. The result shows that as the gas temperature increases, the exit pressure reduces, which directly affects the flow rate. It was observed that the exit pressure of 2400.83kPa has the highest gas temperature with 302.84K. This implies that the gas temperature has a great influence on the gas exit pressure of the pipeline flows. At this pressure and temperature, the highest flow rate was recorded. From this curve, one can predict an economic pipe size suitable for a corresponding flow at any desired pipe exit pressure and gas temperature. For instance, it was observed that 1550mm pipe diameter can conveniently deliver 30.2bcm/year within an exit pressure of 2400kpa and 4800kpa when the gas temperature is between 295.61K and 302.84K. However, if the pipeline operates at an environment below this temperature, the pipe exit pressure will be higher. Figure 3-8 also represents a pipeline operating in an isothermal condition where it was assumed that the gas temperature remained the same throughout the compression process. Also, the exit pressure was assumed to be the same throughout the compression process. The value used for both gas temperature and exit pressure for the isothermal condition was 302.84K and 6440.69kPa respectively, from the result, one can predict a particular pipe diameter with respect to the desired gas flow. However, a condition when gas temperature is not steady, the use of an isothermal model leads to significant errors. Therefore, a

real pipeline system does not operate on isothermal condition as there will be friction and other conditions that can vary the performance during operation.

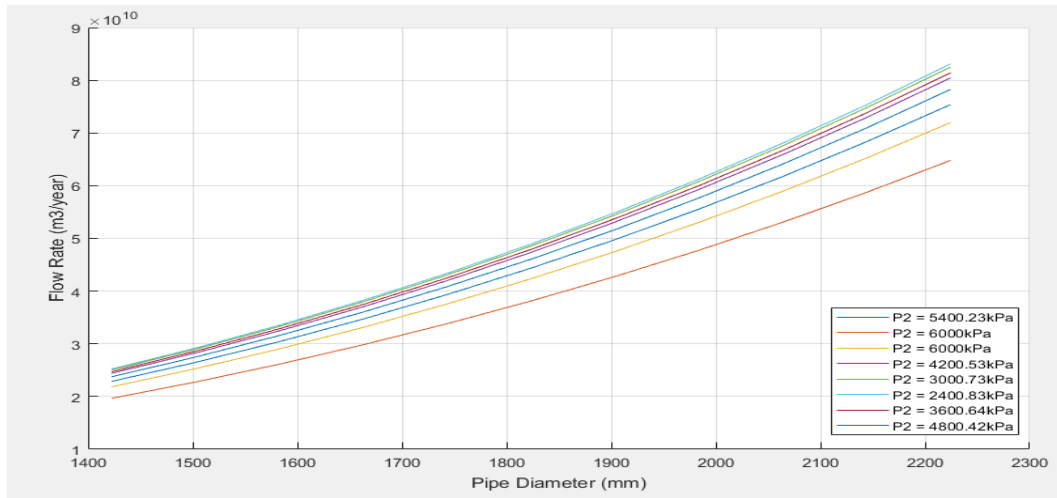


Figure 3-7 Effects of varying exit pressure and gas temperature on gas flow in a pipeline

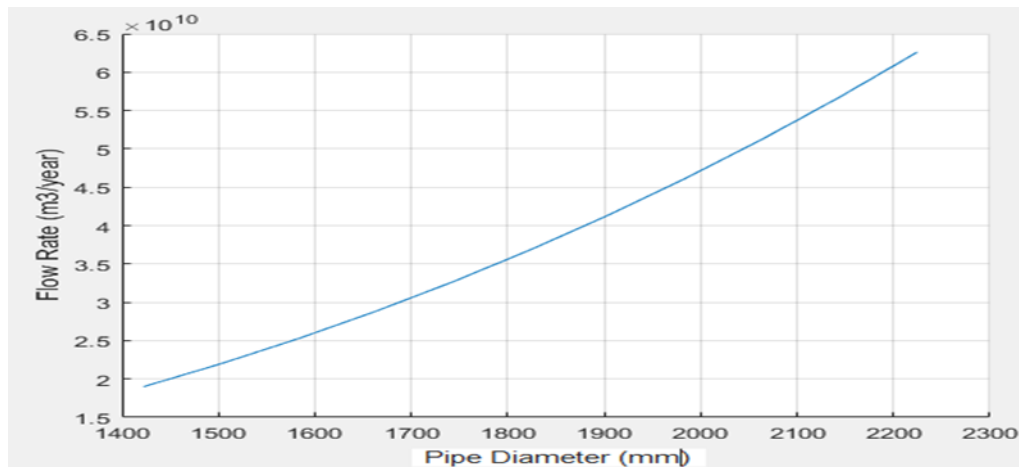


Figure 3-8 Effects of constant exit pressure and gas temperature on gas flow in a pipeline

3.5 Evaluation of Gas compressor

3.5.1 Overview

The main focus of this section is to determine the power requirement of the gas compressor (GC) across the 18 compression station in a non-isothermal condition with the available data from the TSGP report and the pipeline performance at a different operating condition. The evaluation was done in a time-based system. The

effect of the suction pressure of the GCs on the power requirement was investigated. Aspen-Hysis was used to validate the gas compressor power evaluation. Finally, the newly developed gas compressor map to monitor the performance of the gas compressor on the pipeline compression system is also presented.

3.5.2 Evaluation of Gas compressor Pressure Ratio at Various Compression stations

The discharge pressure (P_2), of the GC, was fixed at 95bar. The pipeline discharge pressure becomes the suction pressure (P_1) of the GC after station one. The elevation and distances between each CS, as shown in Table 3-2 have a significant role in evaluating the gas compressor powers. This is because the elevation and distances between each CS were taken into consideration when evaluating the discharge pressure of the pipeline, which became the inlet pressure of the GC. The desired range of PR of the GC in the entire CS was determined by dividing the discharge pressure (P_2) by the suction pressure (P_1) which thus, enhanced the accurate evaluation of gas compressor power. Equation (3-7) shows the PR calculation

$$PR = \frac{P_2}{P_1} \quad (3-7)$$

Figure 3-9 shows the result for the gas compressor PR, across the 18 CS with respect to time of day. The PR shows the ratio of the GC outlet to the inlet pressure. The pipeline discharge pressure becomes the GC inlet pressure as presented in Figure 3-5. Since the GC discharge pressure was already given, with this pipeline discharge pressure, the PR was calculated. The result of the gas compressor PR shows the correlation between the pressure of the pipeline and the GC. The gas compressor power was calculated based on these PRs, as shown in the polytropic head calculation.

It is worth noting that only the hot season was analysed because the hot season will give the highest power of the gas compressor upon which the prime mover of the various stations can be selected. The evaluated power within this PR will be capable of delivering the desired amount of gas in each CS irrespective of the varying ambient temperature and elevation. Also, since the GC is a pressure booster, the calculated pressure ratio falls within the range of PR as recommended in the TSGP report. The

result obtained shows that for every 1°C rise in gas temperature result in a 0.2% rise in GC PR. This shows the effects of increasing gas temperature on the PR of the GC

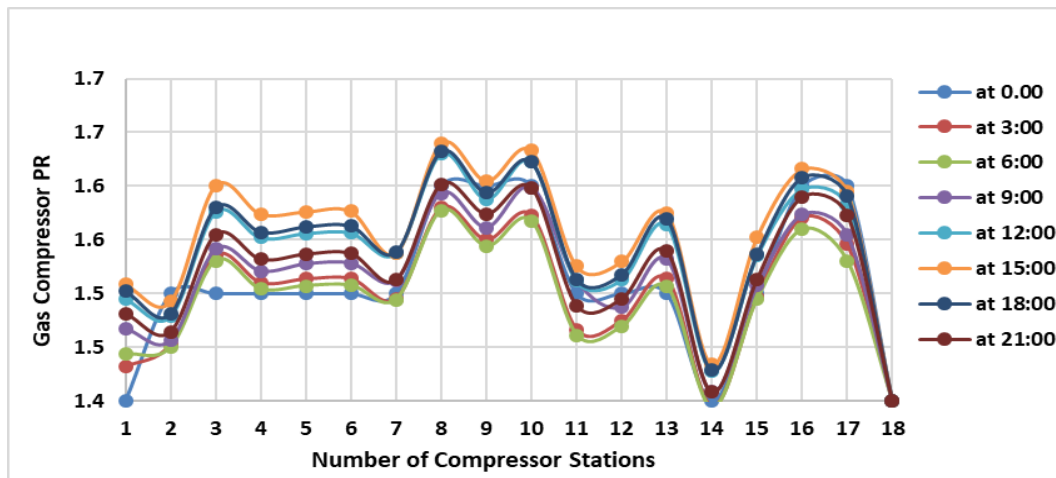


Figure 3-9 Gas compressor PR at varying ambient temperature

3.5.3 Gas Properties and GC power evaluation using Aspen-Hysis

Figure 3-10, shows the method introduced in achieving the gas compressor power using Aspen-Hysis. Aspen-Hysis is a complete thermodynamic process modelling tools used to optimise process design and general operation of the pipeline transportation system. Aspen-Hysis was used to determine the power of the gas compressor that is capable of maintaining the discharge pressure of 95bar. It is also useful in calculating temperatures and pressures on pipeline design module, also polytropic head and efficiency, which help to achieve the gas compressor power. Aspen-Hysis use a method of software calculation, which is based on numerical integration.

The fluid components (gas mixtures) which are solely the hydrocarbon series such as methane, ethane, propane etc (methane having the highest volume fraction of 92.18% from the available data) with their respective composition are selected and the fluid package (See TableA-1 and A-2 of Appendix A). Peng-Robinson was chosen as the fluid package in Aspen-Hysis because it is better used for hydrocarbon and accommodate a high temperature and pressure range.

Having ascertained the fluid composition and package, the temperatures (T), PR and the volume flow (Q) were inputted into the Aspen-hysis, and all necessary parameters such as molecular weight, molar density, compressibility, etc will be

calculated. These parameters defined the polytropic head and efficiency. After that, the gas compressor power is then calculated. It is interesting to note here that as the temperature, PR and volume flow are inputted, the gas compressor power will also be calculated. This makes the software an easy tool in evaluating the GC power in a process pipeline.

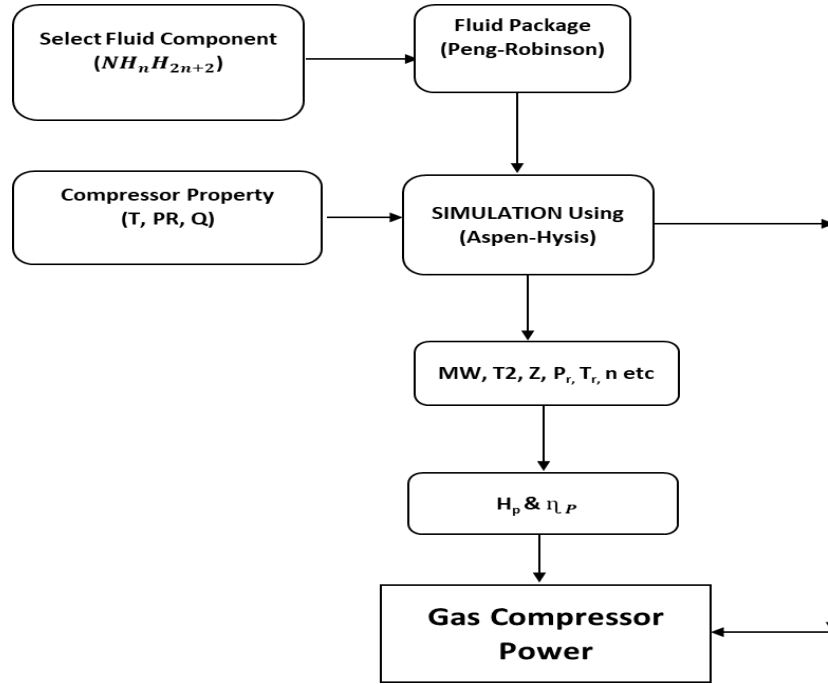


Figure 3-10 Steps in achieving the gas compressor power using Aspen-Hysis

3.5.4 GC power calculation

The gas mixture analysis is been evaluated using the basic thermodynamic equation, as presented below. The gas mixture properties were calculated from the individual gas property from the following basic thermodynamic equation: These equations are also used in Aspen-Hysis

$$MW_m = \sum_{i=1}^{i=n} V_i MW_i \quad (3-8)$$

$$Cp_m = \sum_{i=1}^{i=n} V_i CP_i \quad (3-9)$$

$$Pc_m = \sum_{i=1}^{i=n} V_i PC_i \quad (3-10)$$

$$Tc_m = \sum_{i=1}^{i=n} V_i Tc_i \quad (3-11)$$

Where MW_m -the molecular weight of gas mixture; CP_m -Molar specific heat; Pc_m , - critical pressure of the gas mixture; Tc_m = critical temperature of the gas mixture; MW_i -The molecular weight of individual gases; CP_i - specific heat of the individual PC_i -Molar specific heat of the individual gas; Tc_i - the critical temperature of the gas mixture

The reduced temperature (T_{rm}) and reduced pressure (Pr_m) of the gas mixture were calculated using the inlet temperature (T_m) and pressure (P_m) with the critical temperature(Tc_m) and pressure (Pc_m) of the gas mixture, as shown in Equation (3-12) [162]; [163]. The inlet temperature was the ambient temperature while the inlet pressure after CS 1 was the discharge pressure of the pipeline of the various CS.

$$T_{rm} = \frac{T_m}{Tc_m} \quad Pr_m = \frac{P_m}{Pc_m} \quad (3-12)$$

Having achieved Pr_m and T_{rm} of the gas mixture, Z was evaluated using the generalised Nelson-Obert chart, in which compressibility factor is plotted as a function of reduced temperature (T_r) and reduced pressure (P_r). This chart has been shown that it gives a satisfactory value of several gases and gas mixture[163]

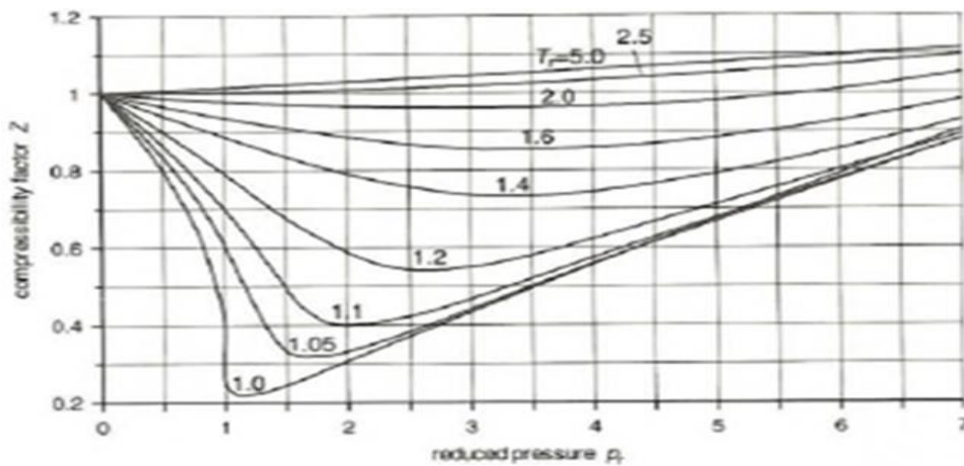


Figure 3-11 Nelson-Obert compressibility chart [163]

To calculate the polytropic exponent (n), and the polytropic efficiency (η_p), the ratio of specific heat (k) has to be known. However, $k=1.296$ as stated on the table in Appendix A1. It can also be calculated using Equation (3-13).

$$K = \frac{CP_m}{CP_m - R_0} \quad (3-13)$$

Where R_0 is the gas constant. From the polytropic exponent, the Polytropic efficiency is calculated thus: The polytropic efficiency of the compressor at DP was estimated using polytropic efficiency & Head coefficient vs flow coefficient chart, as shown in **Error! Reference source not found.** [164]. 0.13 of the flow coefficient was assumed[165]

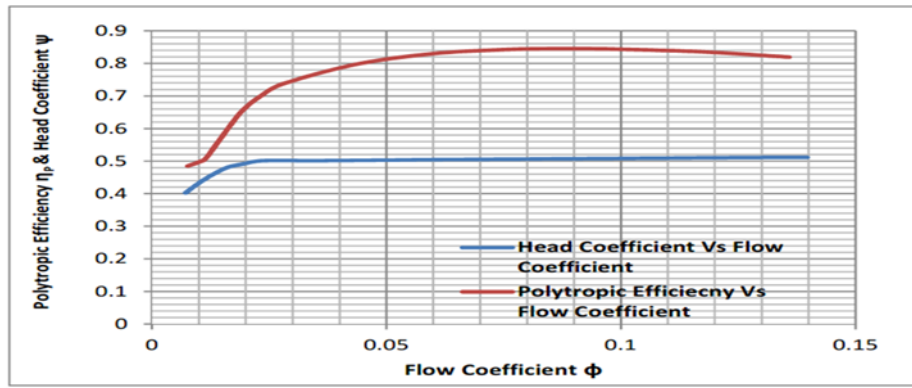


Figure 3-12 Polytropic efficiency & Head coefficient vs flow coefficient

$$\frac{n-1}{n} = \frac{k-1}{K\eta_p} \quad (3-14)$$

$$n = \frac{K\eta_p}{K\eta_p - (K-1)} \quad (3-15)$$

Where K is the ratio of specific heats and n is the polytropic exponent. The discharge temperature of the gas compressor is one of the factors that limit the stage compression ratio. Based on this reason, it was kept within a reasonable value. Meanwhile, the discharge temperature depends on pressure ratio, process exponent and inlet temperature and it is calculated using equation (3-16)

$$T_2 = T_1 + T_{actual} \quad (3-16)$$

$$T_{actual} = \frac{T_1 \left[PR^{\frac{n-1}{n}} - 1 \right]}{\eta_p} \quad (3-17)$$

The compressibility factor, specific gravity, polytropic exponent etc. are essential parameters in the analysis of gas flow which affects the energy required to compress certain volume of the gas and also affects the pressure drop in pipeline. These are also useful in calculating gas power.

3.5.5 Polytropic Head Calculation

The polytropic head is calculated using Equation (3-18) having known the pressure ratio, molecular weight, and all necessary gas properties from the pipeline calculation,

$$H_p = \frac{8314}{MW} Z_a T_1 \frac{n}{n-1} \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right] \quad (3-18)$$

The gas power is computed from the polytropic head equation in equation (3-18). In general design consideration, compressor designer has to select the gas compressor for multiple operating conditions.

3.5.6 Evaluation of Volumetric Flow

The compressor inlet gas volumetric flow rate was calculated using equation (3-19) and (3-20)

$$p = \frac{P_s}{Z_s R T_s} \quad (3-19)$$

$$Q_1 = \frac{m}{\rho} \quad (3-20)$$

Where- Q_1 stand for volume flow rate at inlet (m³/h), m-mass flow rate (kg/h), p-density of gas ((kg/m³), P_s -suction pressure (bara), Z_s - compressibility factor, T_s - Suction temperature (K) and R-universal gas constant. However, the mass flow rate has been given in the TSGP report. Based on this analysis,

The DP gas power can be calculated, having ascertained all the necessary parameters using

$$P = \frac{W*H_p}{3,600\eta_p} \quad (3-21)$$

Mechanical losses like bearing losses (Leakage losses, coupling losses) including friction between air and impeller, turbulent in fluid flow and shock at entry need to be accounted for to actualise the gas compressor shaft power. 2% of the gas power was used for bearing and other mechanical losses. Also, if a step-up gearbox is used, then gearbox efficiency need to be accounted for, typically, a range of 97% to 99% is been used in the industry[166]. However, the step-up gearbox was not considered in this research.

Summarily, Gas compressor shaft power = Gas Power (P) + Mechanical losses (M_L).

Having analysed the gas power calculation, Table 3-4 shows the GC power for CS 2 in comparison with the calculated values. Where, RS, DS and HS represent the rainy season, dry season and hot season respectively.

Table 3-4 Aspen Hysis and calculated result

GC power Evaluation for CS 2 (kW)						
	Aspen-Hysis Result			Calculated Result		
Time	RS	DS	HS	RS	DS	HS
0:00	10291.01	10719.2	11092.03	10292.01	10721.5	11095.53
3:00	10139.82	10596.12	10904.94	10141.32	10598.12	10907.94
6:00	10169.98	10596.12	10780.96	10171.08	10599.12	10784.46
9:00	10873.89	11123.34	11060.75	10875.19	11125.34	11063.85
12:00	11564.88	11629.62	12015.89	11566.28	11632.12	12019.39
15:00	11725.65	11854.25	12606.11	11727.65	11858.25	12610.11
18:00	11501.52	11438.31	12113.35	11503.02	11440.31	12116.95
21:00	10596.12	10936.02	11406.76	10597.21	10938.42	11410.26

3.5.7 Gas Compressor Operating Characteristics

Compressor characteristic curve defines the behaviour of the compressor at the varying operating condition. On the curve, “polytropic efficiency vs volume flow rate” and “polytropic head vs volume flow”. Such a curve is developed by the OEM’s by

testing the compressor on the testbed. In order to understand the behaviour of the centrifugal compressor at different operating conditions, a map needs to be developed. In this research, the new compressor map was produced by scaling the default map with respect to the operating conditions of the new gas compressor. The OEM map is shown in Figure B- 1 and Figure B- 2 of Appendix B. The newly developed map will best describe the operation of the newly designed gas compressor. The scaling factors are given thus:

$$PRSF = \frac{PR_{DP} - 1}{PR_{DP\ Map} - 1} \quad (3-22)$$

$$PR_{New} = PRSF * (PR_{Map} - 1) + 1 \quad (3-23)$$

$$ETASF = \frac{\eta_{DP}}{\eta_{DP\ Map}} \quad (3-24)$$

$$\eta_{DP} = ETASF * \eta_{DP\ Map} \quad (3-25)$$

$$WASF = \frac{WAC_{DP}}{WAC_{DP\ Map}} \quad (3-26)$$

$$WAC_{DP} = WASF * WAC_{DP\ Map} \quad (3-27)$$

Where,

PRSF-Pressure ratio scaling factor

PR-pressure ratio

ETASF-polytropic efficiency scaling factor

WASF-mass flow scaling

Other parameters as defined in nomenclature.

Parameters with subscript DP and DPMap indicate design point for new maps and the existing (OEM) map, respectively. So the values of scaling factors of each of the performance parameters are derived using the design point calculated value of the gas compressor at each compressor station with the values from the existing map after correction.

In order to produce a compressor map in terms of corrected rotational speed and flow rate, the corrected rotational speed can be derived by maintaining a fixed tip Mach number alongside the gas properties variation. This enables the equivalent

speed at various off-design conditions in addition to suction parameters and gas properties to be determined[167]. This is represented in Equation (3-28).

This equation takes into consideration the influence of gas properties on the performance,

$$N_c = N_{act} \sqrt{\frac{k_{Ref}}{k_{act}}} \sqrt{\frac{Z_{Ref}}{Z_{act}}} \sqrt{\frac{R_{Ref}}{R_{act}}} \sqrt{\frac{T_{Ref}}{T_{act}}} \quad (3-28)$$

Accordingly, the corrected mass flow as represented in Equation (3-29) was derived based on Buckingham Pi Theorem and was used for the present research.

$$\dot{w}_c = \dot{w}_{act} \frac{P_{Ref}}{P_{act}} \sqrt{\frac{k_{Ref}}{k_{act}}} \sqrt{\frac{Z_{act}}{Z_{Ref}}} \sqrt{\frac{R_{act}}{R_{Ref}}} \sqrt{\frac{T_{act}}{T_{Ref}}} \quad (3-29)$$

The effects of humidity factor were negligible because of its insignificant effect on this correction as proposed by Berdanier[168]

Where,

N_c =Corrected rotational speed

$\dot{w}_c$ - corrected mass flow

k_{Ref} -Reference specific heats ratio (1.4)

Z_{Ref} - Reference Compressibility factor(1.0)

T_{Ref} – Reference Suction Temperature; (288.15K)

P_{Ref} – Reference Suction Pressure(101.325kpa)

R_{Ref} – Reference gas constant (287.058J/(kg.K)

All parameters with subscript act indicate the actual values for the map.

3.5.8 Discussion

Figure 3-13 to Figure 3-16 shows the gas compressor power at various ambient temperature across the 18 compressor station with respect to the ambient temperature for the three seasons according to time of day.

It was observed that while each station shows the power with respect to time of day, the highest power of the gas compressor was shown at 15:00 across the 18 CS where the ambient temperature was high indicating that, as the ambient temperature

increases, the power of the compressor increases also. While at 6:00, the least power was calculated. The result also revealed the correlation between ambient temperatures and GC power. As the temperature was low at this time of day, the GC power was low also. The hot season has the highest power of the GC. Significantly, the investigation revealed that the power required by the gas compressor to move the gas under varying gas temperature increases by 3.4% for every 1% rise in the gas temperature. It implies that a gas temperature of 17.22°C with a corresponding gas compressor power of 10291.01kW will have a corresponding increase of 17.39°C and 10651.19kw respectively, when the gas temperature increased by 1%.

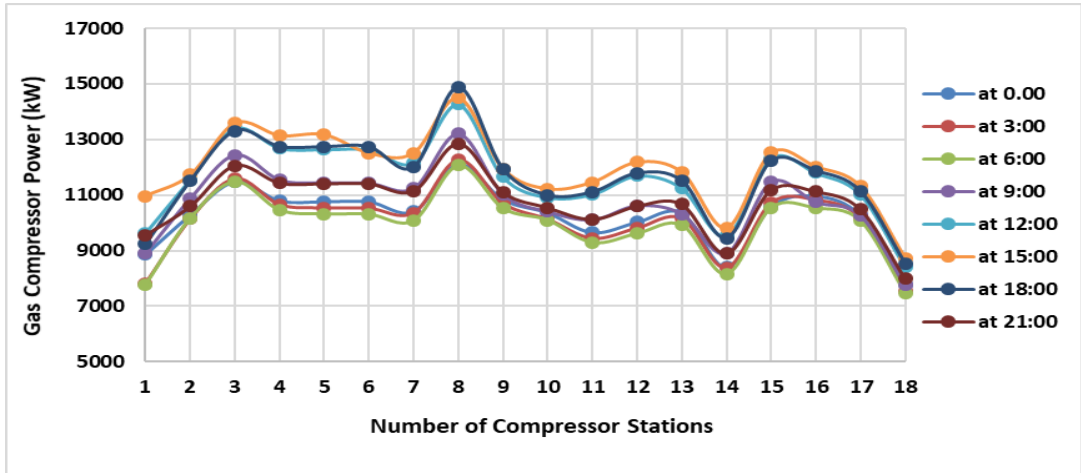


Figure 3-13 Gas compressor power at varying ambient temperature (RS)

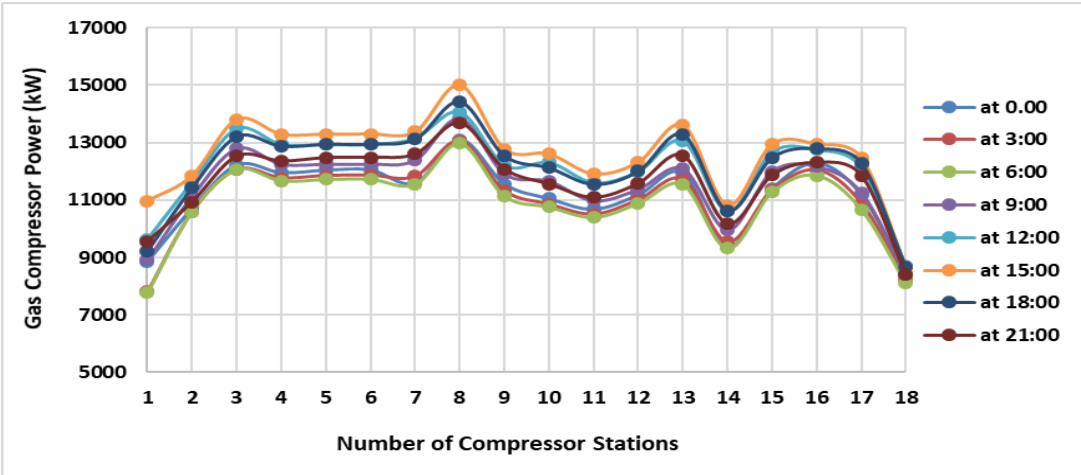


Figure 3-14 Gas compressor power in a varying ambient condition (DS)

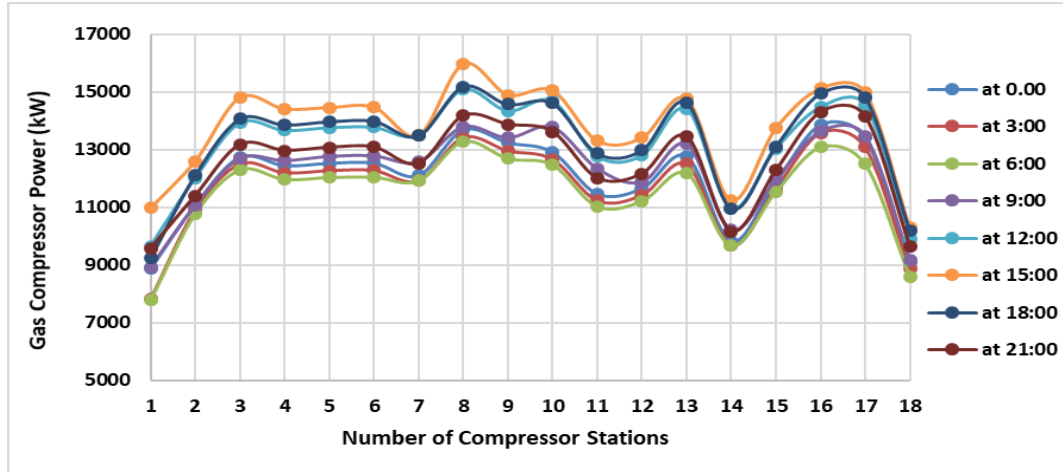


Figure 3-15 Gas compressor power at varying ambient condition (HS)

3.5.9 Effects of Suction Pressure and Ambient condition on Gas compressor Power

Significantly, gas compressor power varies in each of the stations as a result of the variation in ambient temperature. Using station two as a case study, Figure 3-16 to Figure 3-18 shows the effects of variation in Inlet temperature on gas compressor power. It was observed that as the inlet temperature increases, the gas compressor power increases also. This is because of the increase in the polytropic head as a result of the increase in the ambient temperature, though, the increase is not linear because of the non-linearity in the ambient temperature measurement with the time of day. The highest power of the gas compressor was seen at 15:00 producing 11723 kW and the least power was seen at 6:00 producing 10169kw for winter season while dry season as shown in Figure 3-17 has the highest and least power at 15:00 and 6:00 hour at 11854 kW and 10596 kW respectively. The hot season has 12,606 kW and 10596 kW for 15:00 and 6:00 hours respectively as represented in Figure 3-18, indicating that the hot season has the highest power requirement of the gas compressor as previously explained.

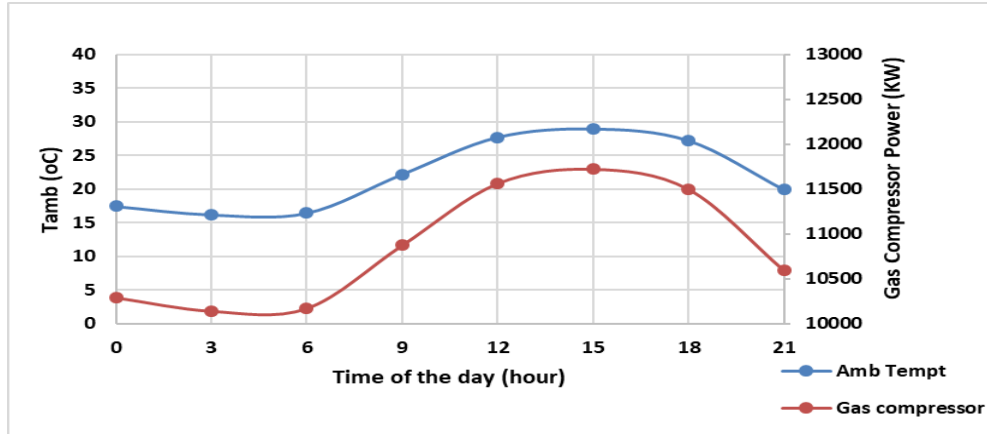


Figure 3-16 Effect of ambient temperature on GC power (WS)

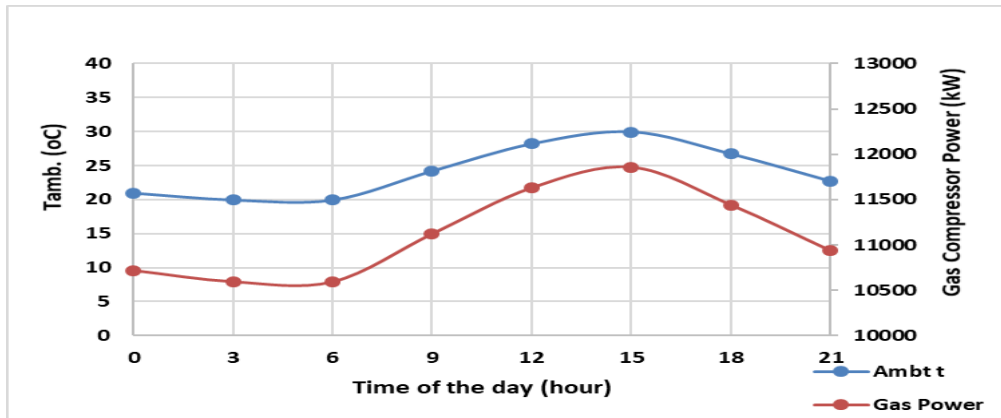


Figure 3-17 Effects of ambient temperature on GC power (DS)

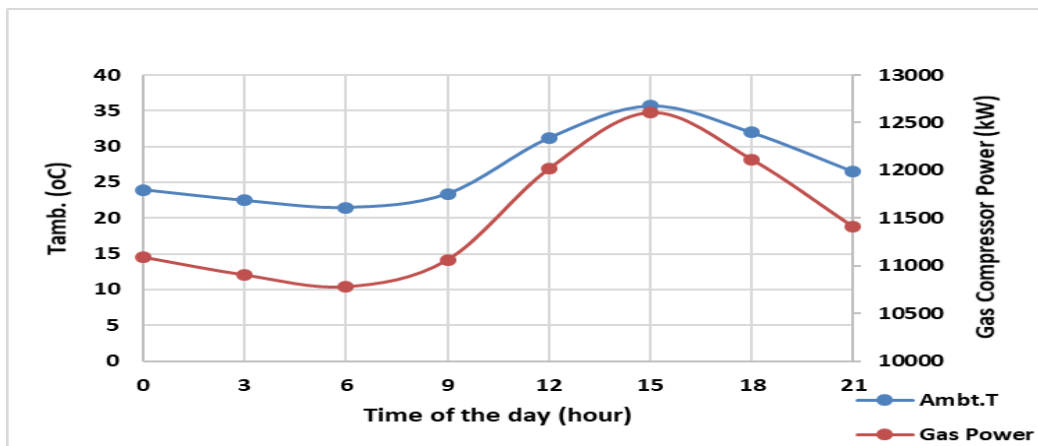


Figure 3-18 Effects of ambient temperature on GC power (HS)

Figure 3-19 to Figure 3-21 showed the effects of variation in Inlet pressure on gas compressor power in all the seasons. As the inlet pressure increases, the gas compressor power decreases, the reason is that the polytropic head decreases with

a decrease in inlet pressure of the gas compressor. Also, it affirmed the fact that an increase in ambient temperature results to increase in GCs power and a corresponding decrease in inlet pressure of the pipeline in real operation.

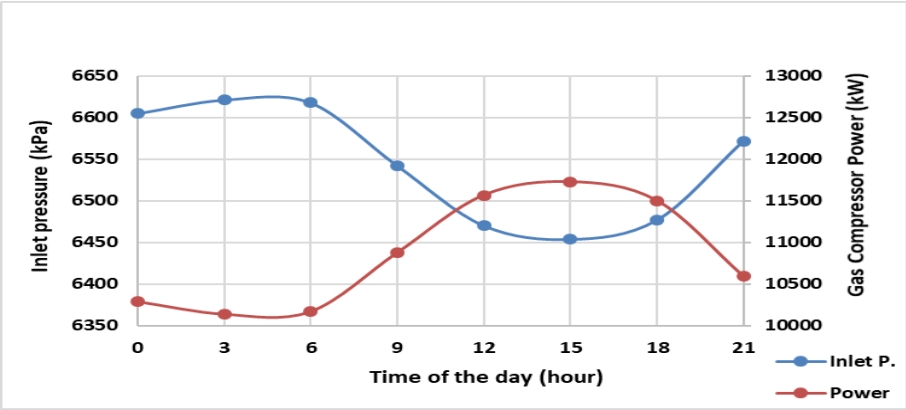


Figure 3-19 Variation in inlet pressure with GC power (WS)

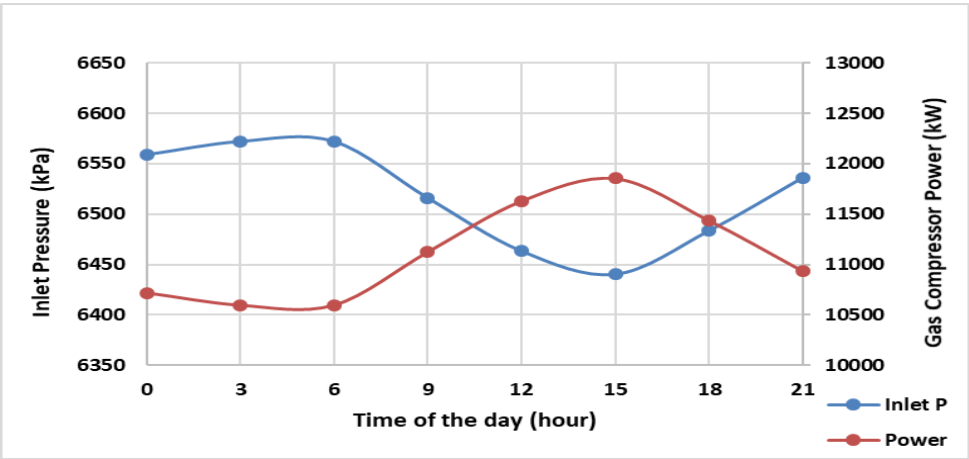


Figure 3-20 Variation in inlet pressure with GC power (DS)

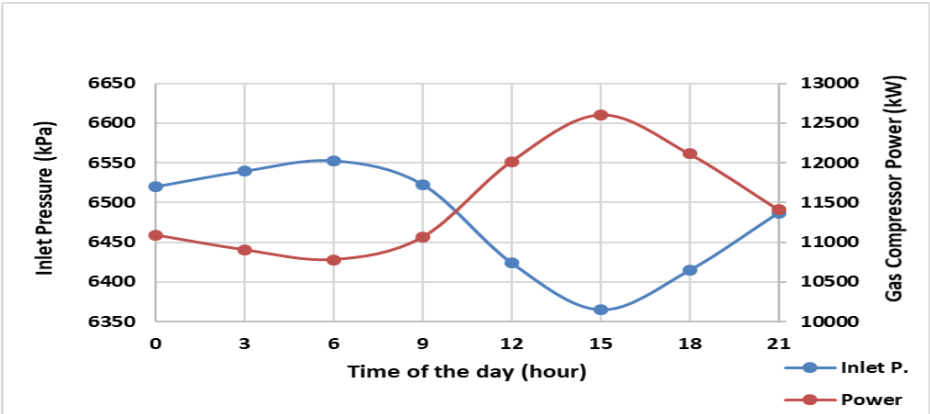


Figure 3-21 Variation in inlet pressure with GC power (HS)

Figure 3-22 showed the variation in the inlet pressure at constant discharge pressure (9500kPa) with respect to time of day. This is the MAOP required to deliver the required volume of gas through the pipeline. It is worthy of note that the least suction pressure was seen at 15:00 in all the seasons with a value of 6453.83kPa, 6440.69kPa and 6364.97 kPa for rainy, dry and hot season respectively. These correspond to the temperature values of 28.69°C, 29.69°C, and 35.41°C, respectively. It implies that the gas compressor is required to do more work at this time in order to deliver the required volume of gas. RS, DS, HS and AS on the curve represents "rainy season", "dry season", "hot season" and "all season" respectively."

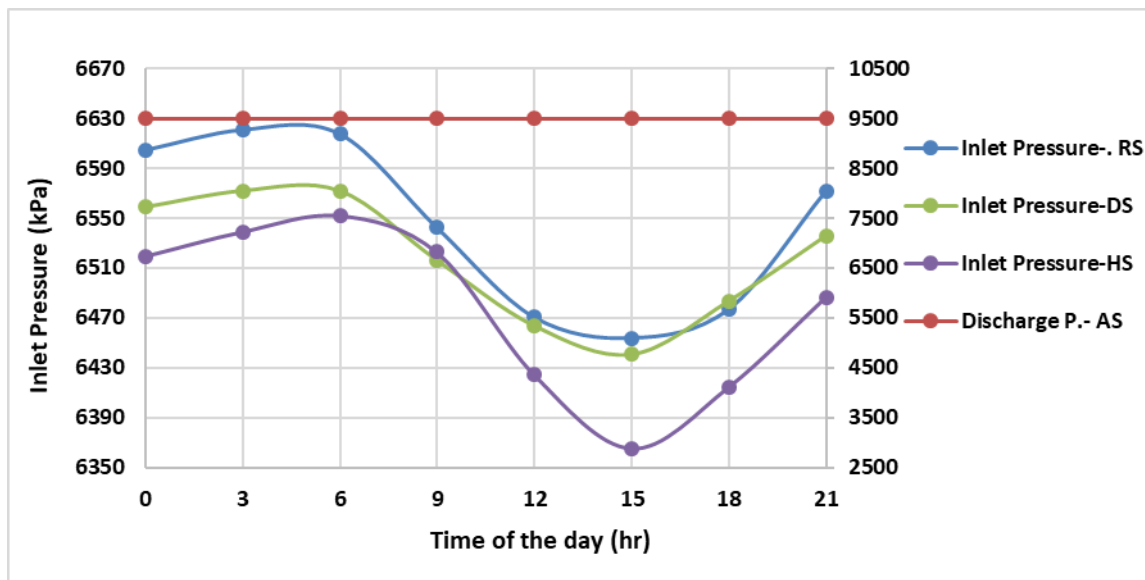


Figure 3-22 Variation in inlet pressure with constant discharge pressure

3.5.10 Gas Compressor Highest Power for 18 Compressor Station

Figure 3-23 shows the highest gas compressor power for the three seasons under investigation for all the eighteen compression station. Hot season shows the highest power requirement of the Gas compressor at 15:00 time of day. It was based on this highest power at 15:00 for each season of the years that the prime mover was selected. Considering the high volume of gas to be delivered per annum and the low compression ratio of the gas compressor, 4 GC is recommended to be placed in each of the compression stations across the entire pipeline. This will enhance adequate delivery of the gas. because of the little value of PR of the individual GC

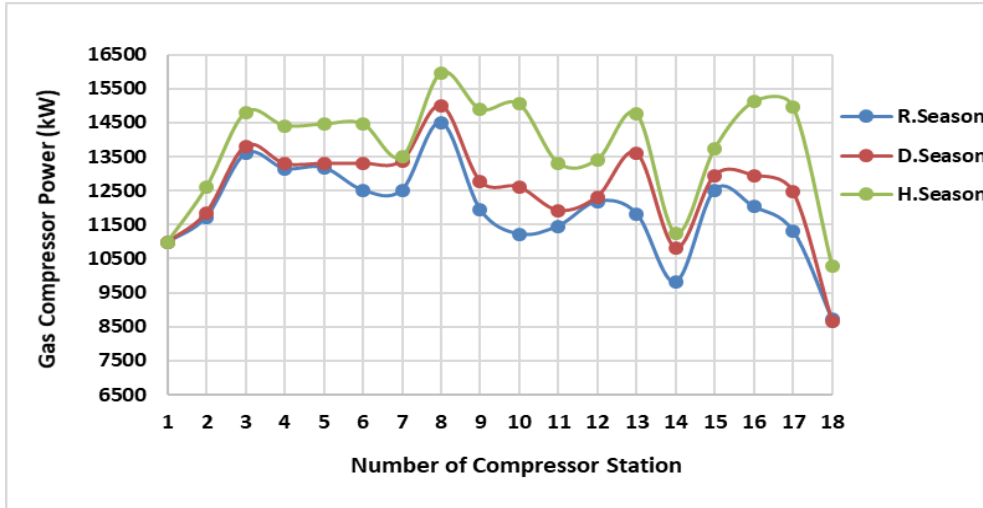


Figure 3-23 Gas compressor power for three seasons of the year

3.5.11 Centrifugal Compressor map on Pipeline Compression System

In order to keep constant gas transportation along with the pipeline network, booster stations are needed. The main equipment used in the booster station is the gas compressor. The result hereunder shows the dimensional operational characteristic curves of the gas compressor used on the pipeline network.

Based on the methodology on how to develop the characteristic curve, Figure 3-24 to Figure 3-26 shows the typical performance characteristics of the centrifugal compressor. All the important parameters used to define the performance of a gas compressor and its selection are represented on the characteristics curve. The map was scaled from the default OEM map. The map shows the plot in a variation of the pressure ratio (PR), polytropic efficiency and discharge temperature at various constant rpm with respect to the actual volume flow rate.

Figure 3-24 illustrates the newly scaled map that shows the relationship between the PR and the actual inlet volume flow. The map shows that an increase in the rotor speed gives a correspondence increase in the actual flow rate of the compressor. An increase in flow rate at constant rpm occurs when there is a decrease in the resistance at the compressor exit. An improvement of compressor flowrate at constant delivery pressure is also possible. This requirement, however, can be met by not only changing exit throttle but also increasing the compressor rpm. Also, it was observed from the map that as the actual volume flow increases, the PR also

increases up to the point of maximum efficiency where it reaches its highest value. At this point, if the flow goes further can result in a fall in pressure ratio. The surge line which traces all the constant speed line is shown on the map in addition with the compressor operating point. It is important to state that the surge line of the compressor indicates the minimum flow allowance during operation to avoid compressor damage as a result of vibration.

Figure 3-25 shows the polytropic efficiency with the actual flow. It was observed here also, that at a given speed, the polytropic efficiency varies with the actual flow. However, at all speed, the maximum efficiency value tends to be approximately the same when all the speed lines are represented on the map. In the same approach, Figure 3-26 also illustrates that as the speed increases, so also the actual flow which must correspond to their discharge temperature, respectively.

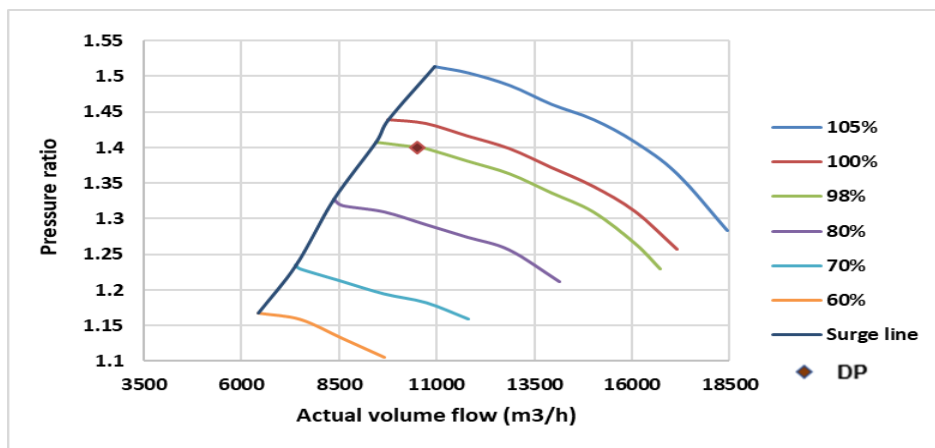


Figure 3-24 Pressure ratio-volume flow curve

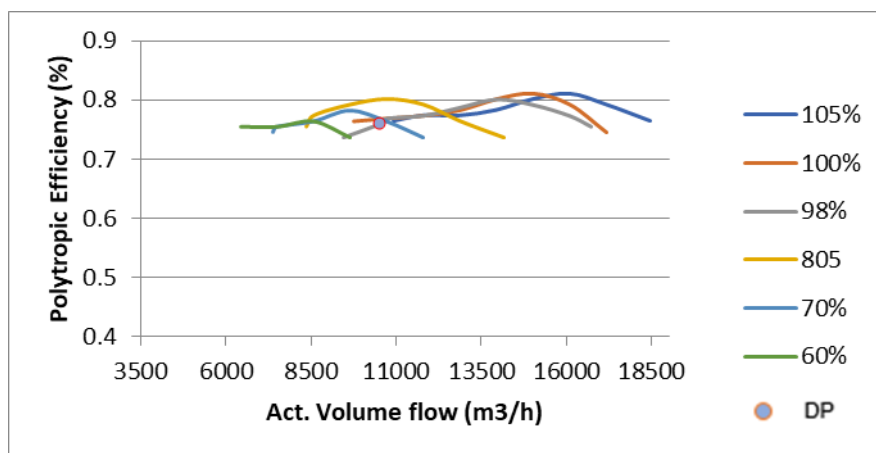


Figure 3-25 Polytropic efficiency-volume flow curve

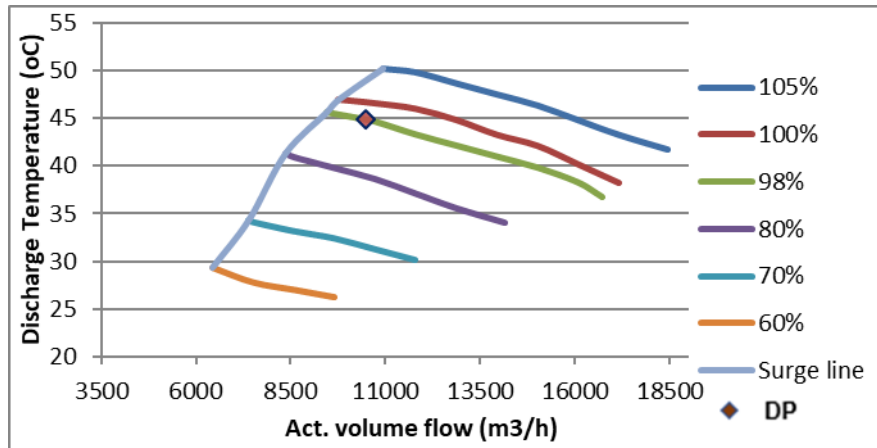


Figure 3-26 Discharge temperature-volume flow curve

3.5.12 Effects of molecular weight and Gas temperatures on Gas compressor performance

The various curves illustrated above expressed the gas flow under constant operating point with the assumption that the molecular weight of the gas remains unchanged, Figure 3-27 shows the conditions were different gas molecular weights were used for the analysis. From the curve, the various molecular weight values of 15.55, 17.54 and 19.57 were used to investigate its impacts on the polytropic head of the GC with respect to gas flow. The result showed that natural gas with higher molecular weight exhibits lower polytropic head when compared with the gas with lower molecular weight. This validated the fact that the polytropic head has an inverse relationship with molecular weight. So, as the gas molecular weight increases, the polytropic head reduces, this gives a different surge line and operational point as shown on the curve. Again, the higher polytropic head moves the operating point on the higher speed line. This implies that the power required of the GC to transport the gas will fall at an increasing gas molecular weight when compared with the lower molecular weight of the gas.

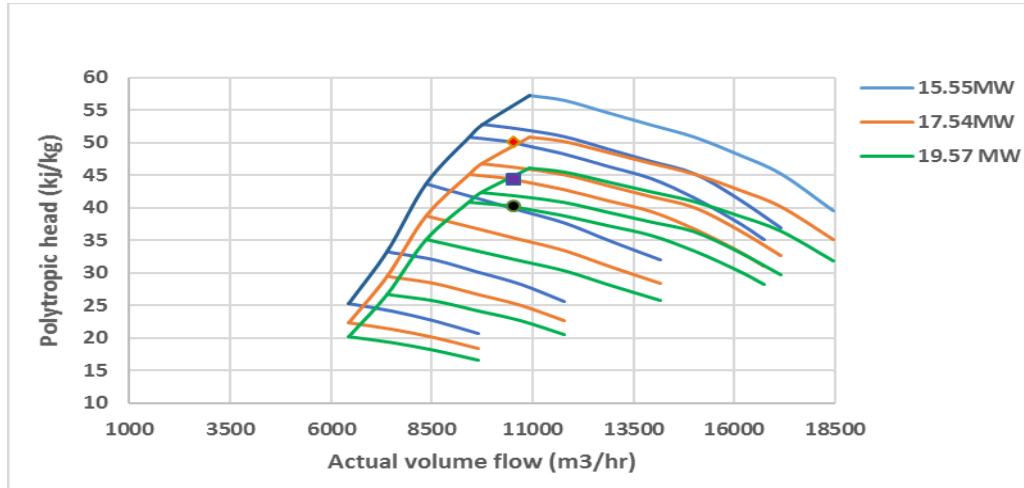


Figure 3-27 Effects of gas molecular weight on the polytropic head

In the oil and gas industry, the operation of gas compressors (centrifugal compressor) is within a specified envelope, which comprises of many operating points. The difference in the operating point is due to the changes in process conditions. One of the process conditions is the variation in gas temperature or ambient temperature. In view of this, three conditions of operation were examined in the flow through the GC. Condition one (SC.1 OPT) examined an Isothermal condition with a gas temperature of 293.87K. Condition two (SC.2 OPT) examined the average of all the gas temperatures between the isothermal and non-isothermal condition, this having an average value of 307.93K. While condition three (SC.3 OPT) examined the average of the non-isothermal gas temperatures having an average value of 321.98°k all at the same gas composition. These conditions were used to investigate their effects on the polytropic head of the GC. These conditions of operation were chosen because it captures the entire time-based gas temperature of the pipeline. Only compressor station two was used as a reference station for this analysis. The approach can be replicated to all the CS.

The effects of the variation in these gas temperatures on the polytropic head of the GC was evaluated, and the result is represented in Figure 3-28. The result revealed that the polytropic head increases with an increase in gas temperature. This makes the operating point of the three conditions to change. This analysis is important because the gas temperature is a strong factor that influences the performance of the centrifugal compressor and the gas turbine shaft power. The result shows that for every 1% increase in the gas temperature, the polytropic head increases by

0.15% in all the conditions. This result authenticated the facts that increase in the gas temperature result in an increase in polytropic head and decrease in prime mover power. Figure 3-29 showed the effects of different gas temperature on the gas compressor power at constant volume flow. The result showed that the gas compressor power experienced a proportional increase with the increases in the polytropic head. From the result, the SC.1 OPT, SC.2 OPT and SC.3 OPT had different operating points, which are 9287.9kW, 9732.16kW and 10176.4kW, respectively with a constant volume flow of 10510 m³/hr. The result further affirmed the fact that gas compressor power increases with an increase in gas temperature, hence the different operating point. The result of the analysis revealed that the designed gas compressor is capable of operating within the given condition of operation.

It is recommended that the variation in the power for both the driver and the driven has to be balanced so that a steady flow across the pipeline can be maintained. This is because gas temperature variation in pipeline flow through GC is a natural phenomenon that will happen during the gas compression process.

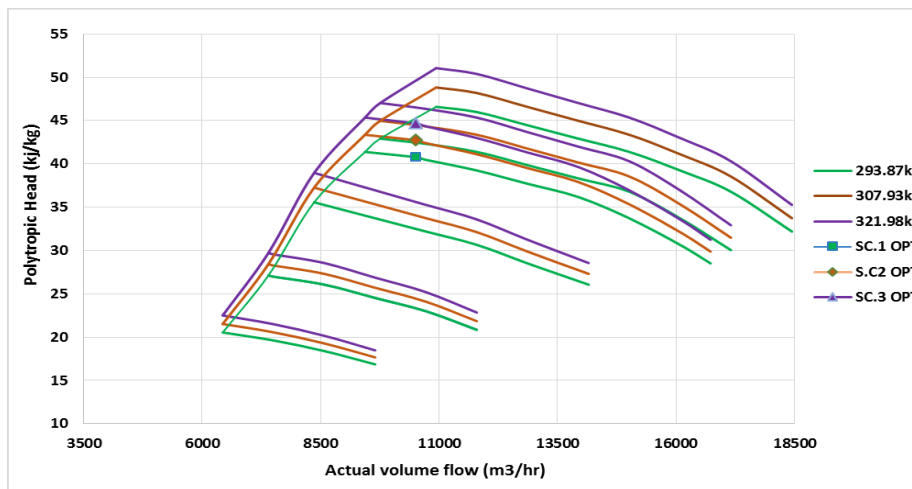


Figure 3-28 Effects of varying gas temperature on the polytropic head

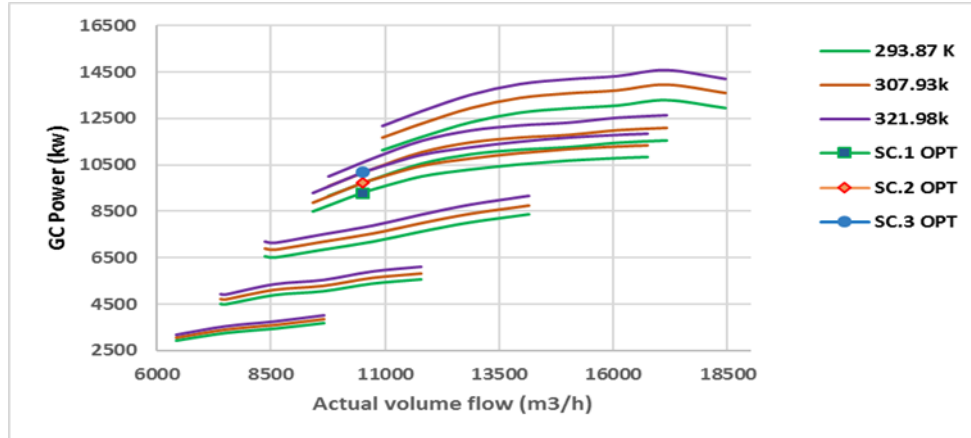


Figure 3-29 Effects of varying gas temperature on GC power at a constant volume flow

3.6 TERA framework for the natural gas pipeline application

Considering the operation of the eighteen compressor station amid harsh ambient temperature and engine degradation in quantifying the overall investment cost of this project, an approach that can compress all variables becomes very important.

TERA concept, as mentioned in Chapter 2 of this thesis, become useful in addressing the aim of this research. **Figure 3-30** presents the pipeline techno-economic and environmental risk assessment framework produced for the present study to evaluate the investment cost (life cycle costs (LCC)) and further evaluate the net present value (NPV) of the natural gas pipeline project using the TERA approach. Various modules such as pipeline, gas compressor, engine performance, emission, lifing, and the economic module are been utilised by the developed TERA framework. The framework also illustrates the various performance parameters that are required by the economic model in order to calculate the LCC and NPV. The investment cost for the natural gas pipeline investigation was achieved under the influence of degraded gas turbines. The degraded mode of the engines was divided into three scenarios. The scenarios include optimistic (opt), medium (med) and pessimistic (pess). These scenarios define the stages or levels of annual engine degradation. That is, from bad to worst case of engine degradation during operation.

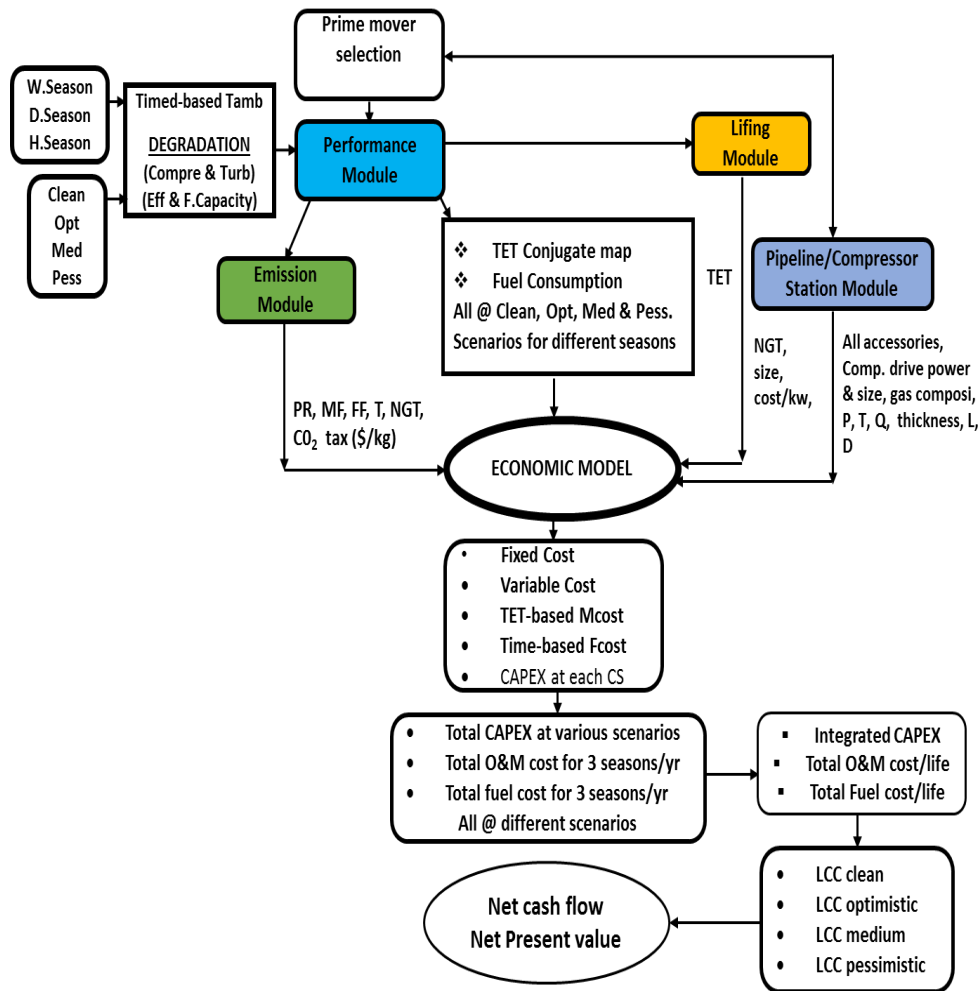


Figure 3-30 TERA framework for pipeline natural gas investment

Based on the developed TERA framework as shown **Figure 3-30**, each of the module as identified in the framework are discussed below

3.6.1 Engine Performance Module

The performance module serves as the fundamental of TERA framework. TURBOMATCH simulation software was used to simulate the engine thermodynamic behaviour at both design point (DP) and off-design (OD) conditions. The software was developed in FORTRAN language for several years ago and has continuously been improved upon over the years[169]. One of the significant merits of the code is in its modularity which enables any engine configuration to be simulated. It also analyses the performance degradation (fouling) of the gas turbine, which was useful in this research.

The result of the performance model was linked with other modules such as emission, lifing and pipeline/compressor station module. Based on this linkage, the performance module becomes the key element in the TERA concept. During the simulation process, both the compressor and turbine map of the input file is been scaled by TURBOMATCH while the OD points are been calculated based on the environmental conditions. In this research, the effects of time-based ambient temperature and degradation on the engines are been simulated. The present study also categorised the engine degradation into three scenarios (optimistic, medium and pessimistic scenarios) as indicated in Figure 3-30 The impacts of these scenarios are discussed in chapter 5. The performance module supplies the result of the different scenarios under investigations and links them to the economic module. The performance parameters include the TET based on the different number of GT engines used at different scenarios (conjugate map), the time-based fuel consumption in each of the compressor stations, the number of GT engines used in each compressor station etc,

Project life was thirty years. The performance result of each season of the year with respect to time of day is fed into the economic model where sequential calculations per seasons are been done by the developed economic model. The model was developed in MATLAB based on the result from the performance module, necessary data from gas compressor module, pipeline and the various accessories of the pipeline.

3.6.2 Lifing Module

The factors that limit the working cycle of an engine are the disks and high-pressure turbine and are been influenced by the operating TET. Because of this, the stress on the turbine blade and disk are been impacted as the operating TET varies. The operation affects the engine's creep life. Therefore, evaluating the remaining life of the engine will be reliant on the operating TET.

In this research, a simple creep life model was developed. This model uses a simple relationship between the engine creep life and the TET. This relationship was used to produce trend line polynomial function with an RMS value of 0.998. The trend line was further used to develop the lifing model in MATLAB to estimate the creep life of the engines in a time-based system at various seasons of the year throughout the

project life of thirty years. This technique was adopted because the creep life has a direct relationship with the engine TET[170]. It is essential to state that as the engine operational TETs increases, the creep life reduces, vice-versa. The simple developed model assumed that for every 20K rise in the operational TET, the engine life reduces by 50% and uses the DP TET as the reference TET with the corresponding assumed creep life. The assumption used to develop the creep life model was based on the research work of Gad-Briggs et al. [170] which investigated the relationship between the TET and creep life of the engine and presented that for every 20K increase in TET the HPT blades lifetime decreases by half. Also, researchers like Obhuro[68] have used this principle for a simple lifing mode. This justifies the selection of the model. However, the implications associated with the simplified assumption could be that the exact creep life of the engine may not be accurate but approximately reasonable as Miner's law/ Larson miller parameters were not taken into consideration. Hence, lifing model that will take into account the Miner's law/ Larson miller parameter has been recommended

Furthermore, 25000 hours was also assumed to be the DP creep life that corresponds with the DP TET of the selected mixed fleet engines. Finally, the model established that the maximum operational engine creep life should not exceed 25,000 hours. Although, there was no possibility of decreasing TET due to degradation and harsh ambient temperature at fixed power output. This is because The TET increases to maintain the power output as the engine degrades with time. However, increasing the number of engines whenever the limit set on the TET was exceeded will enable the engine to operate on part load, thereby allowing the engine to run on a lower TET. This technique will increase the creep life of the engine. Therefore keeping the creep life within the engine DP will enhance engine availability and also avoid unnecessary upsurge of creep life at some points during the calculation process. This will enhance uniformity during the model calculation. Also, the least creep life was kept at the least limit set on the operational TETs.

Creep life of the engines was estimated based on the scenarios under investigation (level of degradation) throughout the project life. The estimated creep life at different scenarios was used to calculate the major maintenance cost of the mixed fleets of engines at various seasons of the year throughout the project life and at all

compressor stations. The annual maintenance cost serves as one of the key factors in the economic investigation of the natural gas pipeline investment.

3.6.3 Economic Module (EM)

EM is another core module of the TERA framework that accommodates all other modules. This module handles the calculation of the entire project life cycle cost, simulate the cash flow and finally evaluate the benefits of investing in the project. In this research, the economic module evaluates the results from the various module based on the variation of the ambient temperature with respect to time of day (time-based system) and the impacts of GT degradation on the investment cost at different seasons (rainy, dry and hot season) of the year. This ambient temperature for the entire 18CS was segmented based on three-hours different in each season of the year. The impacts of TET on the overall maintenance cost and the corresponding cost of fuel consumption were also evaluated in the module. It also calculates the economic sensitivity analysis of the pipeline investment with respect to fuel and O&M cost.

Finally, the effects of engine divestment and leasing due to redundant fleets of engines after overhaul are been evaluated by the economic module. It implies that after overhaul, the total number of engines used as a result of degradation in the various booster stations were reduced. The redundant fleets after overhaul were divested through sales or leased. The developed economic model evaluates the influence of engine divestment in one case and leasing in another case on NPV. The case of divestment and leasing makes the research to have two baseline result The economic model was developed in MATLAB based on the result from the performance module, necessary data from gas compressor module, pipeline and the various pipeline accessories.

On the economic model capability, the developed model was able to select the number of GT in each CS and their corresponding power as it continues to change with time due to degradation. This analysis was done in each scenario and each season of the year in all the CS for thirty years of operation. Also, the economic model was linked with the TURBOMATCH result to evaluate the investment cost using the performance result obtained from the TM code

3.7 Conclusion

The fundamental approach in evaluating the pipeline performance and to calculate the power of GC for natural gas pipeline export service has been presented. With the available data from the TSGP report, the inlet and the outlet condition of the pipeline have been analysed, taking into account the distances and elevation between each compression station. More importantly, the pipe exit pressure has been evaluated for all compression stations taking into account all necessary parameters to be considered. Knowing the exit and inlet pressure of the pipeline can enhance pressure loss evaluation in the pipeline system. The correlation between the pipeline and gas compressor shows that the outlet condition of the gas compressor (compressor station) becomes the inlet condition of the pipeline, while the outlet condition of the pipeline during process compression becomes the inlet condition of the gas compressor (compressor station). Based on this condition, the pipe inlet pressure was constant throughout the pipeline compression process at 9500kpa. This is because the discharge pressure of the gas compressor was fixed at 9500kpa. It was also assumed in this research that the gas composition remains constant, taking into account that the pressure drop within each of the pipe segment depends largely on the pipe length, elevation and site temperature, However, the effect of varying the gas molecular weight was evaluated on the GC map. The result authenticated the facts that as the molecular weight of gas increases, the polytropic head reduces which has impacts on the gas flow through the pipeline and the power required of the GC. Furthermore, other operational maps for the newly designed GC were developed and presented for both single and different operating points. The effects of varying the gas temperature on the polytropic head and GC power have been analysed and presented, which leads to a different operating point on the GC map.

Also, The various modules of the developed TERA framework have been discussed, and the core module has been identified as the performance module. The analysis also shows the significant of TURBOMATCH software in this investigation. Due to the involvement of the multi-variables in this research and the number of years of project life, a code that can connect the performance results have also been developed in MATLAB. The details are discussed in the subsequent chapters.

4 Gas turbine Selection, Modelling and Performance Simulation

4.1 Overview

This chapter presents the selection and modelling of the gas turbine engines based on the calculated gas compressor power. Also, assessment and validation of the model were carried out. For instance, subjecting the model at various operating conditions such as ambient temperature and altitude. Also, the impacts of degradation at the preliminary stage on the engines was looked into and presented. The selected engines models were developed based on public domain specification, using TURBOMATCH. At the preliminary evaluation of the developed model, 3% reduction in flow capacity and isentropic efficiency was implanted on the turbomachinery component of the engine model at different operating conditions to evaluates the overall engine performance. Also, 1% to 5% reduction in flow capacity and isentropic efficiency were implanted on the engine when investigating the engine parameter deviation from their DP at a specific ambient temperature and TET. Three cases of the investigation were examined on the engine. Case 1 examined when “only compressor” was degraded, case 2 investigated when “the turbine only” was degraded and case 3 examined when “both compressor and turbine” was degraded to evaluate their impacts on engine parameters to see how they deviated from their DP.

4.2 Introduction

Selection of appropriate prime mover for a particular project taking in to account the environmental conditions and the nature of projects is very important for smooth operation. In this research, GT engines are used as the prime mover. The configuration of GT engines ranges from single shaft to multi-shaft engines, as discussed in chapter 2. Kurz et al. [171] presented that two-shaft configuration of a gas turbine is better in an application for pipeline compression due to its flexibility in speed during operation.

GT efficiency, fuel flow, and shaft power of any installation are cardinal performance parameters, which directly or indirectly affects the techno-economic of a compressor

station project utilising GTs engines. The industrial GT utilised for pipeline compressor station comprises an air compressor, combustor, gas generator turbine, and power turbine. The combustor, which serves as the combustion chamber, receives high-pressure air generated by the compressor mix with fuel and burn in the combustor. The compressor turbine receives hot gases from the combustor. The high temperature, high-pressure combustion products expand in the gas generator turbine; the primary purpose is to provide the power required to turn the pipeline compressor. Then the power turbine receives the hot gases from the gas generator for further expansion. During the operation, the speed of the pipeline compressor and the power turbine are the same, independent of the gas generator turbine.

Generally, the TET and gas turbine speed increases with an increase in the fuel flow, until one of these parameters reach its operating limit. Accordingly, this will provide the power turbine with gas at higher temperature, pressure and mass flow and this help the power turbine to produce more power. Again, the high temperature and high-pressure gasses from the combustion chamber are then expanded in a gas turbine generator to extract useful power. The compressor absorbed part of the turbine power out to provide power for the compression process through the shaft connecting the compressor and turbine. The remaining power out from the turbine is used to drive the pipeline centrifugal compressor. The pipeline compressor and the power turbine run at the same speed, independent of the gas turbine generator.

4.3 Gas turbine Selection

The gas turbine engines are selected based on the power of the gas compressor. After the engine selection, the thermodynamic model was produced. The modelled engine will be simulated and validated. The off-design analysis was carried out on the engines at the different operating conditions. The OD simulation will form the preliminary result of the developed model. The following criteria determine the gas turbine selection:

- (i) Adequate power to meet up the gas compressor DP power requirement after de-rating to the three seasons ambient condition
- (ii) Engine configuration was considered as each configuration has different part-load performance,
- (iii) Sufficient data available in the public domain

Four GTs that are inspired by SGT-400, SGT-500, GT35 and LM1600, referred to as IGTA, IGTB, IGTC and IGTD respectively was selected based on the above criteria as represented in Figure 4-1. The engines were modelled and built based on the information gathered from the public domain of the inspired engines. It is worthy of note that a 10% margin on the power of the gas turbine selected for the compressor drive as required by API 616 was considered[172]; [173]. Meanwhile, all GTs selected are multiple shaft engines consisting of a gas generator and a free power turbine. Studies have shown that multiple shaft machines like the ones selected are better than single shaft engines for mechanical drive applications. The primary reason is that the gas generator speed of two or three shaft engines, when compared with a single shaft, is not tied to that of the driven load. Therefore, it is more appropriate to handle power demand fluctuation and speed variability.

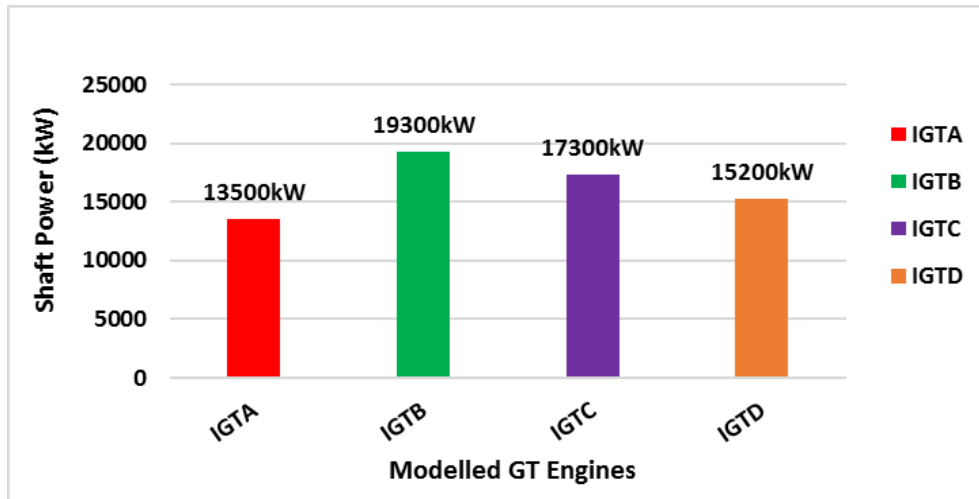


Figure 4-1 DP power of the selected engines

4.4 Performance Modelling

The engines under investigation were modelled and simulated using the simulation software. The design point (DP) simulation was achieved by varying the TET, component efficiencies and mass flow in order to arrive at the values similar to the one in the public domain. The off-design performance of the engine was considered as any deviation from the main DP as a result of any internal alteration of the engine components which could cause by degradation or the alteration that occur externally such as changes in altitude and ambient temperature[174]. The present study has selected some engines of different power output, configuration, and with good

thermal efficiency to meet up the power required by the gas compressor. These engines are considered below. It is important to state that when modelling the off-design condition of the engines, it is necessary to select an engine variable as the “handle”. The handle will help to determine the matching conditions of the engine. The possible variables are rotational speed, TET, fuel flow and exhaust gas temperature (EGT). However, in this chapter and for the preliminary result of the modelled engines, TET was chosen as the handle with different operating conditions like changes in ambient temperature and altitude.

4.4.1 Component Map

The gas turbine engine is a modular designed engine. The most crucial turbomachinery component of the GT is the compressor, combustor and the turbine. These components are inter-related for the adequate performance of the gas turbine. In this research, the compressor maps are presented from the simulation result. The performance of the compressor was evaluated based on the dependence of pressure ratio and isentropic efficiency on the non-dimensional mass flow at different operating speed.

The following are the governing equation for the performance parameters

$$PR = \frac{pr_2}{pr_1} \quad (4-1)$$

Where,

PR= Pressure ratio; pr_2 = Compressor Output Pressure; pr_1 = Compressor Inlet Pressure.

$$CMF = \frac{W \sqrt{\frac{TA}{TA_{ISA}}}}{\frac{PA}{PA_{ISA}}} \quad (4-2)$$

Where,

CMF – Corrected mass flow;

W- Inlet mass flow

TA- Compressor inlet Temperature

T_{AISA} - Compressor Inlet Temperature at ISA condition

P_A - Compressor Inlet Pressure

P_{AISA} - Compressor Inlet Pressure at ISA Condition

$$PCN = \frac{N \sqrt{\frac{T_{ISA SLS}}{T_A}}}{N_{DP} \sqrt{\frac{T_{ISA SLA}}{T_{A_{DP}}}}} \quad (4-3)$$

Where, PCN- Non-dimensional Speed line

4.4.2 Map scaling and Deteriorated mode of the engines

The component maps are scaled in order to relate to a certain size of the component. Scaling of the components takes place during design point calculation that is, the first run. The calculated scaling factors are stored and applied to all subsequent off-design point calculations.

As explained in chapter 2, degradation in the gas turbine is natural during operation. Degradation occurs in the Compressor, Combustor and the turbine. Physical degradation is openly related to the deterioration of performance. When the engine begins to work for the first time, degradation starts from that time, which implies that degradation is a function of time.

When degradation occurs, the component maps are changed completely. The effect of this degradation is felt on the reduction of isentropic efficiency and flow capacity in components, mainly the compressor and turbine. In TURBOMATCH application, isentropic efficiency and flow capacity vary in order to simulate the degraded component performance. Map scaling factors applicable in TURBOMATCH enable maps of degraded components to be simulated. In both the compressor and turbine, it is possible to change the pressure ratio, isentropic efficiency and mass flow. Rate of enthalpy change (DH/T) can still occur, though only for the turbine. The effects of these changes generally affect the overall performance of the engine like the shaft power output, thermal efficiency, fuel flow. Deterioration is introduced by reducing the scaling factors calculated in the design point. On the TURBOMATCH code, for 1% deterioration, a factor of 0.99 is been introduced, hence subsequent reduction in

deteriorated value for a specific component follow the same trend. The engine model is shown in Appendix C.

4.5 IGTA Performance Specification

Having ascertained the criteria by which the gas turbine can be selected, it becomes crucial to select IGTA, which was inspired by the SGT 400 engine. Moreover, it is a two-spool engine with a useful output power of 13.4 MW for a mechanical drive. The system has an efficiency of 35.4% for operation in a simple gas-fired power. It also has a great power-to-weight ratio. The two-spool structure delivers excellent Speed and loads turndown characteristics to permit maximum flexibility of operation. When using for mechanical application, it drives the gas compressor directly. Gearbox connection is not necessary since the turbine speed ranges from 4800 rpm to 10000 rpm. Figure 4-2 shows the schematic diagram of the IGTA 2-spool engine with a free power turbine. While Table 4-2 shows the performance specification of IGTA

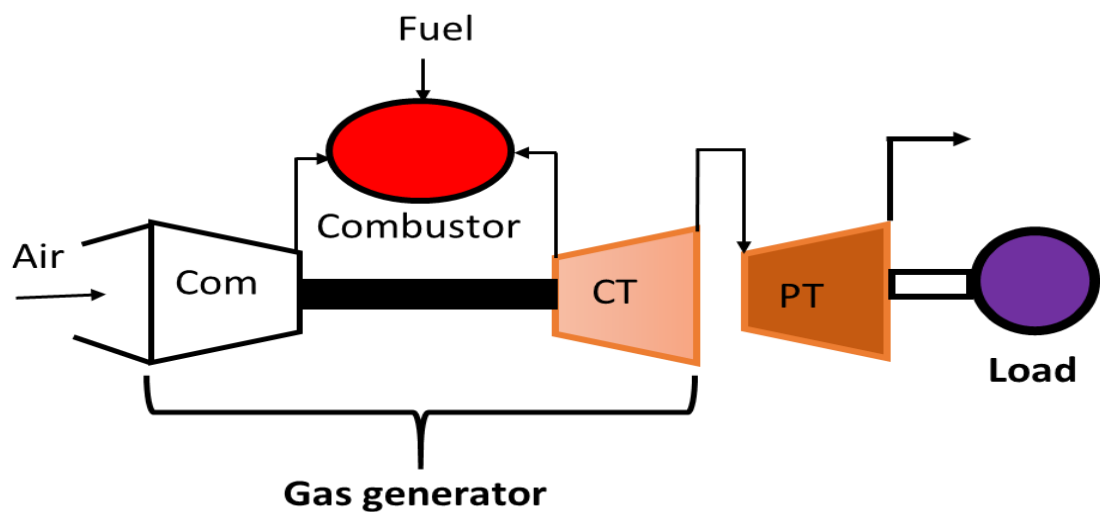


Figure 4-2 Schematic diagram of the IGTA 2-spool engine

Table 4-1 shows the specified parameters of the modelled engines at ISA condition, while Table 4-2 shows the engine model accuracy at design point when compared with the OEM data. The design point result of the key performance parameters is shown on the table. With this result, the model is validated at DP.,The OD performance analysis can be carried out.

Table 4-1 Design point parameters of IGTA engine at ISA condition

Parameters	Parameters	Values
1	Inlet mass flow (kg/s)	38.64
2	Pressure ratio	16.8
3	Shaft power (MW)	13.4
4	Compressor efficiency (%)	87
5	TET (K)	1440
6	Combustor efficiency (%)	99.9
7	Compressor turbine efficiency (%)	88
8	Power turbine efficiency (%)	88

Table 4-2 Performance validation of IGTA

No	Parameters	Reference Website	TURBOMATCH Result	% Difference
2	Thermal efficiency (%)	36.4	37	1.6
4	Inlet mass flow (kg/S)	38.6	38.64	0.1
5	Fuel flow (kg/S)	0.7653	0.7941	-3.8
6	TET (K)	-	1440	-

At DP, the drawing of the compressor map was achieved using the variation in the corrected mass flow and pressure ratio. The fort-3 result from TURBOMATCH simulation was used to plot the map as represented in Figure 4-3. A 3% reduction in flow capacity and isentropic efficiency was used to degrade the compressor.

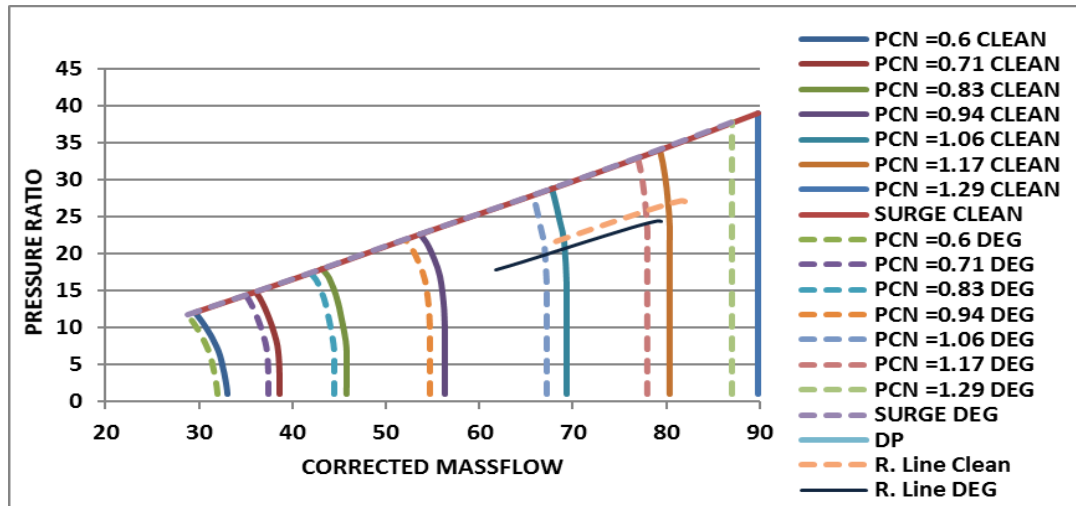


Figure 4-3 IGTA Compressor map (PR vs CM)

4.6 IGTA OD Simulation Result

IGTA engine of 13.4 MW was subjected to various operating conditions such as variation in ambient temperature and altitude. Both operating conditions were investigated with the degraded conditions. The engine was modelled in a degraded mode with a 3% reduction in flow capacity and isentropic efficiency on the compressor component only. Again, different TETs as handle were examined during both the clean and degraded engine simulation. The TETs taken were between "1350K to 1500K" with an interval of 50K. Meanwhile, the DP TET for the IGTA engine is 1440°C. The preliminary results are presented below.

4.6.1 Effect of Ambient Temperature and Degradation

The result of the clean engine was shown alongside with the degraded engine. The engine was subjected to a range of ambient temperature between "10°C to 40°C". These range of temperature for the preliminary analysis were chosen because it covers the ambient temperature where the pipeline project is located.

Figure 4-4 represents the impacts of ambient temperature and degradation on the shaft power of the engine at different fixed TET. The result shows that shaft power decreases with increase in ambient temperature. The reason is that the compressor does much work to compress hot air, which contributed to the overall shaft power reduction. Also, the density of the air reduces as temperature increases, which consequently reduced the air mass flow rate. In a precise term, the decrease in the

inlet air density as a result of elevated temperature means a decrease of mass flow into the turbine. Hence the reason for the reduction in power output.

Also, from the result, the power of the engine increases with increasing TET. As degradation occurred, the power at a particular TET reduces. This is because of the reduced mass flow and component efficiency as a result of fouling. It is important to note that reduction in shaft power is as a result of fouling and variation in ambient temperature.

It was observed that for the clean engine operating at different TETs (for example 1400K) under varying ambient temperature that for every increase in 1°C above the ISA, the engine shaft power reduced by 2.5%. However, when the engine was degraded (3% reduction in flow capacity and isentropic efficiency of the compressor) at the same operating condition, the shaft power reduced by 3.2%. The result further clarified the fact that even without degradation in the gas turbine engine, the shaft power of the engine can reduce at an elevated ambient temperature. The analysis also shows that the shaft power increases with increasing TET. Meanwhile, engine degradation reduces the engine power at various TETs as represented in Figure 4-4. Conclusively, the decrease in shaft power in a fouled engine is higher than the clean engines during operation.

Figure 4-5 shows the effects of ambient temperature and degradation on the engine thermal efficiency. In this case, as the ambient temperature increases, the thermal efficiency reduces, this is because the temperature ratio is reduced as a result of the high compression work on the high-temperature day. It is noticed here that the thermal efficiency increases as the TET increases and the temperature ratio and the pressure ratio decrease further. Furthermore, due to the reduction in flow capacity and isentropic efficiency due to fouling, there was a further reduction in the thermal efficiency of the engine. The thermal efficiency decreases with decreasing TET.

The result obtained also showed that the efficiency of the clean engine reduced by 2% for 1°C increase in the ambient temperature while the degraded condition reduced by 2.5% at the same operating condition. The result also clarified the fact that even without degradation in the gas turbine engine, the thermal efficiency of the engine can reduce at an elevated ambient temperature. The analysis also shows that the thermal efficiency increases with increasing TET. Meanwhile, engine degradation

reduces the engine thermal efficiency at the various TETs. Conclusively, the decrease in the thermal efficiency in a fouled engine is higher than the clean engines during operation.

Figure 4-6 also indicates the effects of ambient temperature and degradation on fuel flow of the engine at different fixed TETs. The result shows that fuel flow reduces as the ambient temperature increases. At lower temperature, the engine was able to reach higher power output. Based on this reason, the fuel flow was higher at a lower temperature than in elevated temperature. Also, the fuel flow decreases because of the increase in the compressor outlet temperature as a result of the increase at inlet temperature. The fuel expected to heat the air to the required TET is reduced due to the increase in outlet temperature. The fuel flow becomes more reduced in the degraded engine than in the clean engine as shown in Figure 4-6. However, for the engine to maintain a required TET, the fuel consumption will be high at the expense of the engine life.

The result obtained also showed that the fuel flow of the clean engine reduced by 0.6% for 1°C increase in the ambient temperature while the degraded condition reduced by 0.9% at the same operating condition. The result also affirmed the fact that even without degradation, the fuel flow of the engine can reduce at an elevated ambient temperature. However, the fuel flow increases with increasing TET while engine degradation reduces the fuel flow at the various fixed TET. It can also be concluded here that the decrease in the fuel flow in a fouled engine at a fixed TET is higher than the clean engines during operation.

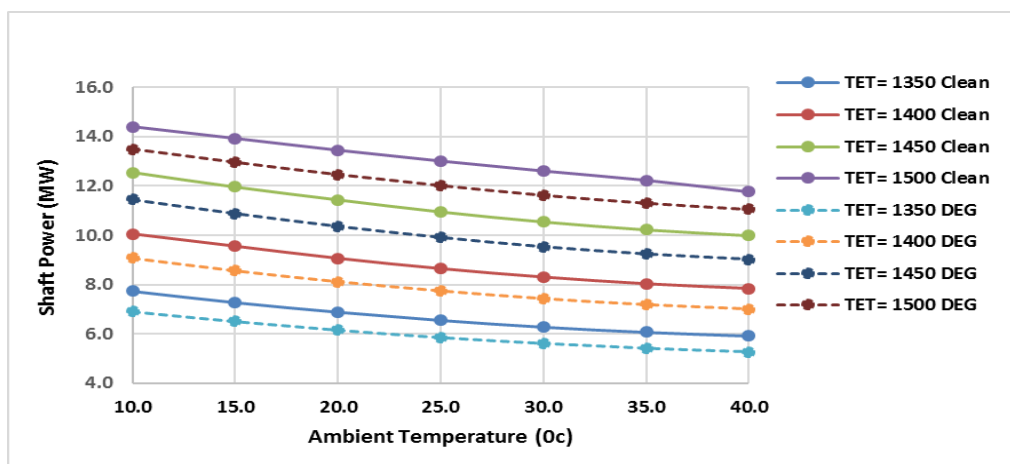


Figure 4-4 Effect of ambient temperature and degradation on shaft power

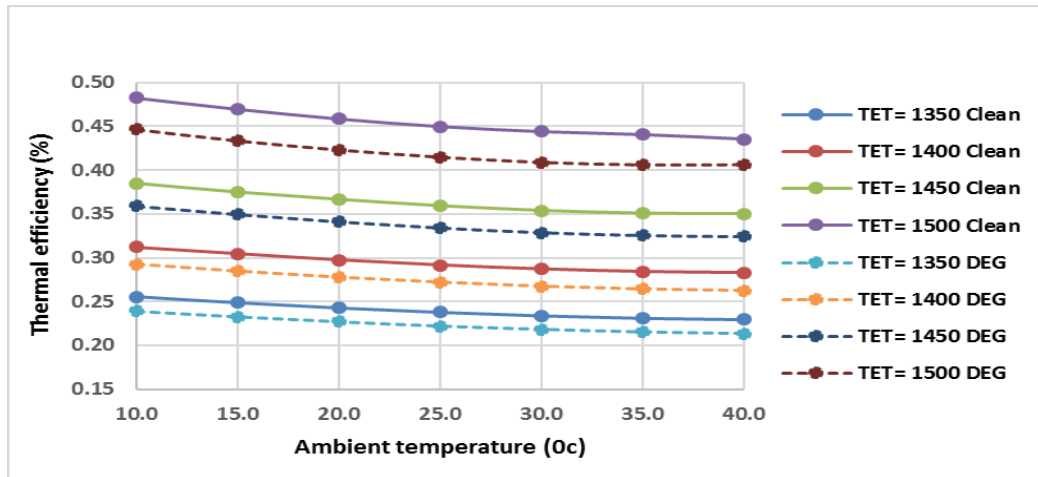


Figure 4-5 Effect of ambient temperature and degradation on thermal Efficiency

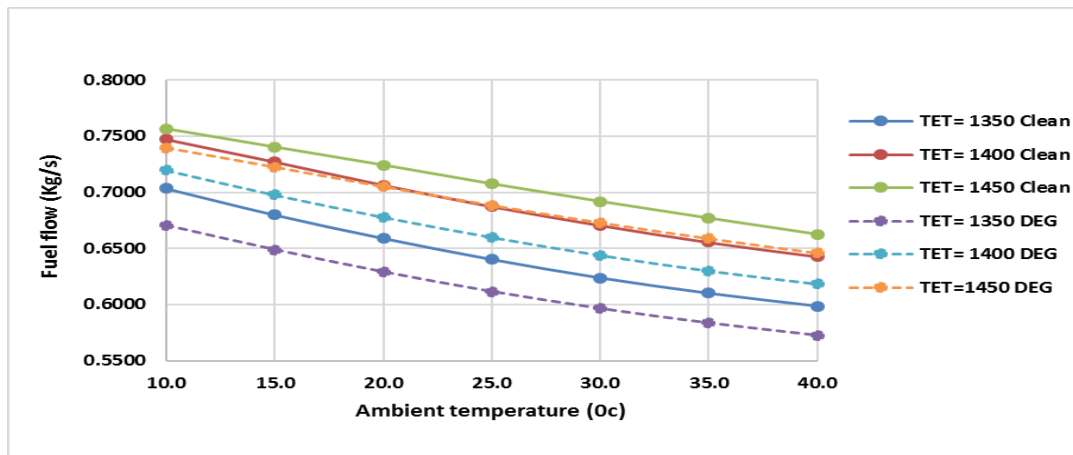


Figure 4-6 Effect of ambient temperature and degradation on fuel flow

4.6.1.1 Effect of Altitude and degradation on IGTA

The effects of altitude on the gas turbine performance is represented in Figure 4-7 to Figure 4-9. 0m to 2000m was used in the model to investigate the effects of different altitude on the performance of the GT. Also, a 3% reduction in flow capacity and isentropic efficiency was implanted on the compressor component only.

Figure 4-7 shows the influence of altitude and degradation on the engine shaft power at different TET. The result shows that power reduces as the altitude increases. Even if the temperature decreases as the altitude increases, it is evident that the power reduces at increasing altitude. The reason is that the ambient pressure decreases as the altitude increases, the drop in the inlet pressure results to the reduction in the mass flow, hence the reduction in shaft power as the altitude increases.

The result obtained showed that at a particular TET (e.g. 1400K) the shaft power decreases by 17.1% for the clean engine for every 500m elevation while the degraded condition reduced by 24.8% at the same operating condition. The result also affirmed the fact that even without degradation, the shaft power of the engine also reduces at high altitude. However, shaft power increases with increasing TET while engine degradation reduces the shaft power at the various TETs. On the other hand, due to the increasing overall pressure ratio, the combustor pressure does not experience much decrease like the mass flow. Therefore, the TET increases a little to maintain constant non-dimensional mass flow. It can also be concluded here that the decrease in the shaft power of a fouled engine at a fixed TET and high altitude is higher than the clean engines during operation.

Figure 4-8 represents how the engine thermal efficiency behaves under different altitude and engine degradation. For gas turbine engine, thermal efficiency is a function of the pressure ratio and the temperature ratio, the trend increases as the altitude increases, the reason is that at a higher altitude, the temperature decreases which result to higher efficiency as the air gets colder. The result agreed with Pilidis [175] that says; gas turbine performs better in cold regions when compared with a hotter region. Meanwhile, the thermal efficiency increases as the TET increases. This happened because the temperature ratio and the pressure ratio increases further at an elevated TET. Although the thermal efficiency increases at increasing altitude, especially with increasing TET, yet, it reduces in the degraded condition. The reduction in the degraded condition indicates the lack of mass flow and a slight increase in temperature when the engine degrades. Based on this condition, the efficiency reduces when compared with the clean engine.

Figure 4-9 shows the result of the fuel flow when operating under varying altitude. The fuel flow reduces as the altitude increases. The result is expected because of the reduction in shaft power as a result of the decrease in mass flow of the engine as the altitude increases. The fuel flow increases more at higher TET

The result obtained also showed that the fuel flow of the clean engine reduced by 4.1% for every 500m increase in elevation above the ISA condition while the degraded condition reduced by 7.4% at the same operating condition. The result also affirmed the fact that even without degradation, the fuel flow of the engine can reduce

at a high altitude of engine operation. However, the fuel flow increases with increasing TET while engine degradation reduces the fuel flow at the various TETs. It can also be concluded here that the decrease in the fuel flow in a fouled engine at a fixed TET and high altitude is higher than the clean engines during operation.

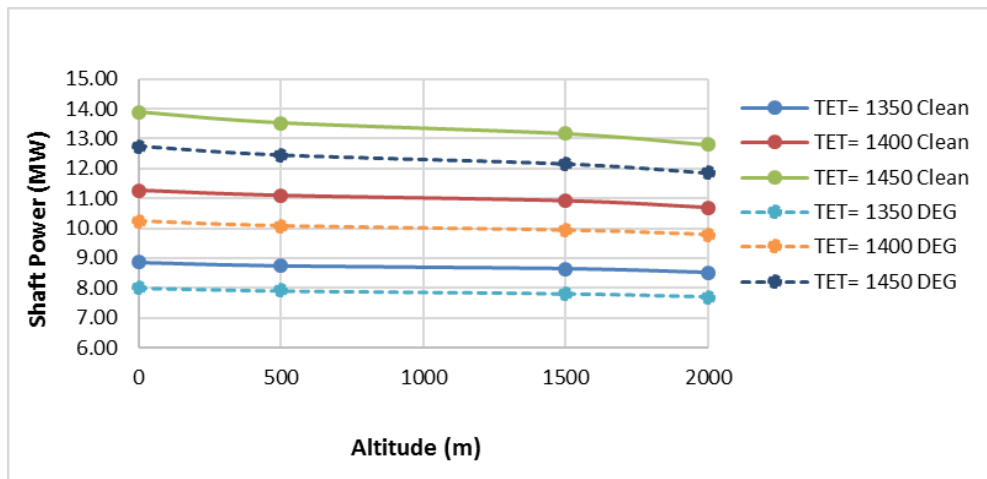


Figure 4-7 Effects of Altitude on shaft power

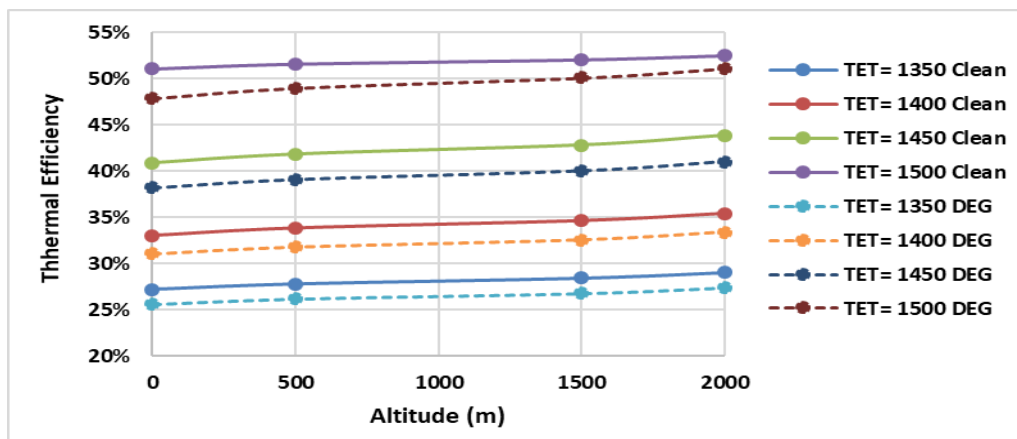


Figure 4-8 Effect of Altitude on thermal efficiency

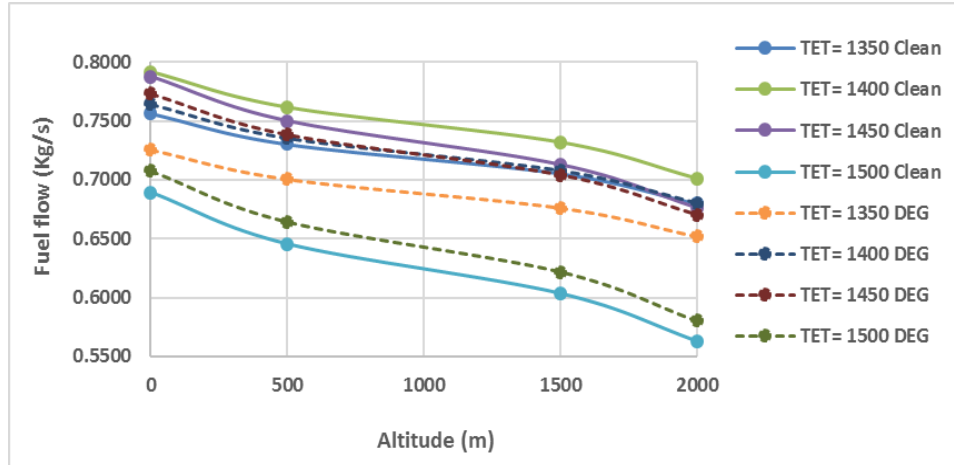


Figure 4-9 Effects of Altitude on fuel flow

4.6.1.2 Effect of Degradation on Engine Component Parameters

In this section, analysing the off-design performance of the model, three conditions are been investigated,

- (i) Degradation of compressor only
- (ii) Degradation of the turbine only
- (iii) Degradation of both compressor and Turbine

In these three conditions, the reduction in both the flow capacity and isentropic efficiency of the components are taken into consideration. Meanwhile, the analysis was investigated at constant TET of 1440K and ISA condition.

In a way to investigate the potential impact and the various ways degradation can affect the operation of GT engines, various engine parameters like, shaft power, mass flow, efficiency, fuel flow and pressure ratio were investigated to the see the extent at which degradation can cause these parameters deviates from there DP. When Gas turbine engine is degraded, it is evident that these parameters will be affected.

In all the three cases outlined above, 1-5% reduction in flow capacity and isentropic efficiency was implanted on the engine model in each case. Figure 4-10 represents the fouling of the compressor only. The result shows that for every 1% reduction in the flow capacity and isentropic efficiency, the shaft power reduces by 2.85%, fuel

flow reduces by 0.7%, pressure ratio reduces by 0.3%, efficiency and mass flow reduces by 2.2% and 0.33% respectively when compared with their DP values.

Again, Figure 4-11 represents the degradation of the turbine only, and the result shows that for every 1% reduction in flow capacity and isentropic efficiency, the shaft power reduces by 3.7%, Fuel flow reduces by 0.95%, efficiency and mass flow reduces by 2.75%, and 0.89% respectively. It was observed that when only the turbine was degraded, the PR was 0.04 less than when only the compressor was degraded

Accordingly, Figure 4-12 represents the degradation of both compressor and turbine. The result obtained revealed that the shaft power decreases by 6.45%, Fuel flow reduces by 1.7%, PR, Efficiency and mass flow reduces by 0.43%, 4.8% and 1.3% respectively.

It can be concluded here that the degradation of the turbine has more effect on the engine than the compressor. However, when both compressor and turbine degrade, a significant reduction in the performance parameters is felt than when either the compressor or turbine degraded.

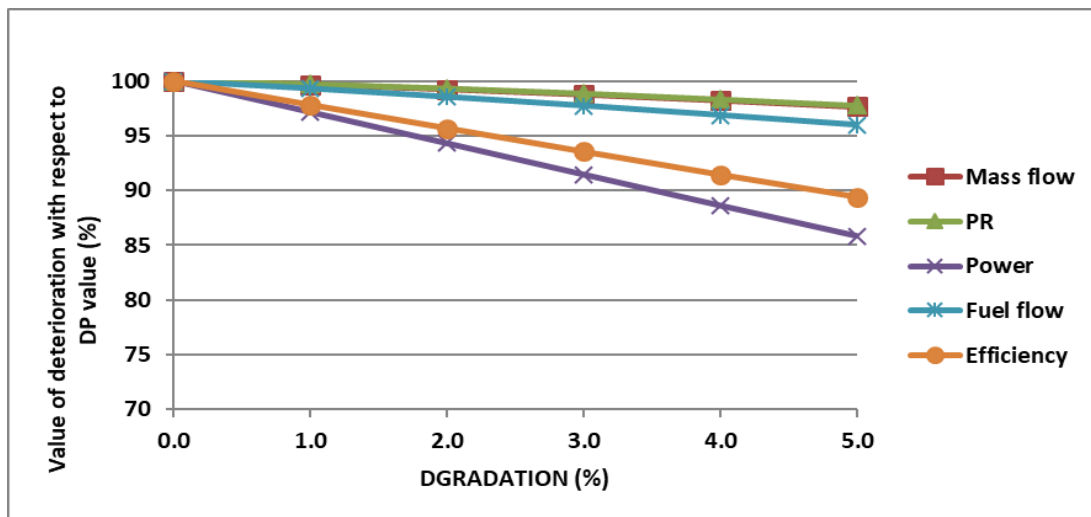


Figure 4-10 Effects of compressor degradation on performance parameters

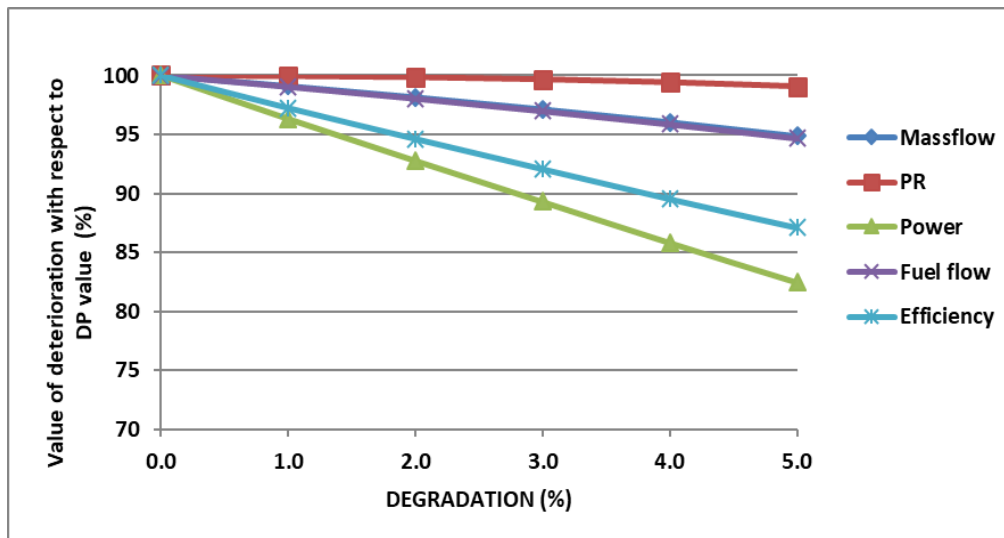


Figure 4-11 Effects of turbine degradation on performance parameters

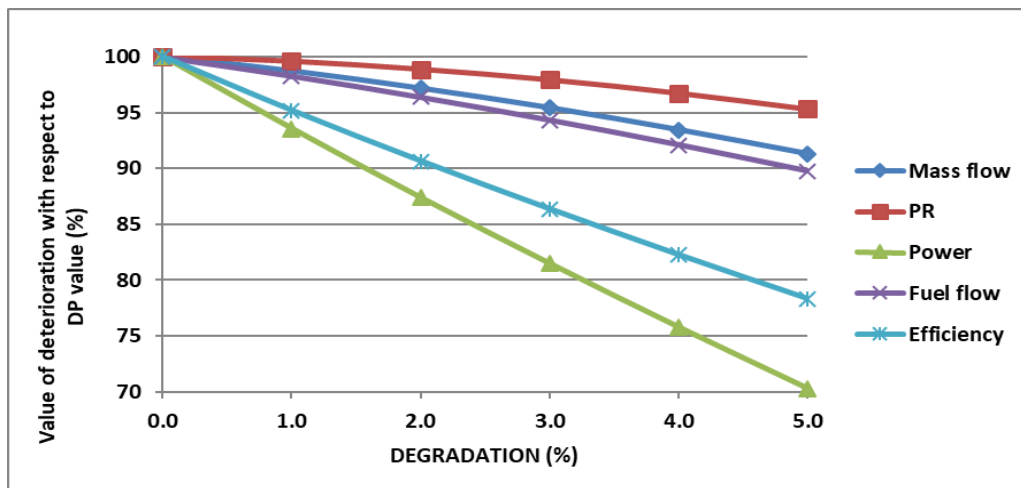


Figure 4-12 Effect of "compressor and turbine" degradation on performance parameters

4.7 IGTB Performance Specification

IGTB was inspired by SGT-500 triple-shaft Industrial gas turbine based on the criteria on GT selection. This engine is a lightweight, rugged machine that produces high efficiency and has been in existence for many years now. The engine was designed for heavy oil operation under harsh conditions. SGT-500 is extensively used in power generation, marine as well as oil and gas industry, it could either be an offshore or onshore environment, floating or fixed. Because of its large capacious combustors and low firing temperature, the engine has the capacity to perform on a wide range

of fuels (gas and heavy fuel oils) particularly gases with a high content of hydrogen sulphide.

Concerning its world-class start-up outstanding operational reliability, the engine is an excellent solution for emergency power application. Its low firing temperature enables the maintenance programme to be impressive. The engine is capable of delivering the required power for the gas compressor. The engine Schematic diagram is represented in

Figure 4-13, while Table 4-3 shows the specified parameters of the modelled engines at ISA condition. Also,

Table 4-4 shows the performance specification and validation for IGTB at DP.

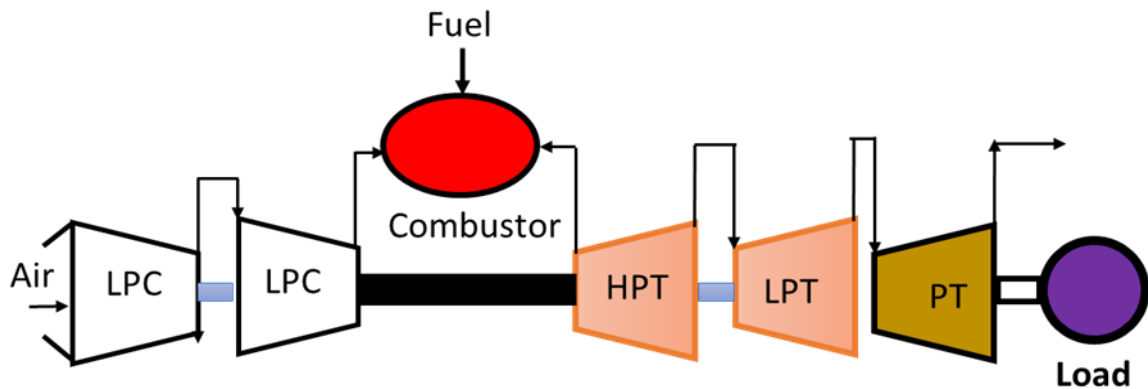


Figure 4-13 Schematic diagram of IGTB 3-spool engine

Table 4-3 Design point parameters of the IGTB model at ISA condition

Parameters	Parameters	Values
1	Inlet mass flow (kg/s)	96.75
2	Pressure ratio	13
3	TET	1300
4	Shaft Power (MW)	19.3
5	Combustor efficiency	0.999
6	Compressor turbine efficiency	0.88

Table 4-4 IGTB Model Validation at DP

No	Parameters	Reference Website	TURBOMATCH Result	% Difference
2	Thermal efficiency (%)	34.2	34.3	0.29
5	Fuel flow (kg/S)	1.1521	1.1345	1.53
	Mass flow (kg/S)	96.73	96.75	0.02
6	TET(K)	-	1300	-

4.7.1 IGTB Compressor Map

Figure 4-14 and Figure 4-15 shows the compressor map for IGTB inspired from SGT 500 engine, indicating the clean and the degraded engine during operation. It is important to note that IGTB is a triple shaft with a free power turbine. With this configuration, it has both low pressure (LP) and high pressure (HP) compressor as indicated in Figure 4-14 and Figure 4-15, respectively.

The results indicate the map for the LP and HP compressor with the running line, surge line and various engine rotational speeds. Meanwhile, a 3% reduction in flow capacity and efficiency was used to degrade the compressor. The key performance parameters that are plotted against each other is the pressure ratio and isentropic efficiency against the corrected mass flow (CMF) of the component showing a range of non-dimensional speeds. (In some cases, it is represented as a fraction of the highest non-dimensional speed, PCN).

From the simulation result, it was obvious that as the compressor speed increases, the mass flow increases accordingly as well as the pressure ratio. As the pressure ratio continues to increase at each of the non-dimensional speed, the engine will experience a geometry change. A point will reach when the mass flow will fall slightly and eventually reach a point where the blade of the compressor cannot produce more pressure ratio without stalling. The compressor will then “surge” as shown in Figure 4-14 and Figure 4-15. The compressor cannot perform its operation at a pressure ratio above this point. Generally, as indicated on the curve, as the non-dimensional speed increases, the curves become steeper and ultimately vertical.

Conclusively, compressor degradation causes a shift in operating point on the characteristic map. It is seen that the corrected mass flow and pressure ratio of the degraded compressor is lower in comparison with the clean engine. Generally and more importantly, from Figure 4-14 and Figure 4-15, at a particular non-dimensional rotational speed and pressure ratio, the corresponding CMF can be deduced. Also, with a given PCN and CMF, the corresponding pressure ratio can be deduced from the map. Having a good knowledge of component map of GT engines will help operators and the industry in general to know the actual behaviour of the engine at various operating conditions taking into account the potential impact of engine degradation on the engine surge line

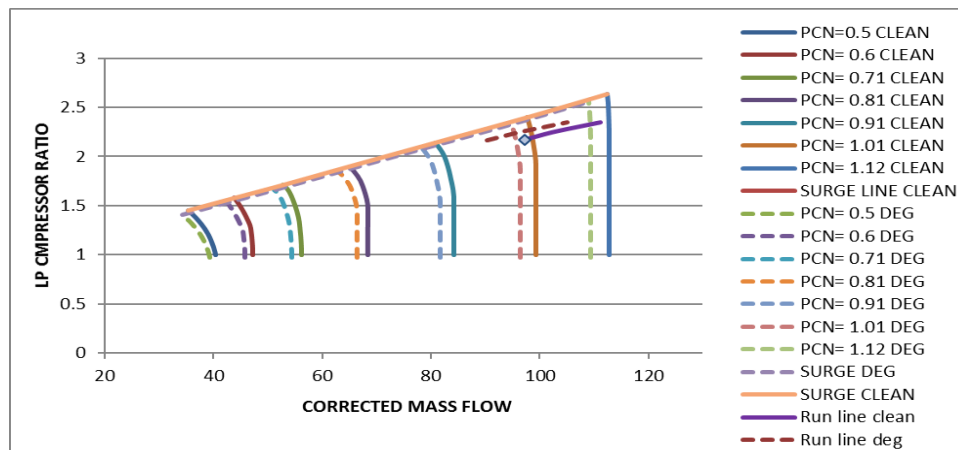


Figure 4-14 IGTB LP Compressor map (PR vs CMF)

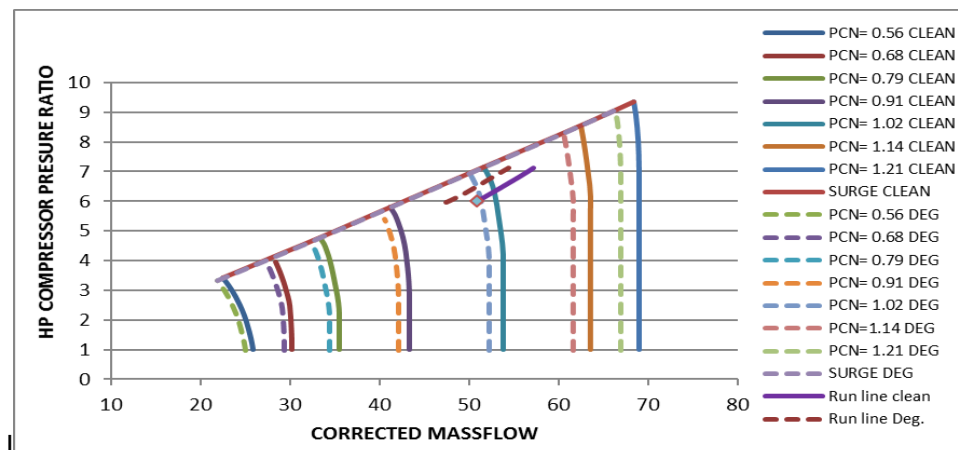


Figure 4-15 IGTB HP Compressor map (PR vs CMF)

4.7.2 The Impact of Degradation on the Deviation of engine Parameters from their DP

Figure 4-16, Figure 4-17 and Figure 4-18 show when the engine parameters by the degradation of "Compressor", "Turbine" and both "Compressor and turbine at constant TET in an ISA condition deviates from their DP. It should be noted here that in all the three cases, 1-5% reduction in flow capacity and isentropic efficiency was implanted on the engine model. Figure 4-16 represents the fouling of the compressor (LPC and HPC). The result shows that for every 1% reduction in the above degradation indices, the shaft power reduces by 3.6%, fuel flow reduces by 2.1%, pressure ratio reduces by 2.25%, efficiency and mass flow reduces on the engine characteristic curve by 1.4% and 2.1% respectively when compared with their DP values.

In addition, Figure 4-17 represents the degradation of the turbine (LPT and HPT) only and the result shows that for every 1% reduction in flow capacity and isentropic efficiency implanted on the model, the shaft power reduces by 4.1 %, Fuel flow reduces by 1.54%, pressure ratio, mass flow and efficiency reduces by 1.6%, 2.18% and 1.9% respectively.

Accordingly, Figure 4-18 represents the degradation of both compressor and turbine. (LPC, HPC, LPT and HPT) The result here indicates that the shaft power decreases by 7.35%, Fuel flow reduces by 3.71%, Pressure ratio, Efficiency and mass flow reduces by 4.2%, 4% and 4.7% respectively.

It can be concluded here that the degradation of the turbine has more effect on the engine than the compressor. However, when both compressor and turbine degrade, a significant reduction in the performance parameters is felt than when either the compressor or turbine degrade.

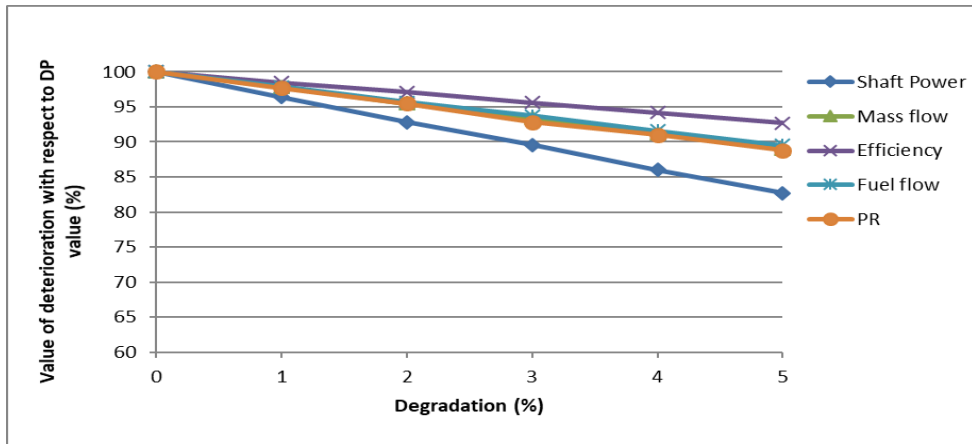


Figure 4-16 Effects of Compressor degradation on engine parameters

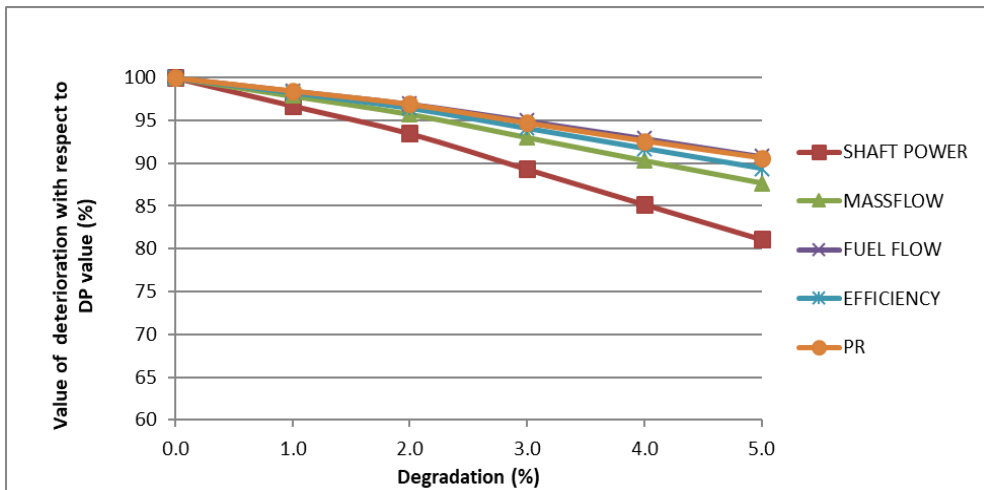


Figure 4-17 Effects of turbine degradation on engine parameters

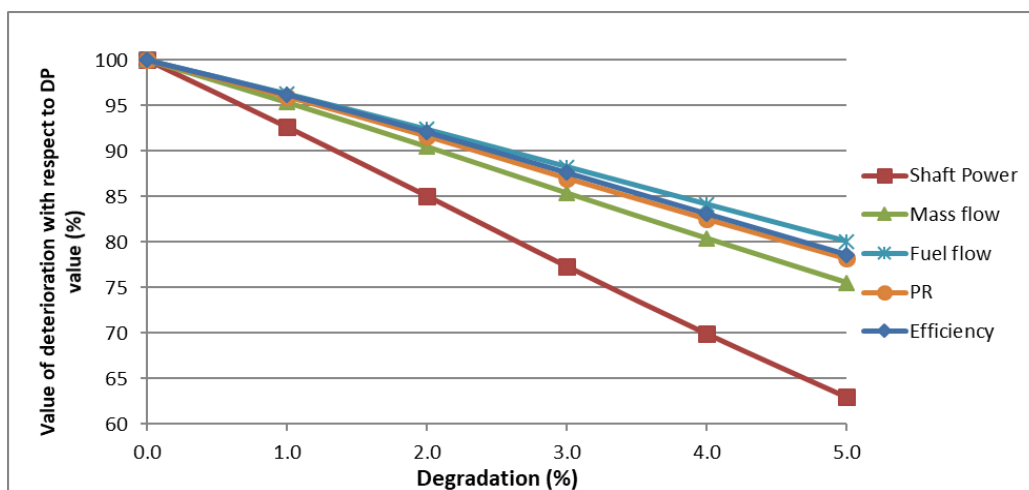


Figure 4-18 Effects of both "compressor and turbine" degradation on engine parameters

4.8 IGTD Performance Specification

IGTD was inspired by GE LM 1600. The General Electric for Land and Marine (LM 1600) is a simple cycle three-shaft high-performance industrial gas turbine engine, a derivative of the F404 aircraft engine series. The three-shaft Industrial gas turbine was selected based on the outlined criteria. This engine is characterised with lightweight and low fuel consumption over a wide-ranging operation. Beside its marine application capability, its usefulness is also seen in the area of power generation, offshore platform power and pipeline compression system. The engine also operates in the steam injected configuration or the simple cycle. IGTD gas turbine consists of a double rotor generator and coupled with a power turbine aerodynamically. The engine has three-stage low-pressure compressor, seven stages high-pressure compressor, a combustor, single-stage high and low-pressure turbines and a power turbine configuration. The gas generator is attached with the power turbine through a transmitting duct while the double concentric gas generator shafts connect the low and high-pressure compressor and turbine rotors.

Table 4-5 shows the specified parameters of the modelled engines at ISA condition, while **Figure 4-19** shows a schematic diagram of IGTD. Also,

Table 4-6 indicates the design point simulation result in comparison with the reference public domain values from the OEM.

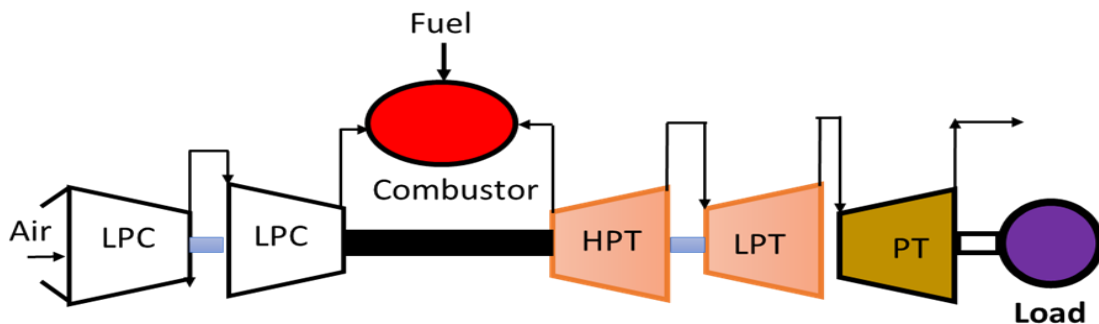


Figure 4-19 Schematic diagram of IGTD 3-shaft

Table 4-5 Design point parameters of IGTD engine model at ISA condition

Parameters	Parameters	Values
1	Inlet mass flow (kg/s)	45.86

2	Pressure ratio (PR)	22
3	Compressor efficiency	0.87
4	Compressor efficiency	0.87
5	Shaft power (MW)	15.2
6	TET (K)	1550
7	Combustor efficiency	99
8	Compressor turbine efficiency	0.88
9	Power turbine efficiency	0.89

Table 4-6 Performance validation of IGTD

No	Parameters	Reference Website	TURBOMATCH Result	% Difference
2	Thermal efficiency	38	38.1	0.263157895
4	Inlet mass flow (Kg/S)	45.86	46	0.30527693
5	Fuel flow	0.8501	0.8751	2.940830491
6	TET	-	1550	-

IGTD LP and HP compressor map are represented in Figure 4-20 and Figure 4-21, respectively. The compressor maps were drawn using the variation in mass flow, PR, and efficiency also with constant rotational speed. The same operational principle as explained for other engines above

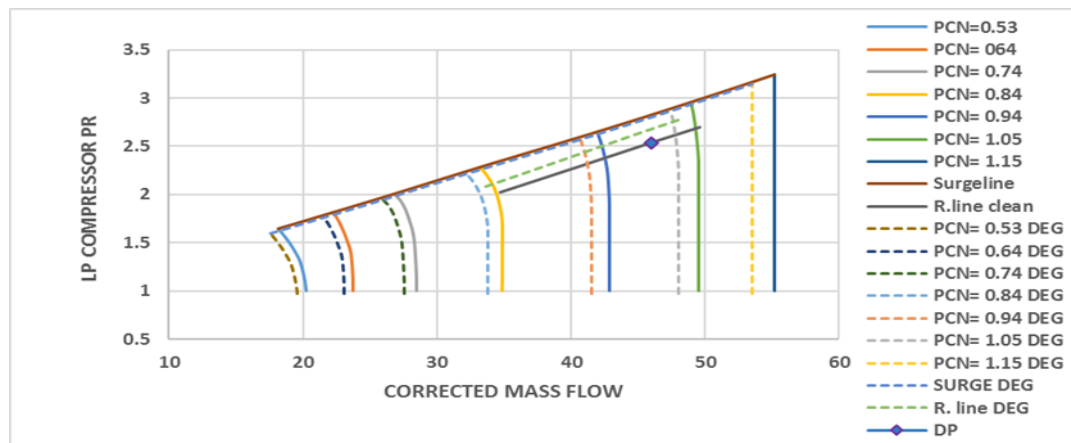


Figure 4-20 IGTD LP compressor map

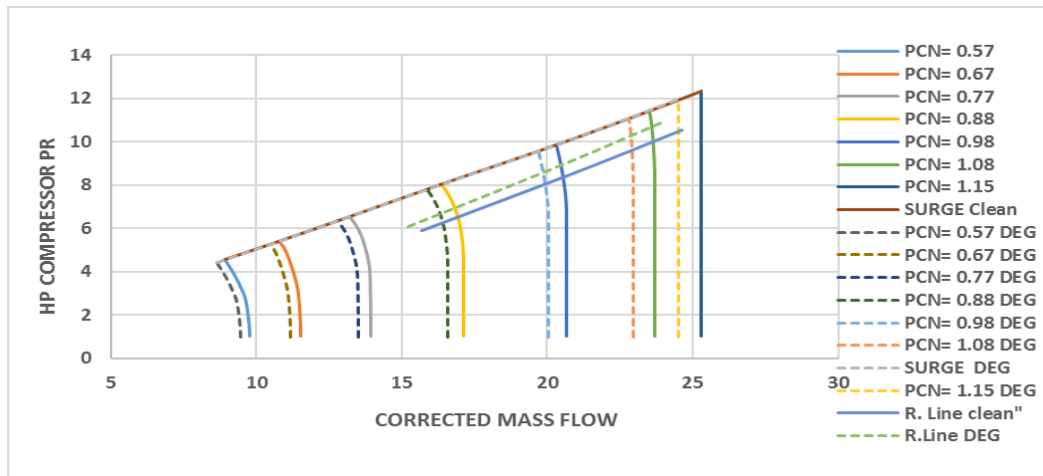


Figure 4-21 IGTD HP Compressor map

4.9 Conclusion

The selection and modelling of the gas turbine have been carried out in comparison with the OEM reference site. Twin shaft IGTA, three-shaft IGTB twin shaft IGTC and three shafts IGTD aero-derivative engines have been analysed and validated for the gas transportation. The thermodynamic models of the selected engines were developed based on the inspired engine DP data available at the public domain. The effects of varying ambient temperature and altitude on the performance of the engines have been analysed and presented.

The simulation of both the clean and degraded engine was carried out using different operating conditions. The analysis helps to investigate the effect of degradation on the overall performance of the engine. Investigation of both the clean and degraded engine on their performance parameters at variable TETs when 3% reduction in flow capacity and isentropic efficiency was implanted into the model has been presented. Again, making the TET as handle at various ambient temperatures with 3% reduction in flow capacity and isentropic efficiency has been analysed and presented in this chapter. Also, the compressor map has been produced in order to understand the operation of the engine. Furthermore, 1-5% reduction in flow capacity and isentropic efficiency was implanted into the model at constant TET and ISA condition. The result of how the performance parameters (shaft power, efficiency, fuel flow, PR.) deviated from their DP at each of the percentage drops is presented. In this preliminary

analysis, three conditions of engine operation were analysed: (i) How much the performance parameters deviated from their DP when only the compressor was degraded (ii) how much deviation occurs when turbine alone was degraded and (iii) How much deviation occurs when both “Compressor and Turbine” was degraded together. It is important to have adequate knowledge on the selection and operation of GT for a specific application. Taking into account the potential impacts of components degradation during operation will help safeguard the industry on the engine limit of operation. Some OD performance result is shown in Appendix D. Meanwhile, in the subsequent chapter, the timed-based ambient temperature profile and the annual degradation in a broader range will be used to analyse the operation of the engines in the area where the natural gas pipeline is located.

5 Influence of Degradation on GT usage and emission evaluation

5.1 Overview

This chapter is divided into two sections. Section one presents the influence of time-based ambient temperature and degradation in determining the number of engine usage in each compressor station at a controlled TET. Section two presents the evaluation of GT emission using some empirical model equation. It also discussed and presented the emission tax based on the influence of the time-based ambient temperature and degradation.

5.2 The Influence of Degradation in determining the number of Gas Turbine usage on the Pipeline application

5.3 Introduction

How long the GT engines can be kept in service safely during the project life amid harsh ambient temperatures and degradation without losing its salvage value has become a growing concern with the compressor station operators. The reason is that GT engines for natural gas transportation are always allowed to run for an extended period due to its economic benefits.,

The major issue with GT turbine operation besides its advantages is usually the effect of the changes in ambient temperature and degradation, particularly in a dry and hot region like Nigeria where the ambient condition has been considered in this research on the overall performance of the engine[96]; [176].

Replacing a deteriorated engine is indeed a huge loss to the industry economically. One solution to solve this problem is to apply a life extension strategy for safely extracting maximum usage of the procured engines on the natural gas pipeline by investing or predicting the actual number of GTs that can serve the entire project life without altering the initial pipeline design. This implies that the initial number of the gas compressor (GC) for a particular compressor station remains unchanged from the initial design while taking the ambient condition and degradation of the prime mover into account.

The predicted number of engines will serve the entire period of the project. Therefore, this section of the research considers how the impacts of degradation and time-based ambient condition will affect the selected engine performance as well as its influence on the number of the GT engines to be used in a compressor's station over the entire life of the project.

Therefore, this section considers how the impacts of timed-based ambient condition and degradation (fouling) will not only affect the engine performance but how it can influence the number of the gas turbine to be used in a compressor's station over the entire life of the project at a controlled turbine entry temperature (TET). This section also addressed the condition where engine overhaul is implemented after three years interval and since the overhaul cannot restore the engine to 100% clean condition, the accumulation of the unrestored degraded condition after each maintenance cycle affected the limit set on the TET of the engine hence the need to acquire more engines which certainly will run on part load to cushion the effect of accumulated unrecovered degradation after routine overhaul. This will enable the engine to meet up the project life, and the set limit on the TET is not exceeded.

It is interesting to note that after the project life, these GT engines can still be divested through sales, unlike the case where the engine will be allowed to deteriorate ultimately, and the operator will go for a replacement.

5.3.1 Implementation of Routine Overhaul on the GT Engine

Three years overhaul interval is applied in this section of the research, and this forms the baseline case of this research. With time as the recovery process continues the engine life may be in danger based on a fixed TET of the engine. Hence the need to acquire additional engine that will run on part load to cushion the effect of accumulated unrecovered degradation due to time of operation. With this approach, the engine can meet up the project life while keeping the TET within its set limit with the mind that time-developing engine operation is characterised by engine performance deterioration

Based on the literature, the implemented overhaul cannot restore the engine to 100% clean condition. Therefore, the accumulation of the unrestored degraded condition

after each maintenance cycle affected the limit set on the turbine entry temperature (TET). This was accounted for in this chapter.

5.4 Methodology

The methodology and the flow of the framework as applied in this section are presented in Figure 5-1. From the flow of the framework, the capacity and the number of GT engines were selected and modelled in TURBOMATCH based on the highest power of the gas compressor in each compressor station for the different season

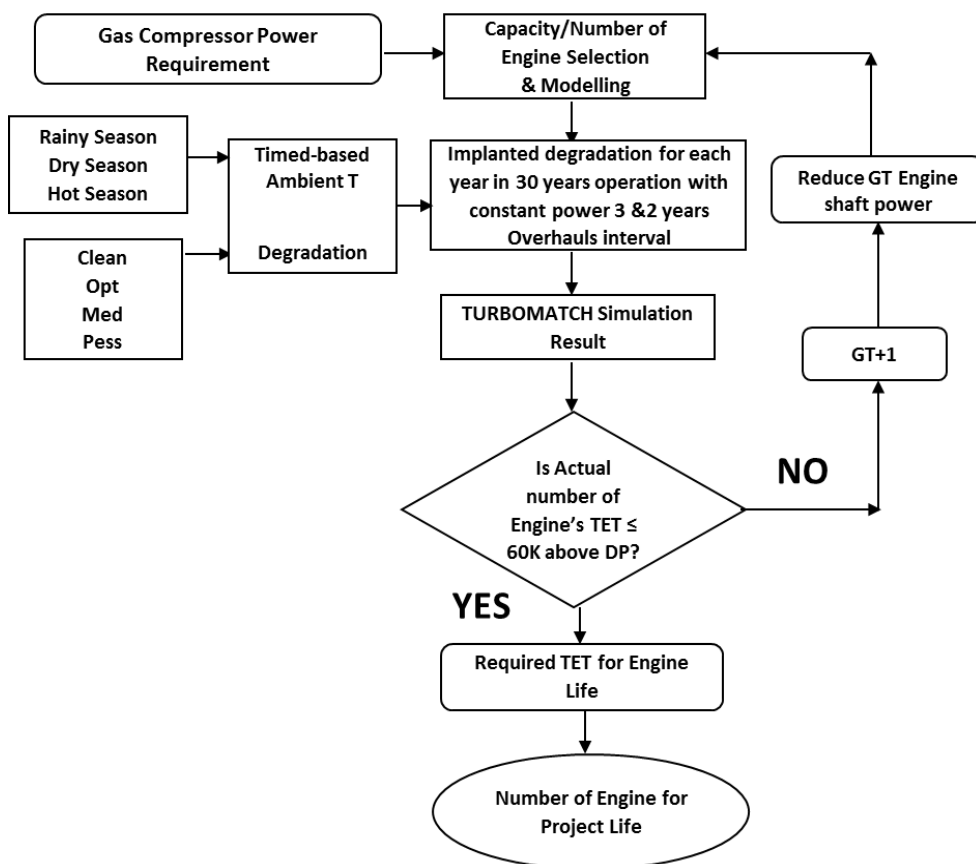


Figure 5-1 Methodology to determine the number of GT at various CS

Three seasons (rainy, dry and hot season) of the region (Nigeria) where the project was investigated was considered in this research. The average ambient temperature profile of the 18 compressor stations of the natural gas pipeline was obtained with respect to the three seasons of the year, as presented in Figure 3-1 to Figure 3-3 in chapter 3. These ambient temperatures represent the average temperature for each season with respect to time of day.

The modelling and performance simulation, including the degradation (fouling) of the mixed fleet's industrial gas turbine, was carried out using TURBOMATCH with shaft power as the handle. The developed code for each season up to 30 years has different levels of degradation implanted for each season of the year. The code also has the implementation of overhaul at three years interval as presented in Appendix E. The code was modelled with respect to three scenarios (optimistic, medium and pessimistic) including the clean condition as represented in Figure 5-1. These scenarios define the level of degradation of the engines, which are discussed in section 5.4.2 of this chapter.

5.4.1 Estimation of Number of GTs to be used for each CS

The engine is expected to run on higher TET due to high ambient temperature and degradation with fixed shaft power. The effect of higher TET as the engine would continue to run on constant power was a concern in this research. To help the engine work to its efficient capability and to ensure that the life of the project is maintained, increasing the number of the gas turbine in each station due to the impact of the ambient condition and degradation with the existing pipeline design become very imperative with a fixed GT shaft power. The engine TET was set not to exceed 60K above the DP value during operation. The TET was so fixed such that irrespective of the adverse effect of engine degradation in addition with the harsh ambient temperature, at least one-eighth of the entire life of the engine can still be available for the engine based on the study of Gad-Briggs et al. [170]

Figure 5-2 illustrates the modelled engine at constant shaft power. The TET increases as degradation increases with time of engine operation. The choice of fixing the 60K above the DP was reasonable but was chosen arbitrarily. Besides the reason stated above, putting the TET limit below this value may not enable the engine to work for the required period, taking into account the worst degraded condition (pessimistic scenario) of the engine operation. However, a different value of TET constraint can easily be accommodated by the model considering the condition of the engine operation.

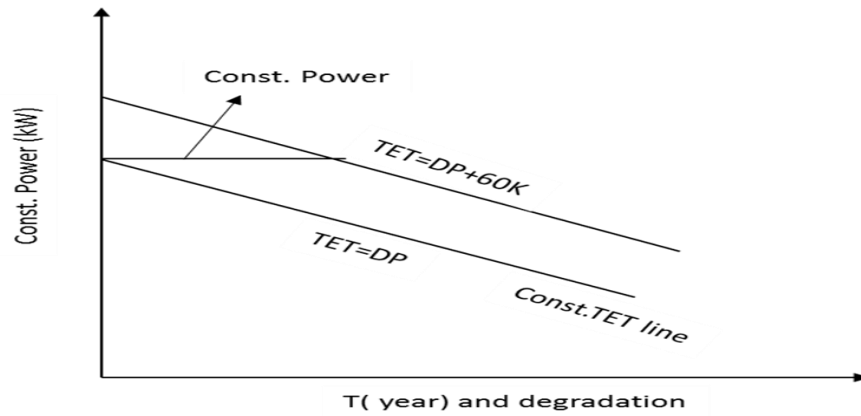


Figure 5-2 Constant GT power with increasing degradation with time

From the illustration in Figure 5-1, If the engine operates for a particular time, and the 60k of the TET above the DP is exceeded, it means that another engine of the same fleet needs to be added. The additional engine will enable all the engine fleet to operate on a lower shaft power. Operating the engine on lower power settings will help to reduce the TET, thereby maintaining the engine within the specified TET. Equation (5-1) gives the developed formula that can evaluate the actual power of the GT engine during the successive engine addition and the corresponding number of the engine that can conveniently serve the gas compressor power requirement for each CS throughout the project life. It should be noted in this equation that the required GT power for operation is inversely proportional to the number of GT per station. This implies that as the number of GT increases, the shaft power of the individual GT reduces and the summation of the reduced shaft power of the GT based on the actual number in each compressor station will serve the power requirement of the GC. Meanwhile, the original pipeline design remains the same even at the case of additional GT engine.

$$GT_p = \frac{NGC * GC_{hp}}{NGT} \quad (5-1)$$

Where,

GT_p = Required constant GT Power operational for each CS

NGC = Number of Gas Compressor per Station

GC_{hp} = Highest Power of the GC based on season and time of the day in each CS

NGT = Number of GT per station

Having achieved the power and the number of GT engine to be used for the CS, at a particular year with the continual operation, the engine TET increases above the limit due to elevated temperature and degradation, thus, resulting to the addition of an extra engine with the achievement of resultant new engine power. This new power of the engine will be used to produce another engine model for a new simulation. The TET may still go higher after some time of operation due to the effect of elevated temperature and the unrecovered degradation at constant power operation until a stable point where the operational TET cannot be 60K above the DP TET.

This engine addition becomes necessary as it enhances continues operation and serves as a palliative measure to improve the life and performance of the GT for uninterrupted gas supply to the consumer and also to preserve the salvage value of the engines amid harsh ambient temperature and degradation.

For instance, in station 2, the highest power of the GC is 12,606kW, and all the compressor stations were assumed to have 4 GT per station on the bases of parallel connection with each GT engine having a DP power and TET of 15,200kw and 1550K respectively. Running four (4) GT connected to the 4GC for the hot season with the implanted degradation as represented in Equation (5-2), the TET limit of 60K was exceeded when the operational TET reached up to 1876⁰k when compared with the DP TET. It therefore, means that additional engine will be needed as shown in Equation (5-3).

$$\text{Power of GT to be used } (GT_p) = \frac{4 \times 12606}{4} = \frac{50424}{4} = 12.606 \text{ MW} \quad (5-2)$$

$$\text{Power of GT to be used} = \frac{GCP}{NGT} = \frac{50424}{5} = 10.085 \text{ MW} \quad (5-3)$$

Therefore, 10.085 MW becomes the new power requirement that five gas turbine will be operating on for the very compressor station. If the engine continues operation and at a particular year the engine TET increases above the limit, an additional engine will be added, and the new power of the engine will be achieved. This new power of the engine will be used to produce another engine model for a new simulation. With this procedure, the stability of TET can be reached until the engine is overhauled where the engine will be assumed almost clean.

5.4.2 Engine Degraded Scenarios under Investigation

It is already established that the GT engine degrades with time. These degradations are mostly intensified with high ambient temperature. Different seasons of the year produces various ambient temperatures that differ from one another. The temperatures of the different seasons as already shown in Figure 3-1 to Figure 3-3 and different levels of degradation are taken into consideration in this research. These are considered not just for the performance investigation of the engines only but also to investigate the impacts of engine degradation on emission generation as well as the economic investigation of the combined effects of time-based ambient temperature and degradation on the economics of natural gas pipeline investment.

There are four scenarios to be examined in each season of the year in this research. These scenarios define the level of engine degradation besides the clean engine condition. The different levels of degradation in each season of the year are referred to as optimistic (opt), medium (med) and pessimistic (pess) in comparison with the clean condition. These are explained below

Clean Scenario

The clean scenario of this research is the condition where the engine was assumed to work throughout the project life without degradation. This condition assumes that both the fuel consumption and maintenance cost of the engine is equal in each season of the year and throughout the operation time. Only the effect of ambient temperature was felt in the analysis. Many researchers based their economic investigation on this scenario.[18]; [152].

Optimistic Scenario

These were the lower range of degradation implanted on the engine model in each season of the year when compared with other degraded scenarios. This implanted degradation covers the entire project life. As earlier explained, these scenarios take into account the various seasons of the year. The assumption here was that the engine performance, especially on the fuel consumptions and maintenance, are not the same in each season of each year. It varies from one year to another. This phenomenon was applied to all degraded scenarios.

Medium Scenario

These are also a range of degradation that is higher than the optimistic scenario implanted on the engine model. These ranges were implanted in various seasons of the year throughout the project life.

Pessimistic Scenario

These were the highest values of degradation (worst case) implanted on the engine model in each season of the year compared to all other scenarios. In all the scenarios, their impacts on both the technical and economic investigation of the pipeline project were carried out.

5.4.3 Consideration and Summary of Implanted degradation in each Season

Three degraded scenarios were implanted on the engines model, and three of the degraded scenarios were examined for each season (winter, dry and hot season) of the year for thirty years. Since the project life was thirty years, a routine overhaul was considered after three years interval as represented in Figure 5-3 to Figure 5-5. Meanwhile, the various years of engine operation have different values of degradation with respect to industrial practice and literature for thirty years.

Figure 5-3 to Figure 5-5 showed the percentage values of degradation implanted for each scenario for the different seasons of the entire project life. In the winter season, optimistic ranges from 0.25% to 2.65%, medium ranges from 1.25% to 3.65% and Pessimistic ranges from 2.25% to 5.01%. Also, in the dry season, the implanted degradation for optimistic ranges from 0.5% to 2.9%, Medium ranges from 1.5% to 3.9% while Pessimistic ranges from 2.5% to 5.26%. Finally, in the hot season, optimistic ranges between 0.75% and 3.15%, the medium is between 1.75% and 4.15% while pessimistic ranges between 2.75% and 5.51%. An interval of 0.75% between each year of the season before and after overhaul was assumed until thirty years of project life. Also, an interval of 1% between each scenario per season was assumed. These values were used to reduce the flow capacity and isentropic efficiency of both the compressors and the turbines. This assumption was made based on literature and industrial practices. Meanwhile, the values of the yearly implanted degradation are shown in the appendix E. The result from these implanted

degradation based on the scenarios under investigation will form the baseline data for the pipeline economic investigation.

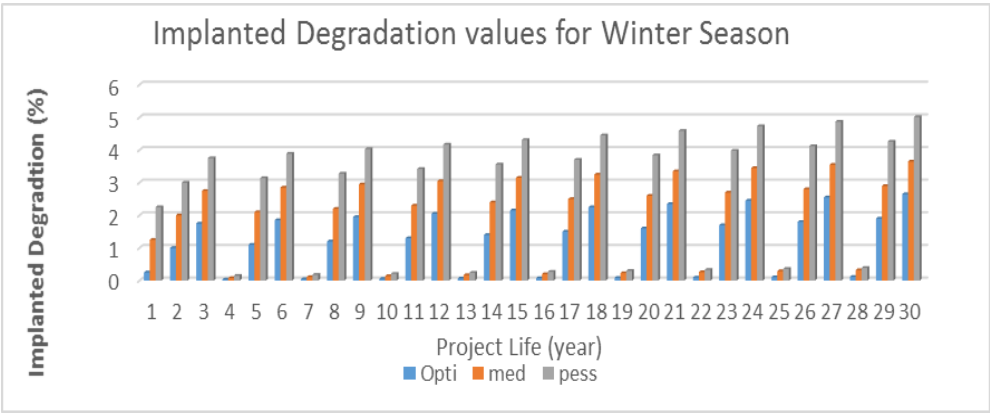


Figure 5-3 Implanted level of degradation for the winter season

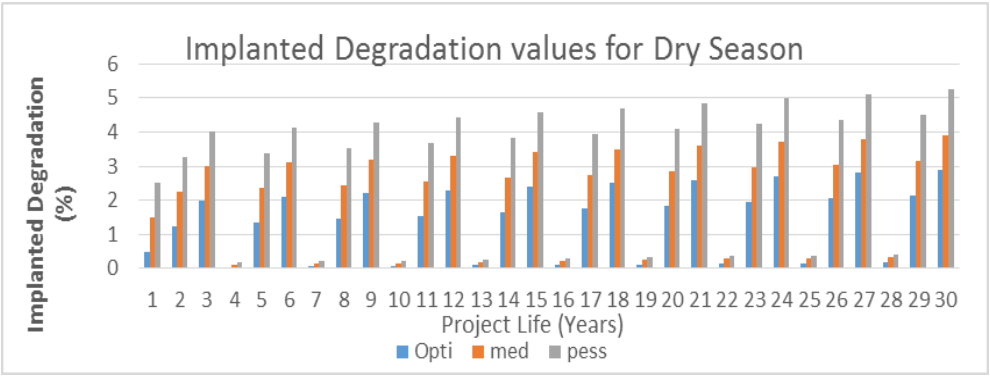


Figure 5-4 Implanted level of degradation for dry season

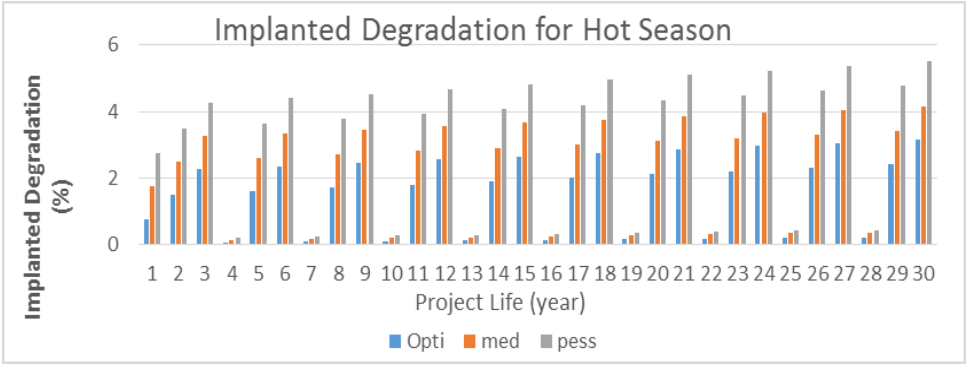


Figure 5-5 Implanted level of degradation for the hot season

Engine overhaul was considered for every three years to maintain the engine for project life. Considering the implanted overhaul in Figure 5-3 to Figure 5-5, as the engine continues operation after each segment of an overhaul, the unrecovered

degradation increases as the year increases also, showing that there are unrecovered degradations which increases with additional maintenance cycle. The implanted overhaul with the trend of unrecovered degradation is showing in year 4, 7, 10, 13, 16, 19, 22, 25 and 28. It also affects other years. This agrees with Beal [47], which stated that "only part out of the total degradation is reinstated in the maintenance process resulting in an engine returned to service with a reduced level of performance". However, the percentage of unrecovered degradation was not mentioned. In this research, unrecovered degradation after each maintenance cycle (overhaul) for the winter season was assumed to increase between 0.035% to 0.39%. The dry season was between 0.05% to 0.415%, while the hot season increases between 0.053% to 0.433%. The percentage of unrecovered degradation for each season stated above covers for all the scenarios under investigation. As the engine continues operation after each segment of an overhaul, these unrecovered degradation increases with additional maintenance cycles until all the assumed unrecovered values are utilised. It was also assumed that each season has an equal length of time.

The assumption for the optimistic, medium and pessimistic was based on literature and industrial practices as seen in gas turbine handbook[177], Naeem et.al [86], Rainer et al.,[178], and Meher-Homji et.al.,[179]

5.4.4 Gas turbine placement to each Compressor station

Table 5-1 Showed the various engines distributed to all the eighteen compressor stations based on the GC power requirement as represented in each season of the year after the simulation of the engine at the clean condition. Each of the GT engines has its different power settings. All their respective shaft power and the number of the engines in each station for the various seasons of the year are clean conditions only, as shown on the table.

The variation in the different settings of the GT is due to the different power requirement of the GC because of the difference in ambient temperature. From the table, IGTA1 and IGTA2 are the same two-shaft engine that was attached to compressor station 1 and 18, respectively. IGTB1 and IGTB2 are the same three-shaft engine that was attached to compressor station 2 and 14. Also, IGTC1, IGTC2, IGTC3, IGTC4, IGTC5, IGTC6, IGTC7, IGTC8, and IGTC9 are the same three-shaft

engines attached to station 3, 4, 5, 6, 7, 11, 12, 13 and 15 respectively. Finally, IGTD1, IGTD2, IGTD3, IGTD4, and IGTD5 are also the same three-shaft engines attached to station 8, 9, 10, 16 and 17 respectively.

Table 5-1 Gas turbine engines attached to each compressor station

Winter Season at clean condition					Dry Season at clean condition				Hot Season at clean condition			
No. of CS	GT Used	Power (KW)	No. of GT Used	GC Power(kw)	GT Used	Power (KW)	No. of GT Used	GC Power(kw)	GT Used	Power (KW)	No. of GT Used	GC Power(kw)
1	IGTA1	13400	5	8805.14	IGTA1	13400	5	8815.18	IGTA1	13400	5	9323.00
2	IGTB1	15200	5	11725.65	IGTB1	15200	5	11854.25	IGTB1	15200	5	12606.00
3	IGTC1	17000	5	13588.18	IGTC1	17000	5	13803.78	IGTC1	17000	5	14802.00
4	IGTC2	17000	5	13142.31	IGTC2	17000	5	13287.60	IGTC2	17000	5	14411.00
5	IGTC3	17000	5	13175.07	IGTC3	17000	5	13295.44	IGTC3	17000	5	14461.00
6	IGTC4	17000	5	13178.65	IGTC4	17000	5	13296.31	IGTC4	17000	5	14467.00
7	IGTC5	17000	5	12512.06	IGTC5	17000	5	13375.39	IGTC5	17000	5	13966.00
8	IGTD1	19300	5	14875.56	IGTD1	19300	4	14995.68	IGTD1	19300	5	15968.00
9	IGTD2	19300	4	11941.94	IGTD2	19300	4	12772.68	IGTD2	19300	5	14897.00
10	IGTD3	19300	4	11218.48	IGTD3	19300	4	12603.03	IGTD3	19300	5	15046.00
11	IGTC6	17000	5	11458.57	IGTC6	17000	5	11910.77	IGTC6	17000	5	13311.00
12	IGTC7	17000	5	12179.37	IGTC7	17000	5	12319.05	IGTC7	17000	5	13412.00
13	IGTC8	17000	4	11825.03	IGTC8	17000	5	13602.86	IGTC8	17000	5	14763.00
14	IGTB2	15200	5	9806.22	IGTB2	15200	5	10814.47	IGTB2	15200	5	11252.00
15	IGTC9	17000	5	12520.10	IGTC9	17000	4	12948.92	IGTC9	17000	5	13739.00
16	IGTD4	19300	4	12024.54	IGTD4	19300	4	12944.06	IGTD4	19300	5	15118.00
17	IGTD5	19300	4	11314.22	IGTD5	19300	4	12467.66	IGTD5	19300	5	14975.00
18	IGTA2	13400	5	8727.70	IGTA2	13400	5	8698.49	IGTA2	13400	5	10292.00

It is imperative to state that the matching mode of the GT engine is such that, at the same time interval, the engine TET for a particular scenario is constant to give the power requirement of the GC. However, The TET for a scenario with less number of the engine will be higher than the scenario with a higher number of the engine. This is because, as the number of the engines increases, the shaft power of the engine for a particular station decreases and the combination of this engine power will serve the power requirement of the gas compressor.

It is imperative to state that the matching mode of the GT engine is such that, at the same time interval, the engine TET for a particular scenario is constant to give the power requirement of the GC. However, The TET for a scenario with less number of the engine will be higher than the scenario with a higher number of the engine. This is because, as the number of the engines increases, the shaft power of the engine for a particular station decreases and the combination of this engine power will serve the power requirement of the gas compressor.

5.5 Result and Discussion

5.5.1 Turbine Entry Temperature (TET) Profile Map (Conjugate Map)

Since the creep life and risk issue of the engine is hinged on the Turbine Entry Temperature (TET), the TET profile conjugate map that matched with the number of gas turbine engine used for the project life was produced. The TET conjugate map is the time-based TET profile result from the different number of engines after simulation. The results helped to ascertain the number of GT to be used for each season of the year and with respect to each scenario under consideration. The map was produced such that, if the engine is running in each season of the year with respect to the time of day and the set limit of 60K TET above DP is exceeded, another engine of the same fleet but with lower power setting will be added. The previous and the additional engines will be allowed to run on part load thereby making the engines to run on a lower TET. The new TET that the engine is operating on will be lower than the previous one. If it continues again in the subsequent years after overhaul, and the fixed operational TET is exceeded, another engine of the same fleet with lower power setting will be added. This exercise will continue until a stable TET value for a particular number of the engine is maintained for the project life. The achieved number of engines that satisfies the condition of operation will serve the entire life of the project. So, the variation of TETs as a result of the different number of engines used for the entire period was mapped together (TET conjugate map) to show the different number of engines used for each year with respect to time of day. So the result was obtained with different power settings based on the fixed operating conditions. This is evidently shown in Figure 5-6 to Figure 5-15.

Figure 5-6 shows the simulation result for IGTD illustrating when the TET limit was exceeded in all the years for using 4GTs throughout the operation for the pessimistic scenario. The engine DP TET was 1550K, and with this value; the operational TET was not supposed to exceed 1610K. However, after the simulation with 4GTs, none of the values of TET can be used for further analysis as almost all the TET exceeds the limits of 60K above the engine DP. The TET reached up to 1878K from the DP value of 1550K. This implies that this engine cannot survive until the end of project life. Hence the need for additional engines. This illustration is made in order to get an insight into how the TET could be high in a degraded condition and also strike a balance on how a conjugate TET map is produced. It is important to state that this evaluation applied to all the engines used in each CS and all engines exhibit the same result.

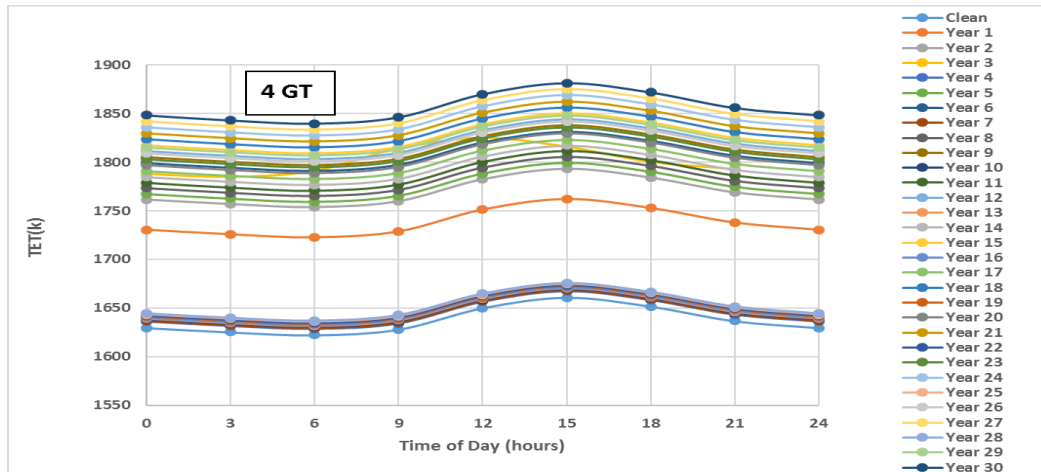


Figure 5-6 TET Profile for the pessimistic scenario without TET limit

5.5.2 Analysis of the Compressor Station TET profile based on Season's of the Year

Only station two TET profile map for all the seasons was analysed and presented in this chapter. The reason is that the methods and results of other compressor stations follow the same pattern. It is important to state that all the three degraded scenarios (opt, med, and pess) including the clean condition was analysed in each season of the years for thirty years of project life. Meanwhile, TET profile map for the various CS, which help in determining the actual number of GT in each CS was evaluated and the results for all stations and their respective seasons are presented in this chapter.

5.5.2.1 TET Profile Map for Rainy Season

From Table 5-1, CS 2 has the engine TET and shaft power DP to be 1550k and 15.2MW, respectively. This implies that the operational TET cannot exceed 1610K based on the operational constraint as represented in Figure 5-1. Again, every change in shaft power as a result of an increase or decrease in the number of engines result to TET increase or decrease, and this power setting must be fixed in the modelling or re-modelling process. This will enhance accurate conjugate map and in turn, produce the required number of engines in a particular station after satisfying the condition. The map shows the various TETs of different engine power settings with respect to time of day.

Figure 5-7 to Figure 5-15 showed the TET profile map from 5, 6 and 7GTs used for both the optimistic, medium and pessimistic scenarios with the clean condition in the rainy season. Figure 5-7 shows the time-based TET profile map for the optimistic scenario in the rainy season. The result showed that from year 1 to year 30, five GTs were used for the compression station since the TET limit was exceeded when 4GTs were used. For this season, an overhaul was implemented after three years periodically for the thirty years of project life as shown in year 4, 7, 10, 13, 16, 19, 22, 25, and 28. So it was observed here that as the year increases, the TET increases also, but the TET limit was not exceeded. The curve also indicated that at fixed power, as degradation increases with time, there is a corresponding increase in TET.

Figure 5-8 represents the Timed-based TET profile map for the medium scenario in the rainy season. The result showed that with respect to time of day, five GTs were used from year 1 to year 29 while year 30 used six GTs as the TET was exceeded. As it is seen on the curve, the TET for year 30 dropped because of the use of six GT. Again, Figure 5-9 shows the TET profile conjugate map for the pessimistic scenario in the rainy season. It was observed here that year 1,2, 4, 5, 7, 8, 10, 11, 13, 14, 16, 19, 22, 25 and 28 used five GTs while the rest of the years used 6 GTs for the project life. It should be noted that it was based on the fact that the operational constraint was exceeded at a particular period of the season (Time of the day) that gave allowance to the use of additional GTs. The arrows in Figure 5-7 and Figure 5-8 indicates reduced and increasing TET due to increasing and reducing numbers of

GT respectively, including the effect of engine overhaul. This applies to all the TET profile map

Summarily, for fixed power operation, within an ambient temperature of 17.45°C to 19.95°C, and with 2.65% degradation (reduction in flow capacity and isentropic efficiency of the turbomachinery), the result shows that the TET increases by 6.04% when compared with the clean condition. This highlights the need for performance augmentation to improve the life of the engine by reducing the TET and consequently, fuel flow.

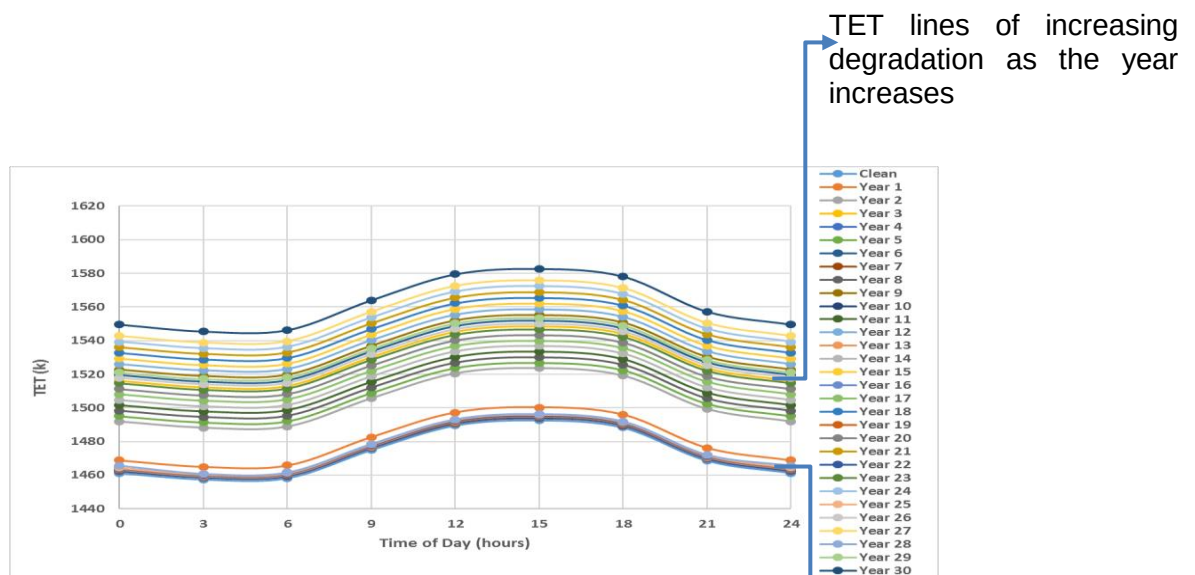


Figure 5-7 TET Profile map for optimistic scenario (Rainy season)

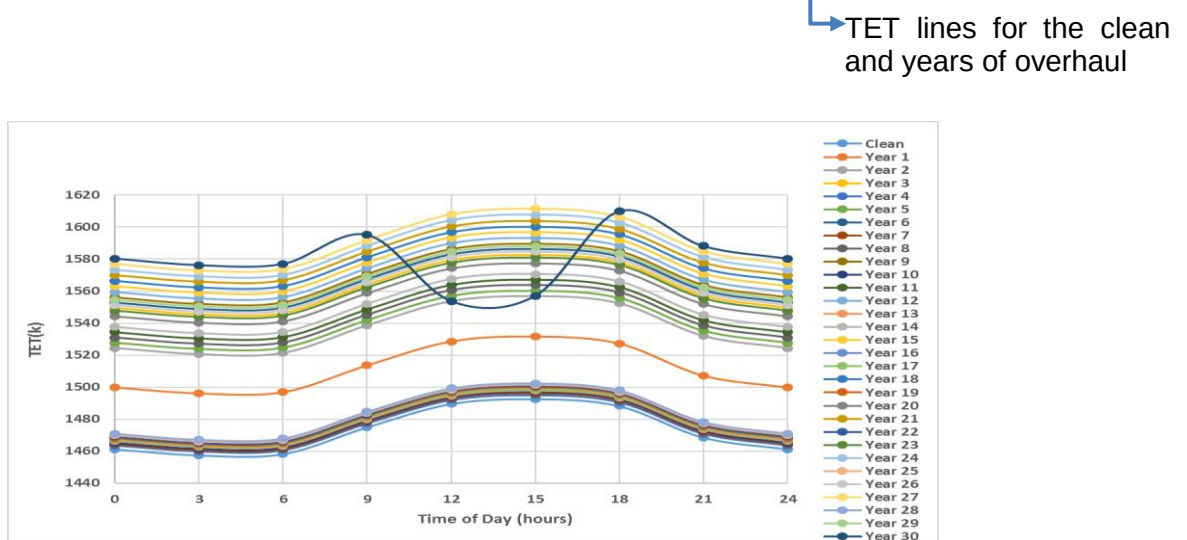


Figure 5-8 TET Profile map for medium scenario (Rainy season)

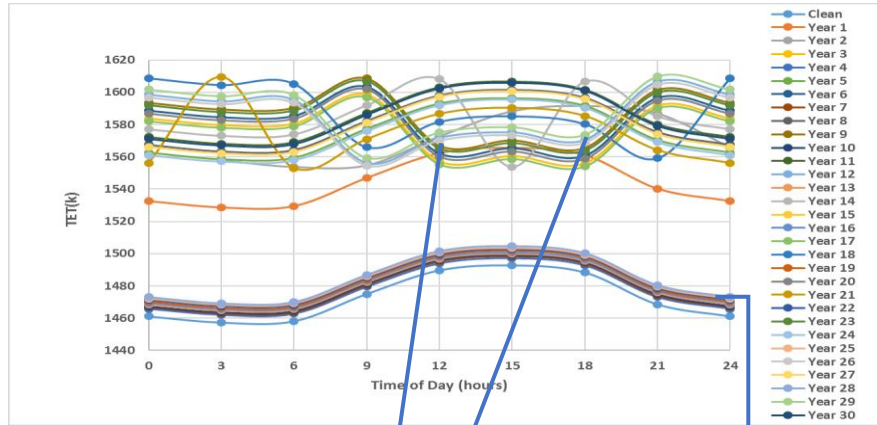


Figure 5-9 TET Profile map for pessimistic scenario (Rainy Season)

Reduced TET due to part-load operation as a result of increasing number of GT

TET lines for the clean and years of overhaul

It was also shown that overhaul was implemented in year 4, 7, 10, 13, 16, 19, 22, 25, and 28 for the scenarios under consideration. This was applied to all the scenarios, as explained above.

5.5.2.2 TET Profile Map for Dry Season

Figure 5-10 showed the timed-based TET profile for an optimistic scenario in the dry season. The result showed that 5 GTs were used from year 1 to year 30. Overhaul implementation was done in year 4, 7, 10, 13, 16, 19, 22, 25, and 28. The result also indicated that at fixed power, as the engine degradation increases with time, there was a corresponding increase in TET also.

Figure 5-11 shows a time-based TET profile map for the medium scenario in the dry season. The result showed that at year 18, 21, 24, 27 and 30, six GTs were used while the rest years used five GTs as shown on the curve. Again, an overhaul was implemented in year 4, 7, 10, 13, 16, 19, 22, 25, and 28. So, the TET drop on the curve indicates where the power settings were low as a result of additional engines. This is because, as the number of GT increases, the TET reduces due to part-load

operation in accordance with equation (5-1). This is applicable to all the scenarios where the TET limit exceeded the condition of operation. Figure 5-12 also shows the TET profile for the pessimistic scenario in the dry season. The result indicated that year 3, 6, 8, 9, 11, 12, 14, 15, 17, 18, 20, 21, 23, 24, 26, 27, 29 and 30 used TET values from six GTs while year 27 and 30 used seven GTs as shown on the curve. The rest year used five GTs. Implementation of the overhaul was also done in year 4, 7, 10, 13, 16, 19, 22, 25, and 28.

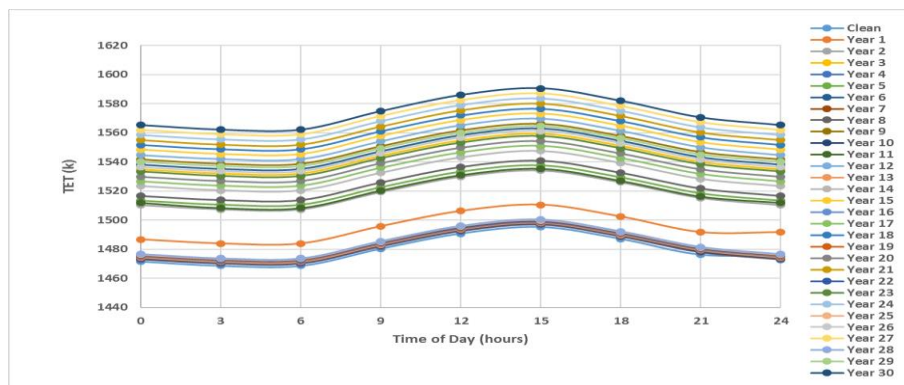


Figure 5-10 TET Profile map for optimistic scenario (dry season)

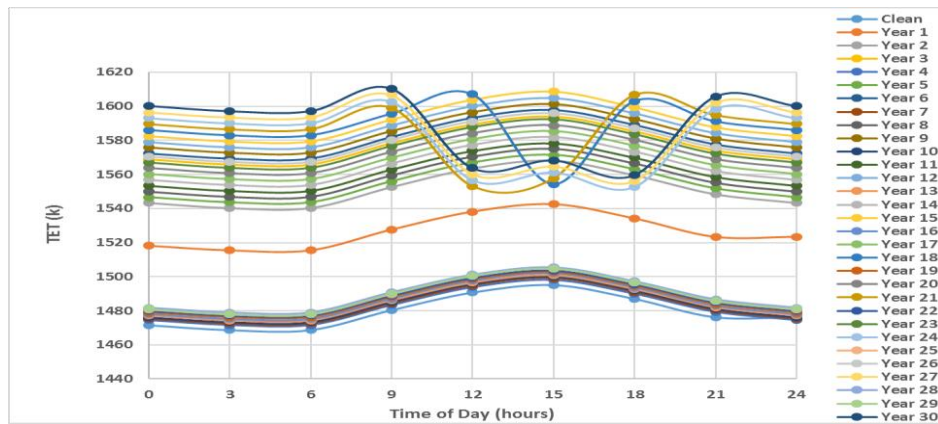


Figure 5-11 TET Profile map for medium scenario (Dry season)

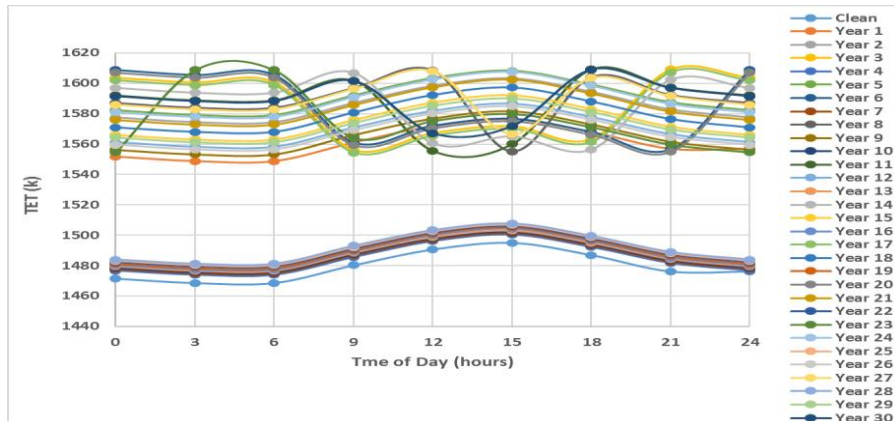


Figure 5-12 TET Profile map for pessimistic scenario (dry season)

5.5.2.3 TET Profile Map for Hot Season

This season shows the highest temperature values with a corresponding high degradation. The reason is that the ambient temperature and engine degradation increases from rainy to the hot season.

Figure 5-13 showed the timed-based TET Profile map for the optimistic scenario in the hot season. The result showed that all the years used the TET values from five GTs except at year 30 that six GT was used. So the engine will be saved to use 6 GTs at year 30. Figure 5-14 shows the time-based TET profile map for the medium scenario in the hot season. The result showed that year 3, 5, 6, 9, 12, 15, 18, 20, 21, 23, 24, 26, 27, 29 and 30 used the TET from six gas turbine. As indicated on the curve, this scenario used a mixture of 5, 6 and 7 GTs throughout the project life. Figure 5-15 shows the TET profile map for pessimistic scenarios in the hot season. Year 2, 3, 5, 6, 8, 9, 11, 12, 14, 15, 17, 18, 19, 21, 23, 24, 27, 29 and 30 uses both mixture of six and seven GT. Particular, year 15, 18, 21, 24, 27, 29 and 30 used seven gas turbine. Year 4, 7, 10, 13, 16, 19, 22, 25, and 28 is the year of engine overhaul where both the clean, optimistic, medium and pessimistic scenarios used 5GTs throughout the project life of this season.

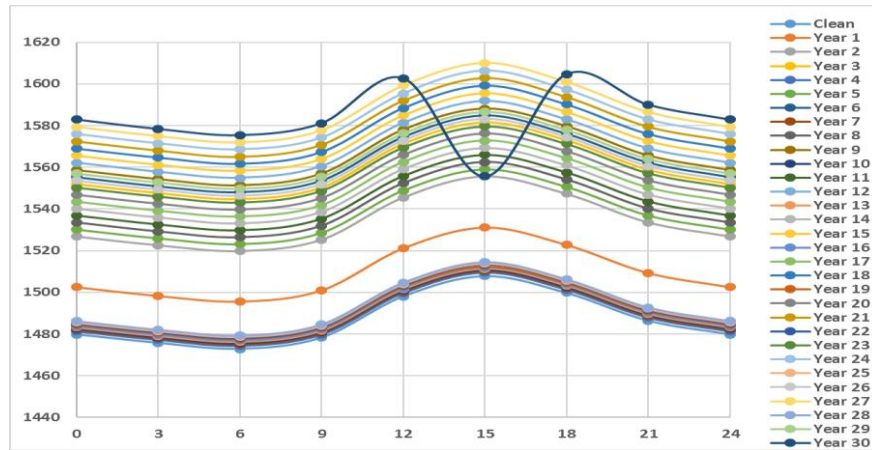


Figure 5-13 TET Profile map for optimistic scenario (Hot season)

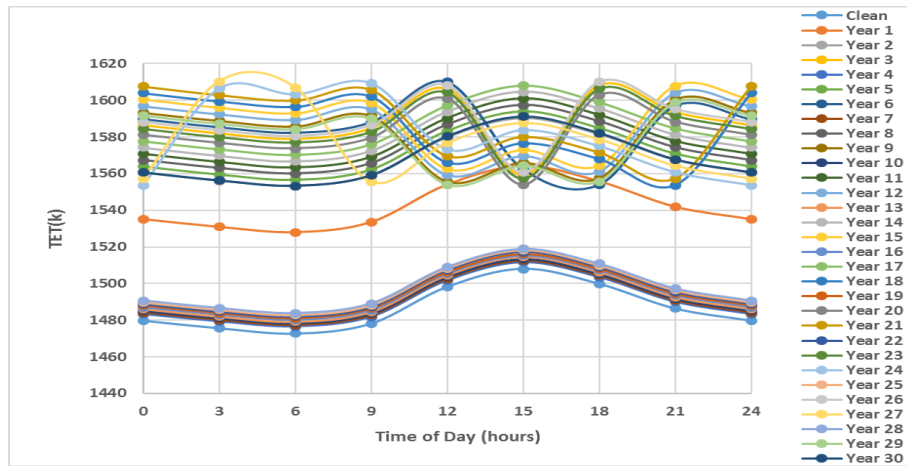


Figure 5-14 TET Profile map for medium scenario (Hot season)

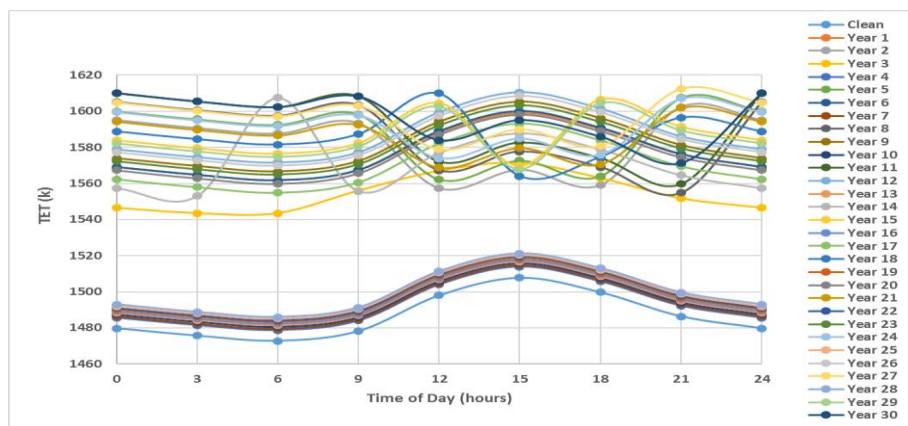


Figure 5-15 TET Profile map for pessimistic scenario (Hot season)

It is interesting to note that this investigation was done in all the eighteen compression stations taken in to account the three seasons of the year with the scenarios under investigation.

It is important to state that the matching mode of the GT with the GC is such that, at the same time interval, the engine TET for a particular scenario is constant to give the power requirement of the GC. However, The TET for a scenario with less number of the engine will be higher than the scenario with a higher number of the engine. The reason is that, as the number of the engines increases, the shaft power of the engine reduces for a particular station and the combination of this engine power will serve the power requirement of the gas compressor.

5.5.3 Number of GT used Per Station with respect to Different Season of the Year

It is now clear from the map that different numbers of GT are been used at different seasons of the year. Also, the three scenarios (opt, medium and pess) including the clean condition are clearly shown for each season for the entire life of the project. Arising from the analysis of the TET profile for each CS, Table 5-2 shows the number of GTs used for all the scenarios in each season of the years throughout the project life. The result shows how degradation influences the number of GT to be used in each season of the year, especially the pessimistic scenarios.

The results showed that clean condition used five GTs in all the seasons and throughout the entire period of the project for the degraded condition, the number of GT used changes in some years. For instances, In the dry season, the med scenario used 6 GTs at year 18, 21, 24, 27 and 30 while the pessimistic scenario used six GTs from year 3, 6, 8, 9, 11, 12, 14, 15, 17,18, 20, 21, 23, 24, 26,27 and 29 while year 30 used 7 GTs. The same analysis was applicable to the hot season as shown in Table 5-2. The same approach was applied in all the 18 CS and for each season of the year.

Table 5-2 Number of GTs used for different season and scenarios for CS-2

	Rainy Season				Dry Season				Hot season			
Year	Clean	Op t	Me d	Pess	Clean	Op t	Me d	Pess	Clean	Op t	Me d	Pess
1	5	5	5	5	5	5	5	5	5	5	5	5
2	5	5	5	5	5	5	5	5	5	5	5	6
3	5	5	5	6	5	5	5	6	5	5	6	6
4	5	5	5	5	5	5	5	5	5	5	5	5
5	5	5	5	5	5	5	5	5	5	5	5	6
6	5	5	5	6	5	5	5	6	5	5	6	6
7	5	5	5	5	5	5	5	5	5	5	5	5
8	5	5	5	5	5	5	5	6	5	5	5	6
9	5	5	5	6	5	5	5	6	5	5	6	6
10	5	5	5	5	5	5	5	5	5	5	5	5
11	5	5	5	5	5	5	5	6	5	5	5	6
12	5	5	5	6	5	5	5	6	5	5	6	6
13	5	5	5	5	5	5	5	5	5	5	5	5
14	5	5	5	6	5	5	5	6	5	5	5	6
15	5	5	5	6	5	5	5	6	5	5	6	7
16	5	5	5	5	5	5	5	5	5	5	5	5
17	5	5	5	6	5	5	5	6	5	5	6	6
18	5	5	5	6	5	5	6	6	5	5	6	7
19	5	5	5	5	5	5	5	5	5	5	5	5
20	5	5	5	6	5	5	5	6	5	5	6	6
21	5	5	5	6	5	5	6	6	5	5	6	7
22	5	5	5	5	5	5	5	5	5	5	5	5
23	5	5	5	6	5	5	5	6	5	5	6	6
24	5	5	5	6	5	5	6	6	5	5	6	7
25	5	5	5	5	5	5	5	5	5	5	5	5
26	5	5	5	6	5	5	5	6	5	5	6	6
27	5	5	5	6	5	5	6	6	5	5	6	7
28	5	5	5	5	5	5	5	5	5	5	5	5
29	5	5	5	6	5	5	5	6	5	5	6	7
30	5	5	6	6	5	5	6	7	5	6	6	7

Now, the number of engines used for each season with respect to all the scenarios under investigation for all the compressor stations as a result of the TET profile analysis has been achieved. There is a need to combine all CS result to see the number of GTs used in each CS

5.5.4 Number of Engine investment for Project Life due to Degradation

The process on TET profile analysis in Figure 5-6 to Figure 5-15 and the summary of the number of engines used for each season with respect to the various scenarios under investigation in Table 5-2 were examined on all the 18 compressor stations (CS). Combining the investigation of all the 18 CS gives the result in Figure 5-16 to Figure 5-19. Figure 5-16 shows the number of GT used for the entire project life in each of the scenarios.

For the technical and economic reason, only the most pessimistic case/operation (hot season) was presented. The reason was that it would be the best determinant factor for the number of engines to be procured since both the ambient temperature and degradation are high at this season. The result was obtained by combining all the results from each CS together. The result showed that the clean condition used 90 GTs all through the project life while the opt, med and pess alternates depending on the level of engine degradation and the ambient temperature of the day. From the result, opt scenario used between 90, 91, 94 and 99 GTs. The med scenarios, used between 90, 91, 92, 95, 100, 102, 104, and 106 GTs, while the pess., scenario used between 90, 91, 102, 103, 105, 107, 110, 111, 113, 116, 118 and 120 GTs

Figure 5-17 represents the number of engines used for each compression station at different scenarios throughout the project life. From the result, station 1 and 18 used 5 GT throughout the period of project life. All the compressor stations used 5 GTs for the clean condition throughout the project life. Station 2 used 5 GTs for optimistic while medium and pessimistic used 6 and 7 GTs respectively. Station 3, 4, 5 and 6 used 6 GTs for optimistic and medium while 7 GTs for pessimistic scenarios. Station 7 used 5 GTs for optimistic, 6 GTs for medium and 7 GTs for the pessimistic scenario.

Station 8 used 6 GTs for optimistic and medium while pessimistic used 7 GTs. Station 9 used 5 GTs for optimistic while medium and pessimistic used 6 GTs. Station 10 used 5 GTs for optimistic and medium while pessimistic used 6 GTs. Station 11 used 6 GTs for optimistic and medium while 7 GTs for the pessimistic scenario. Station 11 and 12 used 6 GTs for optimistic and medium while pessimistic used 7 GTs. Station 13 used 6, 7 and 8 GTs for the optimistic, medium and pessimistic scenario. Station 14 used 5, 6 and 7 GTs for optimistic, medium and pessimistic scenario respectively.

Station 15 used 6 GTs for both optimistic and medium while pessimistic used 7 GTs. Station 16 used 5, 6, 7 GTs for optimistic, medium and pessimistic respectively. Finally, station 17 used 5 GTs for optimistic while medium and pessimistic used 6 GTs. This happens because of the difference in ambient condition, mixed fleet of engines used and levels of degradation for each year. Figure 5-18 summarised the highest number of GTs used in different scenarios. It is also presented here that, clean condition for the entire 18 compressor station for the 30 years of project life used 90 GTs, optimistic used 99 GTs, the medium used 106 while pessimistic used 120. Finally, Figure 5-19 shows the number in the mixed fleets of each engine used at different scenarios. The summation of each of these different fleets gives the total number of engines used in each scenario as represented in Figure 5-18

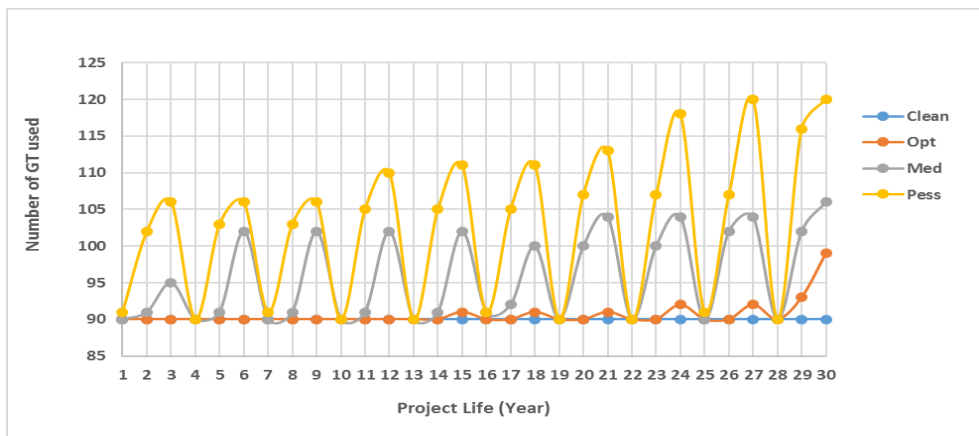


Figure 5-16 Number of GTs used for project life at different scenarios

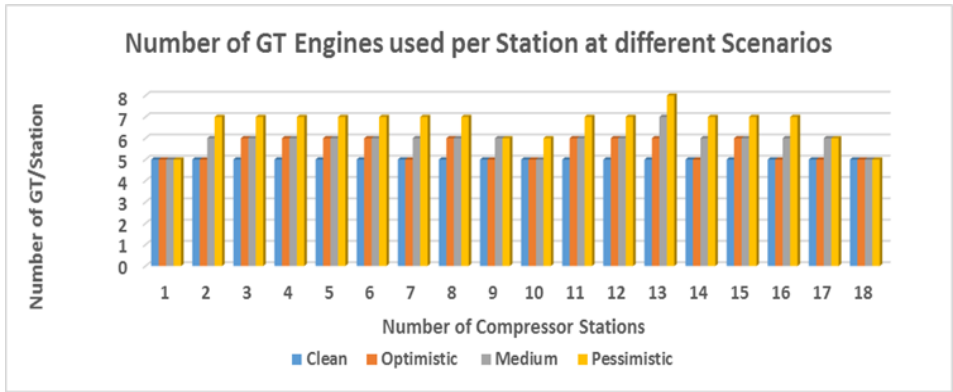


Figure 5-17 Number of GTs used for 18 CS at various scenarios

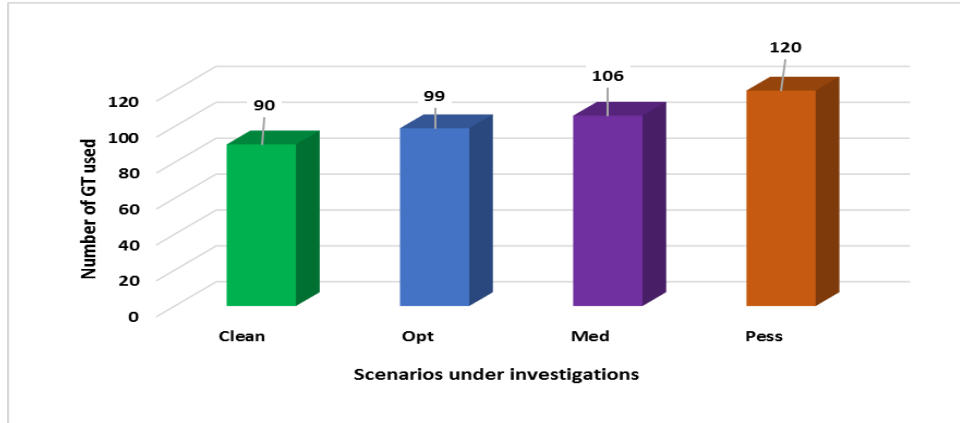


Figure 5-18 Number of GTs used at various scenarios for project life

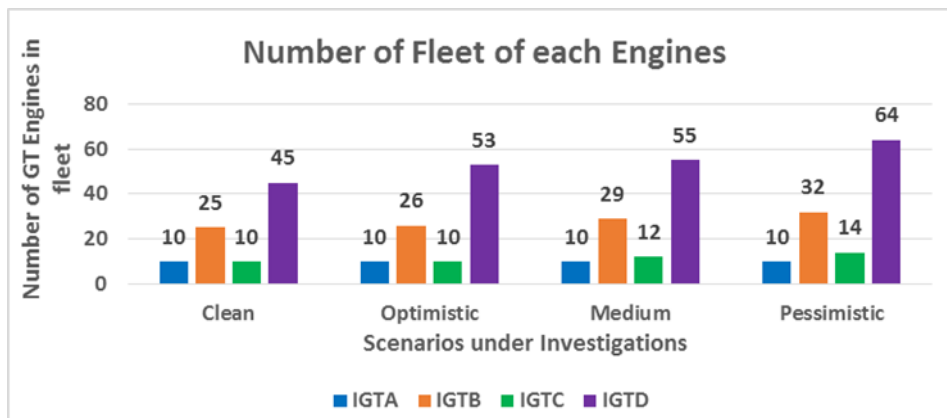


Figure 5-19 Number of different mixed fleets used at various scenarios

5.6 Conclusion

Focusing on the gas turbine engine as an approved prime mover for the pipeline business becomes an important adventure.

The engine TET conjugate profile map for different seasons of the year that help to determine the number of GT to be used per station for the four scenarios under investigation has been analysed and presented. How the impacts of ambient temperature and degradation influenced the number of GT to be used in a particular compressor station throughout the life of the project at a controlled TET without altering the initial pipeline design was also analysed and presented. The result showed that even if the original design of the compressor station was to use 4GTs to 4GCs per station, the impacts of degradation and ambient temperature when the control system set out a limit on the TET, in this case not exceeding an increase in

60K above DP during operation has greatly influenced the use of additional GTs. This help to preserve the engine salvage value for the entire project life. The total number of GT used for the clean, optimistic, medium and pessimistic condition are 90, 100, 106 and 120 respectively for the entire 18 compressor station for 30 years of operation as represented in Figure 5-18. Interestingly, the research was also able to deduce the actual number of the individual GT in the mixed fleets that can conveniently serve for the entire project life, as shown in Figure 5-19. The result of the number of engines is useful in today's industry, especially when the pipeline design and the number of the compressor station for the gas transportation are not expected to change throughout the project life.

Improving the availability of the GT units on each of the compressor station is as important as improving the reliability expectancy of units during the planning phase. Prime mover availability and the causes of unavailability constitute essential performance indicator for assessing service delivery by the gas turbine for pipeline operation; this will help the investor plan ahead of time to ensure uninterrupted delivery of gas to the consumer

5.7 Gas turbine Emission Evaluation on Natural Gas Pipeline

5.7.1 Introduction

The effects of pollutant emission on human health, nature, and materials have become of great public concern. In the past and recent years, rapid changes have been observed both in the regulations for controlling gas turbine emissions and in the technology used to meet this regulation. Introducing some stringent environmental regulations has led to gas turbine dealers to change their focus towards the design of gas turbine with low emissions to meet these regulations. Emissions are among the fundamental challenges in the development of the gas turbine to meet environmental regulations. There are various types of emission generated from the GT engine operation. In this section, CO₂ and NO_x are the two types of emission that are considered in this research.

CO₂ and water vapour (H₂O) has been recognised as the primary products of complete combustion of air and hydrocarbon fuels. Even though these two by-products are not categorised as pollutants, there are environmental and health implications that are associated with this by-product. Also, the emission of NO_x and

CO from the power plant has been a great concern in recent years. The reason is because of the direct impact of exhaust emissions on the quality of air and their inherent potential to cause environmental degradation and deterioration of human health. There are strong indications that the greenhouse gases (GHGs) emitted from industrial power plants are contributing to the increase in ambient temperatures across the globe, a phenomenon referred to as climate change.

Following the impacts of time-based ambient temperature and degradation on the performance and the determination of the number of GT engines used for each scenario under investigation in the first section of this chapter, this section looks into quantifying the amount of emissions (CO_2 , and NO_x) that could be generated during operation of these engines on the entire 18 compressor stations (CS) and to investigate the influence of part-load GT operation on the emission of NO_x .

5.8 Methodology

The only emissions to be considered in this research are NO_x and CO_2 . Considering the vast performance data from different power settings of the engines in all 18 compression stations from TURBOMATCH simulation, it becomes imperative to develop an emission model that would quantify the emission generated in a time-based ambient condition and degradation.

Based on this need, an emission model was developed in MATLAB based on the empirical equations found in the literature, and the necessary thermodynamic equation as explained in this section. The relevant equations, in addition to the programming syntaxes, were applied with respect to the various scenarios under investigation in each season. Figure 5-20 illustrates step by step calculation methods adopted to evaluate the total emissions at the various scenarios under investigation and in each season of the year for the 18 CS

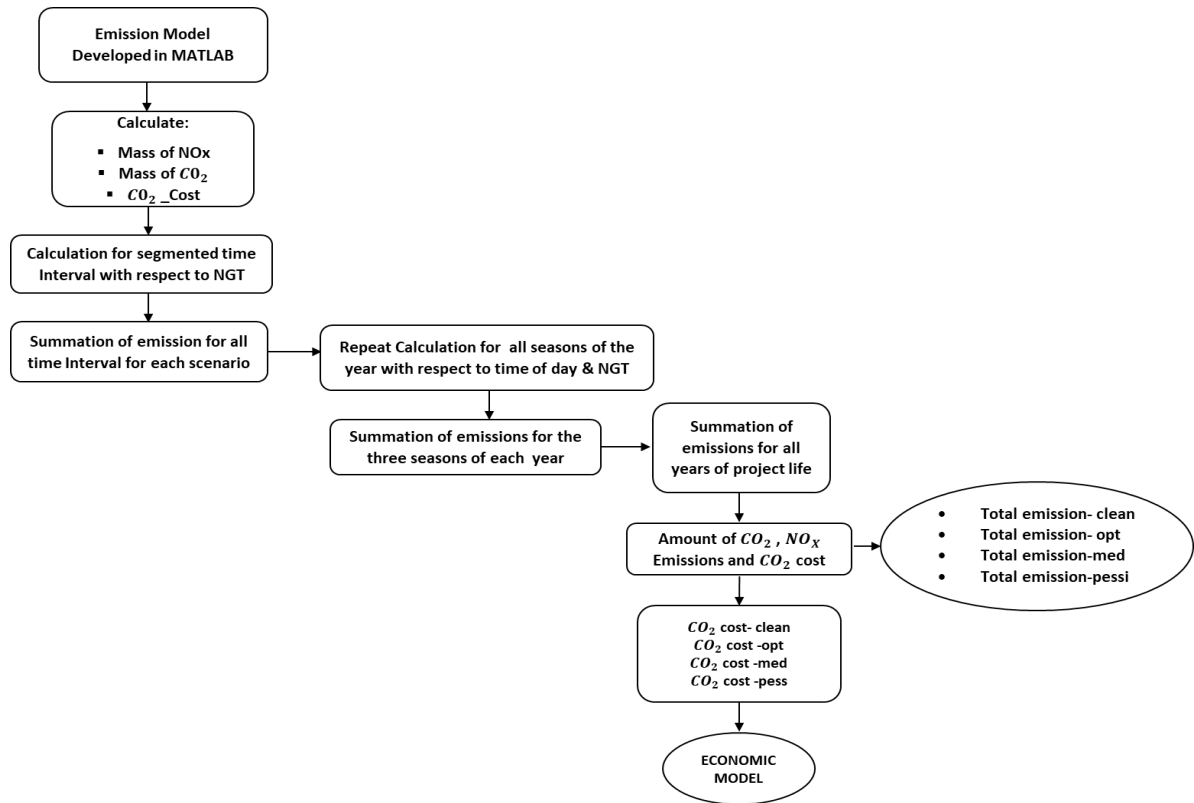


Figure 5-20 Emission calculation method

Considering the emission evaluation procedure proposed in this research, the number of engines to be used in each compressor station must be ascertained based on the TET constraint, as explained in the previous section. The time-based ambient temperature and the implanted degradation in each season were inputted into the TURBOMATCH model, and the economic performance parameters were obtained. The parameters were obtained based on the fact that the engine operational TET must not be 60K above the DP. These performance parameters were further used in the emission MODEL. The main aim of the model was to calculate the mass of NOx and CO₂, in each of the seasons of the year with respect to the four scenarios under investigation and subsequently produce the total emission throughout the life of the project. The model also aimed to evaluate the total emission for each compression station throughout the life of the project.

The calculation was done with respect to the segmented three-hourly temperature of the day in addition to the impact of degradation of the engines. The model added the impact of the time-based emission in each season together. The calculation was repeated for all seasons of the year. After that, the summation of emissions for the

three seasons to achieve the annual emission produced during the operation. For the thirty years of project life, the total emission for the clean, optimistic, medium and pessimistic scenario was calculated.

5.8.1 NOx Emission Calculation

The empirical equation, as produced by Rokke et al. [142] and also used by Lefebvre[138] is represented in Equation (5-4)

$$M_{NOX} = 1.46 \left(\frac{P_2}{P_1} \right)^{1.42} * M_{air}^{0.3} * Fa^{0.72} NGT * ff * time (s) \quad (5-4)$$

The actual NOx in kg can be deduced using Equation(5-4)

Where,

M_{NOX} -Mass of Nitrogen oxide (kg)

P_2/P_1 - pressure ratio

M_{air} - combustor airflow

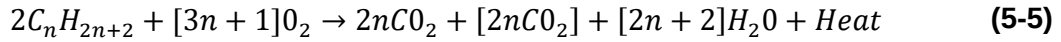
Fa-fuel/air ratio

ff-fuel flow (kg/s); NGT- Number of GT engines

The proposed equation gives a good agreement with measured NOx value for the range of machine in this research. As earlier identified in the literature, the combustion temperature is noticeably absent in the equations, its influence on NOx emission is compensated by the inclusion of fuel/air ratio term[180]. This empirical correlation for calculating NOx emission is within the range of this research. Also, the modelled engines are within the range of the shaft power accommodated by this correlation; hence, the strong reason for using the equation in developing the code in MATLAB to quantify the amount of NOx.

5.8.2 CO₂ Emission Calculation

CO₂ Emission calculation was hinged on the natural assumption that complete combustion of the fuel injected into the engine occurs in the presence of excess air. Natural gas belongs to the alkane group of hydrocarbon, which is predominantly methane. Equation (5-5) shows the general chemical equation for the complete combustion of a hydrocarbon fuel of the alkane group.

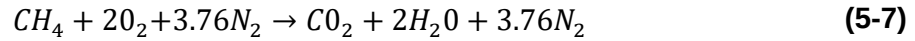


In generic form, the equation for combustion for all hydrocarbon fuel is represented thus:



Considering a molar balance of Equation (5-6), at $n_1 = x$, $n_2 = 0.5y$, and $n = n_1 + 0.5n_2$

The equation for the complete combustion of methane is given thus:



The reaction of 1 mole of methane with 2 moles of oxygen produces 1 mole of carbon iv oxide.

Hydrogen has 1.00794 g/mol, while Carbon has 12.0107 g/mol. This implies that 1 mole of methane weighs:

$$[12.0107 * 1] + [4 * 1.00794] = 16.04 \frac{g}{mol} \quad (5-8)$$

1 mole of CO_2 weighs:

$$[12.0107 * 1] + [15.9994 * 2] = 44.01 \frac{g}{mol} \quad (5-9)$$

This is the CO_2 produced for 1 mole of fuel, of course, the molecular weight of the fuel (gas) was also established. The mass of fuel consumed by the gas turbine over the operating condition and with respect to the various scenarios (clean, opt, med and pess) were obtained from the gas turbine simulation result. The amount of fuel flow results from the engine model was further used in the developed MATLAB code to calculate the amount of CO_2 produced for the entire project life.

Given the analysis from Equation (5-5) to (5-9) with the assumption of no Nitrogen, the amount of CO_2 produced in kg can be deduced from equation (5-10). Meanwhile, it was assumed that 99.9% of the fuel was fully combusted and 0.02 \$/kg was used for CO_2 emission tax[68]. Also, each season was assumed to have an equal number of days.

$$CO_2(kg) = X * \left\{ FF * OPT(s) * NGT * \frac{44.01}{16.04} * 10^{-3} \right\} \quad (5-10)$$

Where

X= Percentage of fuel burn

FF= Fuel flow (kg/s)

OPT = Operating Time (second)

Furthermore, having established the mass of CO₂ in kg, the cost of CO₂ emission was calculated using Equation (5-11)

$$CO_{2Cost} = CO_2(kg) * CO_{2tax} * NGT \quad (5-11)$$

The CO₂ cost was calculated in each season of engine operation and further sum up to an annual cost of CO₂ emission. This is necessary because engine degradation increases with time. The emission cost goes into the economic module as discussed in the subsequent chapter where its impacts on the life cycle cost of the natural gas pipeline were calculated

5.9 Result and Discussion

Figure 5-21) and Figure 5-22) shows the emission of CO₂ and NO_x for the winter season of compressor station 18. The curves show the various scenarios of each emission under investigation (clean, opt. med and pess). Figure 5-21 shows the CO₂ emission at constant power, indicating an equal number of GT engines used at this compressor station throughout the project life in all the four scenarios under investigation. The result showed that as engine degradation increases, the CO₂ emission produced increases also. This is represented when the pessimistic scenarios were higher than the medium and the medium higher than the optimistic while the clean engine has the lowest constant emission.

It should be recalled that the engine overhaul was implemented after three years interval (at year 4, 7, 10, 13, 16, 19, 22, 25, and 28), this implies that the emission produced at the years of engine overhaul was almost the same with the clean condition. The reason was that, after engine overhaul, the engine was assumed to operate almost like the clean engine. The emission produced at the clean condition

was constant throughout the project life because there was no degradation at the clean condition. More importantly, the result revealed that for every 1% degradation (reduction in flow capacity and isentropic efficiency) at a constant power of engine operation, between an ambient temperature of 16.2°C and 29°C, CO₂ emission increases between 0.71% and 0.78% when compared with the clean condition. It was also observed that the amount of CO₂ emitted by the engine was much higher than all the emission considered in this research. This is because the gas turbine engine is believed to have complete combustion. As an air-breathing machine, it operates under the presence of sufficient or excess air capable of burning the fuel injected into the combustor, hence the reason for having higher CO₂. Finally, the increasing trend of the CO₂ emission in each scenario as the years of operation increases shows the effects of the unrecovered degradation after each maintenance cycle. This implies that as the engine continues operation after each segment of an overhaul, the unrecovered degradation increases as the year increases also. The increase in the unrecovered degradation results in the increasing amount of fuel burns, hence the reason for the increasing trend in each scenario. The trend applied to NO_x emission also.

The NO_x emission is presented in Figure 5-22 where the NO_x emission increases with increases in engine degradation at constant shaft power operation. It was also noticed here that for every 1% engine degradation (reduction in flow capacity and isentropic efficiency), within an operational ambient temperature of 16.2°C and 29°C, the NO_x emission increases between 1.66% and 1.8%. However, NO_x emission at different compressor station varies from one station to another due to different ambient conditions and different engine power settings

Conversely, the result of NO_x emitted in some of the CS, for instance, CS 4 for the winter season shows that the NO_x emission was high at the clean condition of the engine different from station 18 that used constant power throughout the operation. This happened because the power setting as a result of using a fewer number of engines at the clean condition was higher compared to other scenarios. At the optimistic, medium and pessimistic scenario, the engine was meant to run on path load (lower power setting) as a result of an increasing number of GT used in the various compression station, this makes the individual engine operational power produced to be low due to gas compressor power requirement. This implies that

increasing the number of GT used in each CS, enabled the use of the engine lower power settings. This affirmed the fact that NOx increases with increasing power settings and reduces with reducing power settings.

It is important to state that this analysis was done for all seasons and in all the 18 CS. The result showed that as degradation increases with respect to the ambient temperature in each season of the year, so also the emission produced at constant engine power operation increases. However, this condition was altered when the power settings of the engine reduce (part-load operation). The summation of the emission from the three seasons of the year gives the annual emission.

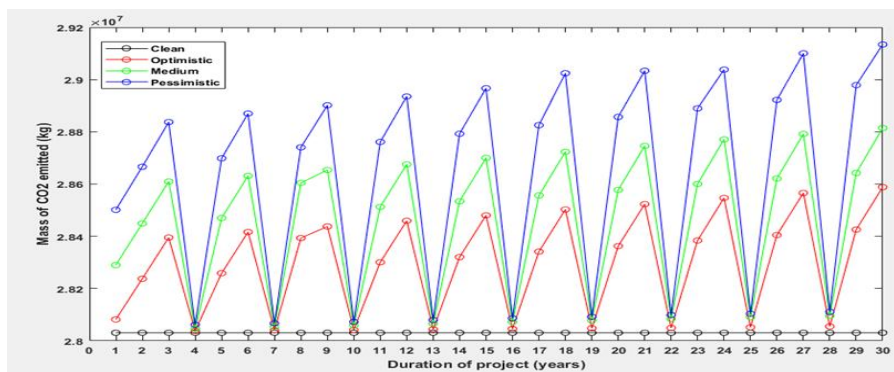


Figure 5-21 Annual CO2 emission for the winter season

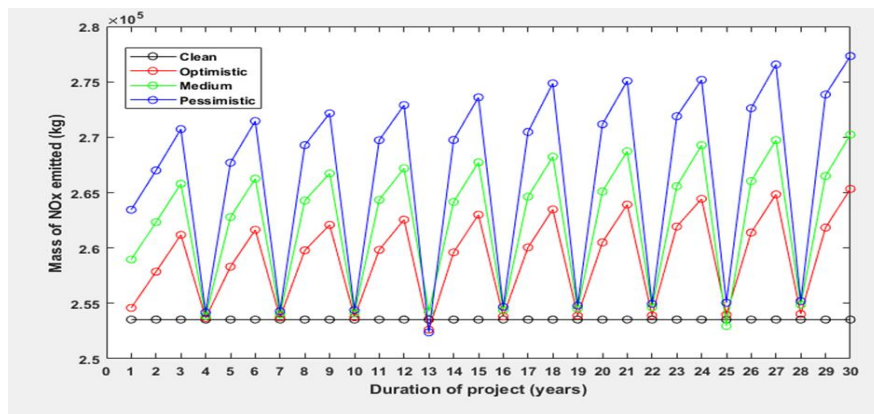


Figure 5-22 Annual NOx emission for the winter season

The above result shows the emissions for the winter season of compressor station (CS) 18 that used the same number of GT with increasing TET. All the CS except CS 1 and 18 alternate the number of GT usage due to increasing degradation. This implies that, as the number of GT increases, so also the TET reduces due to part-load operation of the engine, as shown in Figure 5-23. The curve shows that as the

engine power setting reduces, so also the TET produced reduces. The lower TET produced at the degraded condition has some influence on emission generation.

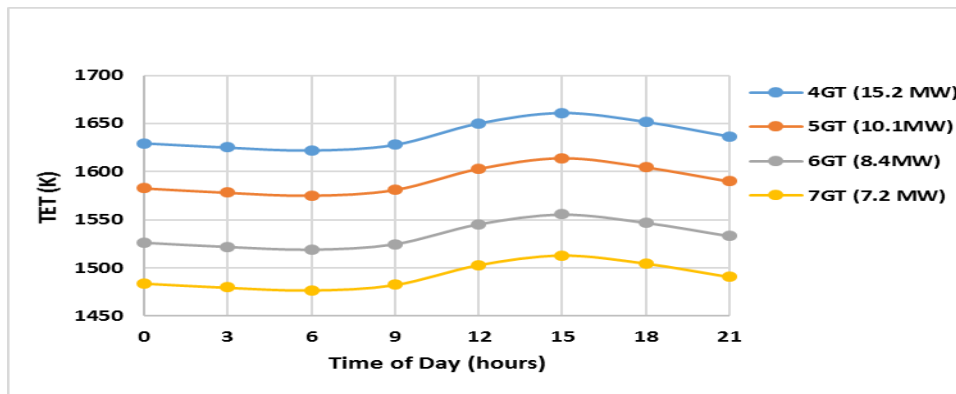


Figure 5-23 The effect of lower power settings on TET

The results from each season throughout the entire CS were evaluated. The addition of the three season's result gives the annual emission produced at the various CS, including the emission tax. Figure 5-24 and Figure 5-25 show the annual CO₂ and NO_x emission, respectively for all the 18 CS throughout the project life. This result was achieved by adding the emissions of the three seasons together. Furthermore, the result shows that the CO₂ emission at the clean condition for the mixed fleets of engine maintained an annual value of 2.46 billion (kg) of CO₂ throughout the project life while the optimistic, medium and pessimistic increases as the year of operation increases, indicating that degradation increases with time. The NO_x emission, as represented in Figure 5-25, maintained an annual value of 19.9 million kg at clean condition throughout the project life all at baseline condition.

Generally, CO₂ emission shows an increasing trend as the engine degradation increases and reduces after overhaul. However, the NO_x emission at the clean condition as represented in Figure 5-25 was higher. The reason is that most of the compression station operate the engine on lower power settings at degraded condition (opt, med and pessi), thereby enabling the engine to produce a lower TET at the degraded scenario than the clean condition. So, the TET produced at the clean engine operation was higher than the degraded condition as represented in Figure Figure 5-23. The result of Figure 5-25 established the fact that NO_x decreases with decreasing power settings, vice-versa.

It is crucial to state that, the clean condition having a higher NOx does not mean that some of the compressor stations did not have higher NOx at the degraded condition. The effect of the higher NOx achieved at the clean condition was the influence of the NOx generated at compressor station 3, 8, 9, 10, 15, 16 and 17 as represented in Figure 5-27. The result of Figure 5-25 showed the total NOx generated from all the 18 CS annually. Because of this reason, the impacts of the higher TET in producing NOx from the clean engine outweighs that of the higher number of engine operating at lower power settings (lower TET) in the degraded condition. However, the variation was felt mostly when the number of engines utilised by the clean condition was not much higher than the degraded condition.

It is worthy of note that emission evaluation was done with different mixed fleets of the engine with various engine capacity and configuration. This implies that each compressor station has different GT engines, and the ambient temperature varies from one compressor station to another. So the NOx reduces as the TET decreases due to part-load operation and increases as the TET increases. The result of NOx emissions also varies from one compressor station to another and from season to season



Figure 5-24 Annual CO2 for all CS and seasons of project life

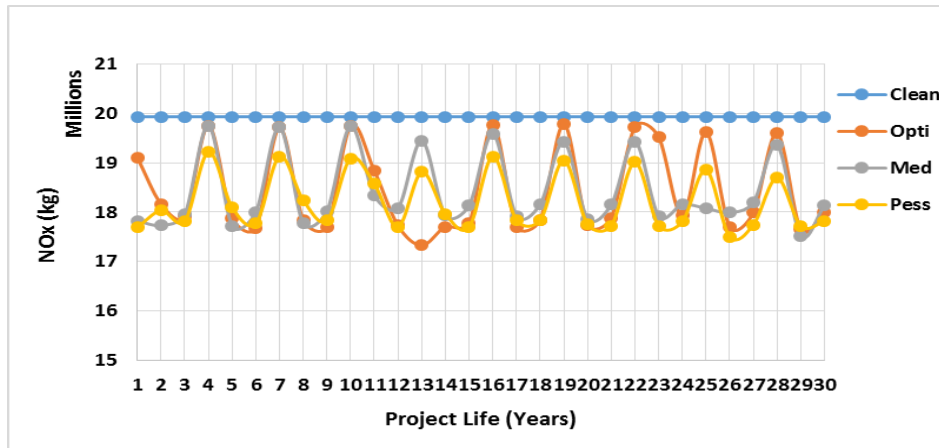


Figure 5-25 Annual NOx for all CS and seasons of project life

5.9.1 Emissions produced at Each Compressor Station

Having known the total annual emission produced by the engine throughout the project life, also, since the number of engines used at the various CS was not uniform, it becomes crucial to evaluate actual values of emissions coming from each CS. This analysis is also necessary because the time-based ambient temperatures that can affect the engine operation vary from one CS to another. Figure 5-26, and Figure 5-27 show the 30 years CO₂, and NO_x emission, respectively, for the individual CS when all the three seasons were added together. It is essential to state here that the variation in emission produced in the various compressor station is as a result of the different number of mixed fleets of engines used in each CS due to the impacts of ambient temperature and degradation

Various scenarios under investigation (clean, opt, med and pess.) are also presented in each compressor station. The results show that the emissions produced with respect to the various scenarios in each CS are not linear. For instance, in station 9, 10, 16 and 17 of Figure 5-26 for CO₂, the clean condition was higher than the optimistic and medium scenario. The result indicates that the fuel flow at the clean condition was high due to higher operational TET as a result of using a lower number of GT, though the influence of the higher percentage of degradation in the pessimistic scenario makes the emission at this scenario to be higher than the clean condition. Generally, the degraded condition releases more emissions than the clean engine operation. This is because as the operation continues, the fuel flow increases as a result of increasing TET. This happened because the engine power for the gas

compressor drive was fixed throughout the life of the operation. However, the result for NOx in Figure 5-27 shows that the NOx emission of the clean condition from most of the compressor stations was higher than the degraded condition. This is because the majority of the CS used a higher number of GT. The condition makes the power settings of those stations to be lower, thereby enabling the engines to run on lower TET as earlier explained. This also affirmed the fact that NOx increases with increasing power settings and reduces with reducing power settings.

It should be noted that as the year of operation increases, the number of gas turbine utilised increases also. This means that a higher number of GT used in each CS implies less value of engine TET as a result of lower power settings with a less fuel burn per engine. Hence the reason for lower NOx emission at the degraded scenarios.

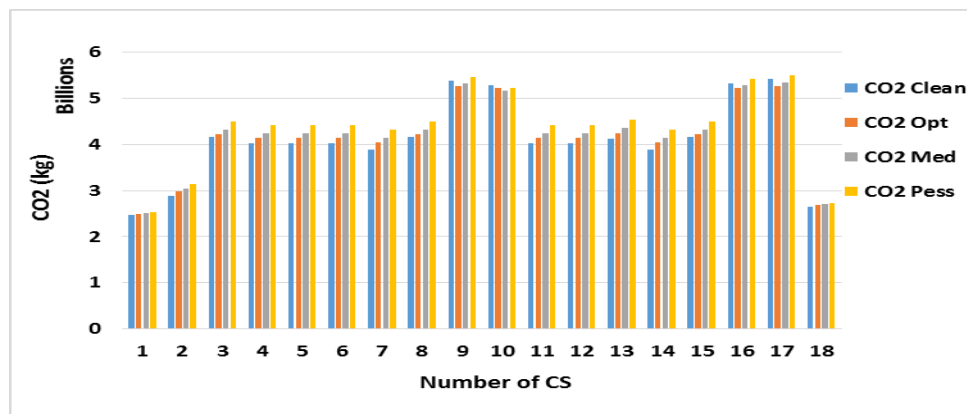


Figure 5-26 Annual CO₂ emission for each CS throughout the project life

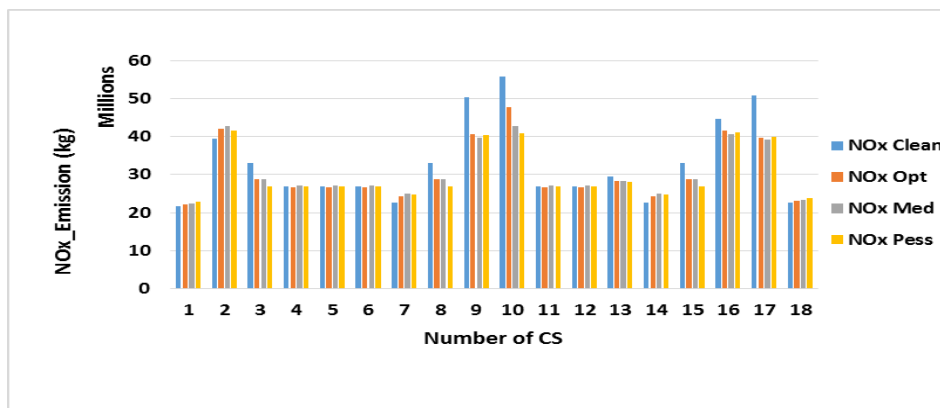


Figure 5-27 Annual NOx emission for each CS throughout the project life

Having known the values of emissions produced at each CS, Figure 5-28, and, Figure 5-29 shows the total mass of CO₂ and NO_x emission produced from the 18 CS throughout the project life. This is the total mass of emission produced throughout the project life with respect to the various scenarios under investigation. The result of the CO₂ emission in Figure 5-28 revealed that the clean condition produced 73.4 billion kg of CO₂. However, the optimistic, medium and pessimistic scenarios produce 1.8%, 3.5% and 7.1%, respectively higher, when compared with the clean condition. This clearly shows the impact of gas turbine degradation on emission production.

Figure 5-29 indicates the total NO_x emission produced throughout the project life at the various scenarios. Unlike the CO₂ emissions, the clean condition was found higher than the degraded conditions. In this case, the optimistic, medium and pessimistic scenarios were 7.4%, 7.9%, and 9.2% respectively lower than the clean condition. This happened because the power setting at the clean condition of all the compressor stations was higher than all the CS except compressor station 1 and 18 that used an equal number of GT throughout the operation.

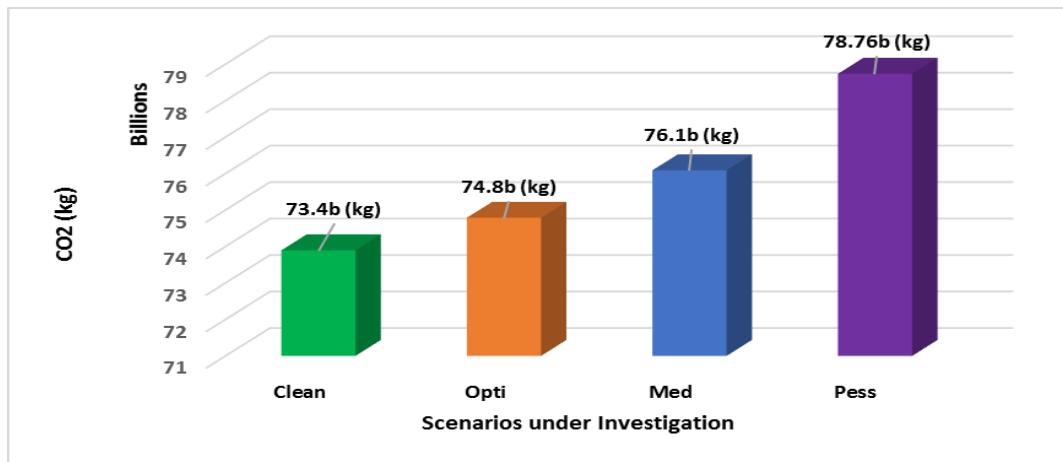


Figure 5-28 Integrated CO₂ generated for project life

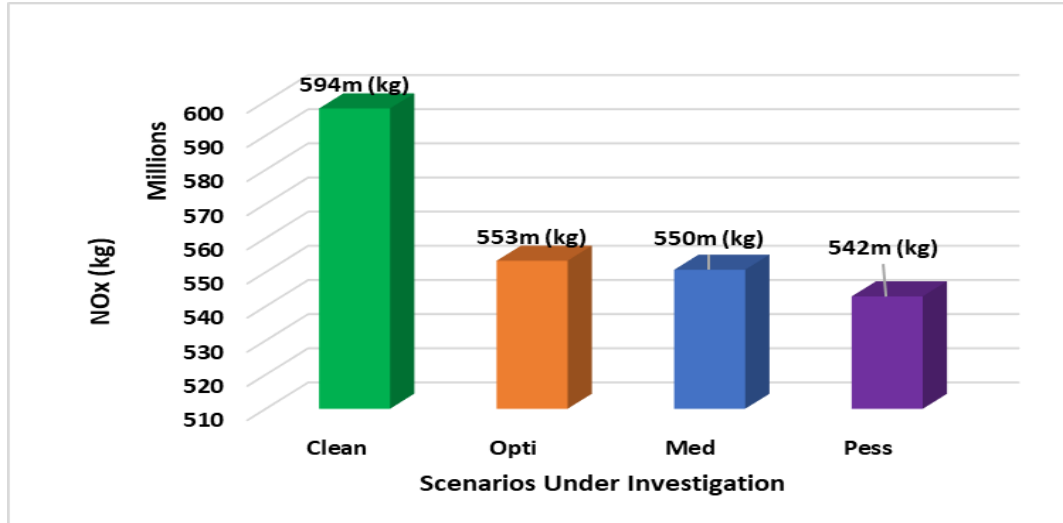


Figure 5-29 Integrated NOx for Project life

5.9.2 Evaluation of emission Tax

Figure 5-30 shows the emission tax for the various scenarios in the winter season of compressor station (CS) 18. It should be recalled that CS 18 used an equal number of GT throughout the operation. This implies that the power settings do not change despite engine degradation. Based on this operational condition, the result shows that CO₂ emission tax increases with increasing degradation. However, the compressor stations that use a higher number of GT has lower values of emission with respect to a particular scenario. The result further revealed that within an operational ambient temperature of 16.2°C and 29°C and for every 1% engine degradation (reduction in flow capacity and isentropic efficiency), the CO₂ tax increases between 0.71% and 0.78% when compared with the clean condition for a 13.4 MW engine. It was also observed that the emission tax increases from winter season with lower ambient temperature to the hot season with the higher ambient temperature in addition with increasing engine degradation at different seasons of the year as represented in Figure 5-30, Figure 5-31 and Figure 5-32, for winter, dry and hot season respectively

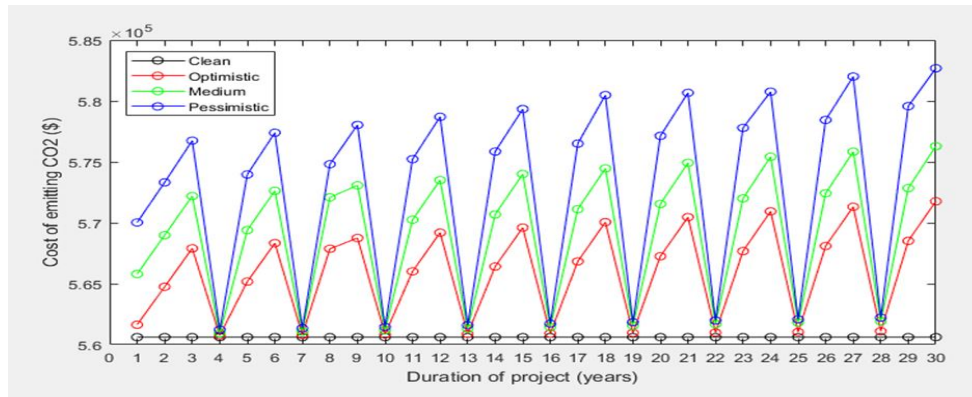


Figure 5-30 Annual emission tax for the winter season

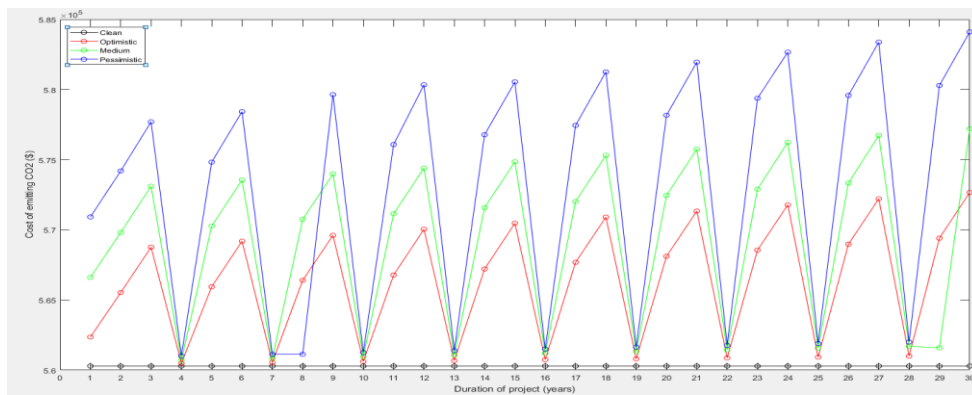


Figure 5-31 Annual emission tax for dry season

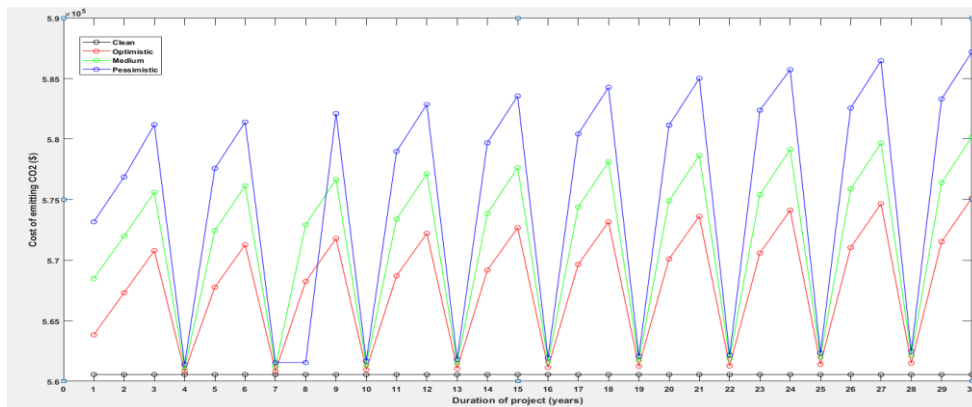


Figure 5-32 Annual emission tax for the hot season

The analysis above was repeated for all the compressor stations in all the seasons of the year with respect to the various scenarios under investigation. The summation of the three seasons per CS in each year throughout the project life is shown in Figure 5-33. The result showed that the emission tax for the clean engine remains the same in all the years of operation with a constant annual tax of \$49.26 million.

The reason has been that it was assumed the engines remain clean throughout the operation of project life. The result also shows that emission tax increases with increasing degradation. Meanwhile, the increasing trend in each scenario indicates the effects of unrecovered degradation on the emission tax after each maintenance cycle. However, emission produced at the years of engine overhaul was almost like the clean conditions.

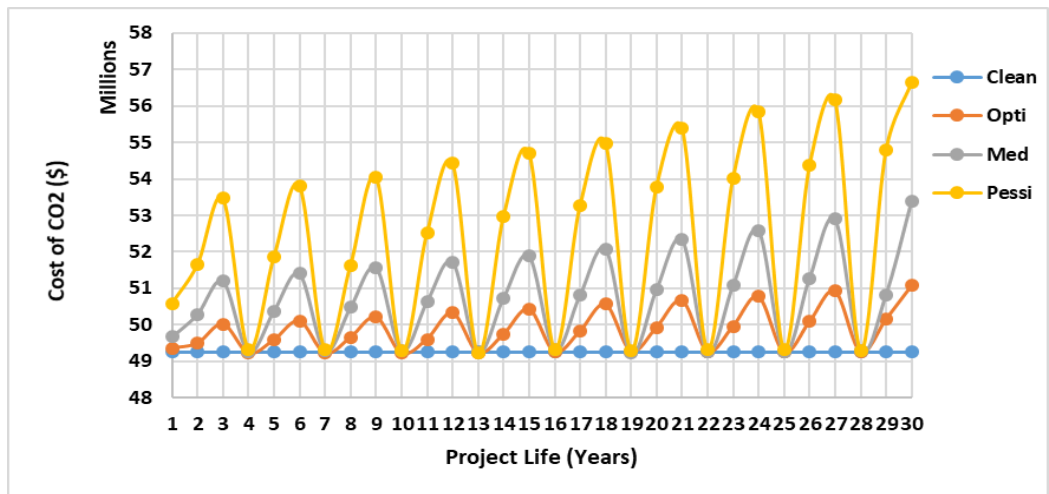


Figure 5-33 Annual cost of CO2 emission for the project life

The result for the emission tax produced at the various compressor station is presented in Figure 5-34. The result revealed the impacts of different power setting of the mixed fleets engines and the variable number of engines used at the different compressor station

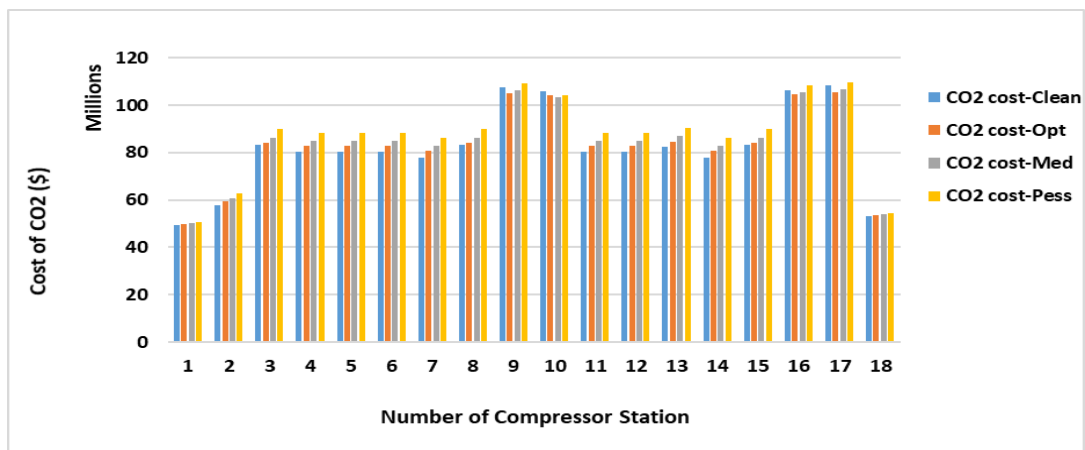


Figure 5-34 Annual cost of CO₂ emission for each CS for the project life

Figure 5-35 shows the total CO₂ emission cost produced from the 18 CS throughout the project life. The result shows the total emission cost for various scenarios under investigation. From the result, the effect of degradation in addition to the higher number of GT at different compressor station makes the emission tax for the degraded scenarios to be higher than the clean condition. The result further revealed that the CO₂ tax for the clean condition was \$1.46 billion USD while the optimistic, medium and pessimistic scenario was 2.7%, 4.8% %, and 8.2%, respectively higher when compared with the emission tax of the clean condition. This also affirmed the facts that running the gas turbine on degraded mode increases the CO₂ tax. It should be noted here that wide margin was not felt between the various scenarios because the engine was not allowed to run more than 60K above the DP, thereby making the degraded conditions to run on part load most time of the operation

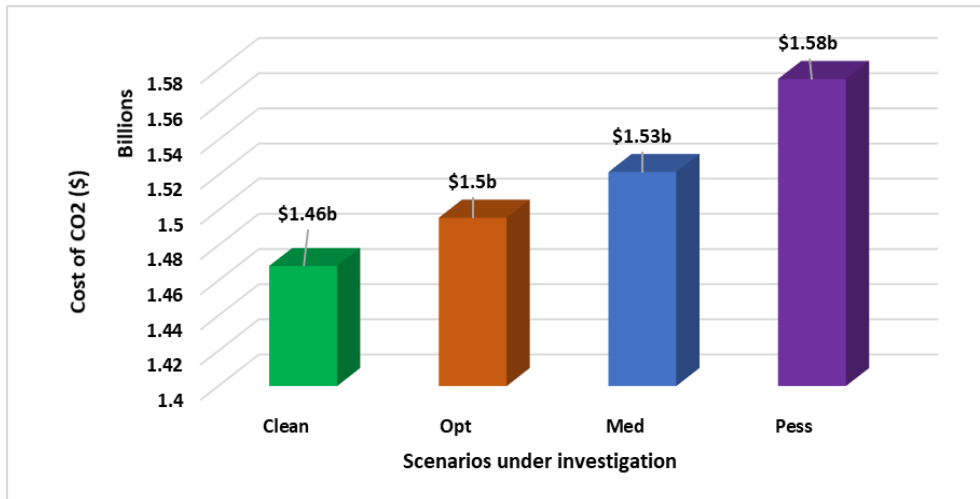


Figure 5-35 Total costs of CO₂ emission for the project life

5.9.3 Conclusion

Having identified and analysed 18 compressor stations used as booster station and the required number of GT engines at each CS for the natural gas transportation, this chapter evaluated CO₂ and NO_x emission including the CO₂ emission tax on the pipeline operation. This chapter has also described in sequential order the approach used to achieve the objective of emission evaluation in all the CS from the review of relevant literature using relevant correlation, and empirical modelling equations of emission to develop a model in MATLAB.

The developed model was connected with the TURBOMATC performance result. The model produces both technical and economic result such as the emission result (CO₂, and NO_x) and CO₂ emission cost. The result of the emission in each season of the operation with respect to the individual CS has been presented. The total result of emission generated in each compressor station throughout the project life has also been presented. (Both clean and degraded scenarios). The economic implication shows that emission tax increases with increasing engine degradation. The specific result revealed that within an operational ambient temperature of 16.2°C and 29°C and for every 1% engine degradation, the CO₂ tax increases between 0.71% and 0.78% when compared with the clean condition.

6 Development of Economic Model of the Gas Pipeline

The natural gas pipeline business is adjudged to be a capital intensive investment. The main cost associated with pipeline transportation investments is the capital and operating cost. The operating cost consists of the yearly maintenance and fuel cost of the pipeline network. In general terms, the natural gas price comprises of the actual price of the gas commodity and the cost associated with the transportation and distribution of the gas. Bejan et al. [181] presented that economic analysis of a project after some relevant assumptions and predictions covers some major costs such as the

- (i) Total capital expenditure for the project (CAPEX)
- (ii) The operating and maintenance costs (O&M) and
- (iii) Fuel costs

However, the cost elements considered in this research consist of the above-itemised costs and also includes the cost of CO₂ emission and costs related to non-operating activities such as the expenses paid on interest regarding loan repayment. Interestingly, this research produces a method of estimating the capital cost of the various components of natural gas pipeline transportation system based on literature. Also, the O&M costs and fuel costs were investigated in a time-based system under the influence of engine degradation. The generated revenues from the research are derived from the sales of natural gas, engine divestment, and leasing. The engine divestment and leasing came into the analysis as a result of redundant fleets after overhaul. Then the feasibility of the economic investigation was determined by the NPV, and it will be evaluated with respect to the various scenarios under investigation using a developed model in MATLAB.

Various components and accessories constitute the capital cost of pipeline investment. Generally, in the pipeline transportation system, the cost of pipe and compressor stations dominates the initial capital cost. Menon [42] stated that the gas compressor and pipeline takes about 96 percent of the construction materials. Similarly, the major operating cost for pipeline investment is the fuel cost that system utilised both yearly and throughout the project life.

It is very significant to state that without the accuracy of the technical content of the various module of the pipeline investigation, the economic analysis cannot be actualised. Hence, the need to incorporate the TERA framework that will encapsulate the various modules in order to actualise the economic result of the natural gas pipeline investment. Again, looking at the impact of engine degradation for the different scenarios as explained in the previous chapter on the economics implies that each season of the year will have different costs of fuel and maintenance due to a different level of degradation and in addition with the impact of ambient temperature in all the 18 CS.

Having a project life of thirty years with each season of the year having four scenarios needs a tool that could be able to calculate the yearly O&M cost and fuel cost for 18 CS and to evaluate the benefits on investment. Therefore, this chapter presents the development of an economic model in MATLAB to determine the economics of yearly fuel consumption and maintenance cost for the entire 18 CS of the TSGP and subsequently calculate the benefits on investment. The developed model takes into account the time-based ambient temperature and yearly degradation of the GT engines throughout the project life. Most importantly, the impacts of the GT degradation on the economics of natural gas pipeline is been analysed with the model. Also, the various costs of equipment, Life Cycle Cost (LCC) and the net present value on investing in the project are investigated and presented. Having known that the life cycle (LC) of the project will take thirty years, this study annualises the fuel and O&M costs and also determines the revenue stream necessary to remunerate the total investment in the natural gas pipeline project

6.1 Cost Components of the compressor stations under consideration

The compressor station under consideration comprises of several components that aid gas transportation which also constitute the capital cost. These components include the pipeline, gas compressors, gas turbines, all pipeline accessories such as SCADA and telecommunication, mainline valves (MLV), meter station and regulators (MSR), environmental and permitting (E&P), right of ways (ROW) engineering and construction (E&C) and Contingencies. All these components have been discussed in chapter two. The technical, as well as the economic analysis of these components,

elucidates the awareness for investing in a natural gas pipeline. The costs of each of these components constitute the CAPEX of the pipeline project.

O&M cost is one of the significant costs element when analysing the economics of investing in a natural gas pipeline project. In this case, the O&M cost was based on the costs of labour and the costs of keeping the GT working at the desired power setting so as to keep the gas pipeline in an excellent operating condition. The O&M cost consists of fixed, variable and major maintenance cost (TET-based maintenance) while the fuel cost is very important in analysing the economics of pipeline transportation. The advantage of fuel cost on pipeline operation to the operator is that gas used in GT is gas not purchase from an external seller. This is advantageous to the pipeline operators as the GT uses the gas within the pipeline system.

6.1.1 Economic Sensitivity Analysis

Sensitivity analysis here referred to the effect of inflation (escalation rate) on the annual O&M cost and fuel cost. For instance, to account for the inflation of relevant engine parts such as spare parts, consumables, chemicals, etc., an annual escalation rate needs to be implemented. Implementation of escalation rate implies that a certain assumption has to be made for the inflation rate on the fuel and the O&M cost in order to investigate their impacts on the life cycle cost of the project

6.2 Methodology

The methods adopted to achieve the various cost of the project are hereby presented. The adopted method evaluates the cost of pipeline investment for natural gas transportation network using a concept that can capture all variables involved in the overall cost. Figure 6-1 shows the economic module of the project.

As earlier discussed in chapter three, the economic module is the core module of the TERA framework that accommodates all other modules. It does the calculation of the entire project life cycle cost, simulates the cash flow and finally evaluates the economic viability of the project. All these are analysed and presented below. From the flow chart,

- i. the number of engines that satisfy the condition of operation in each compressor station must be ascertained.

- ii. The number of engines divested or leased after overhaul must be known.
- iii. The performance parameters from the different number of engines from TURBOMATCH simulation are connected with the economic model to calculate the fuel cost, O&M cost, cost of CO₂ and LCC
- iv. The developed model will simulate the cash flow (revenue and CAPEX)
- v. finally evaluates the economic viability of the project for the various scenarios under investigation using the NPV approach.

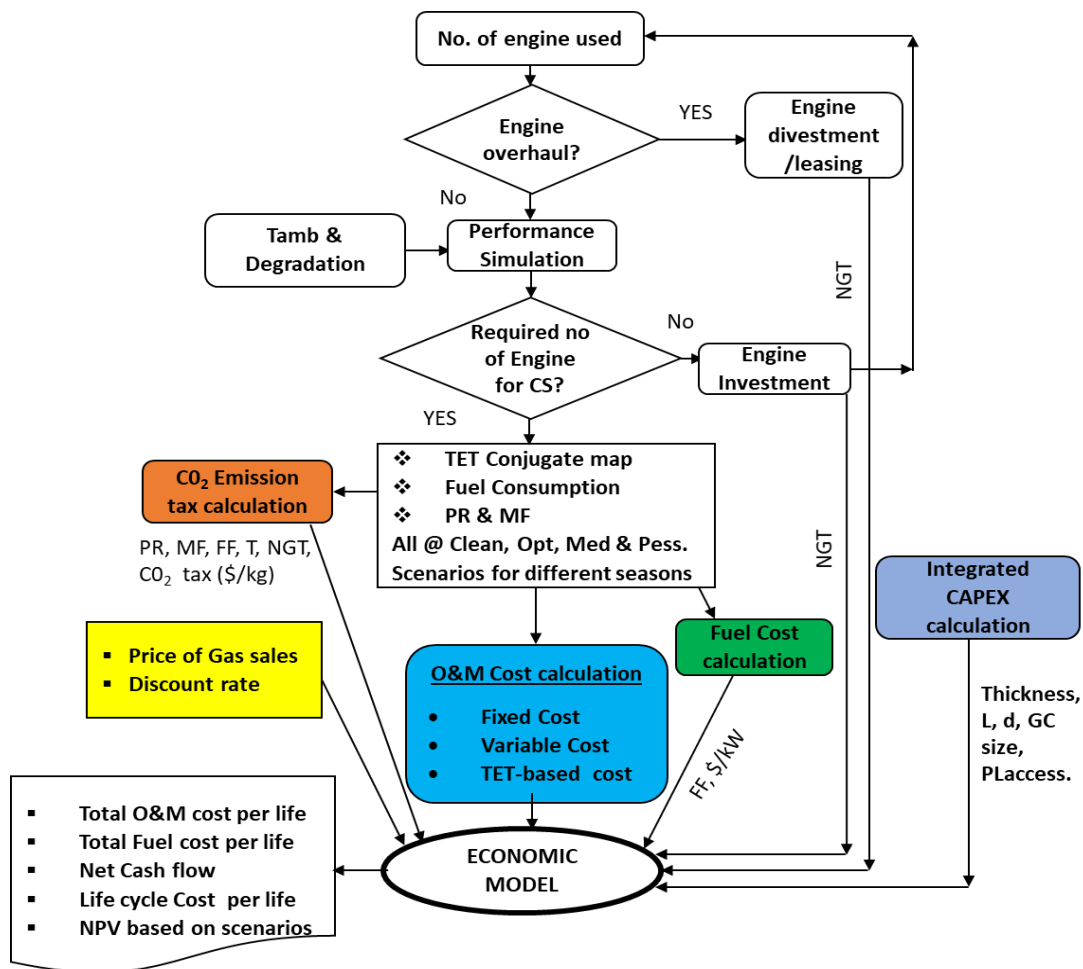


Figure 6-1 Economic Model for pipeline investment

Where, PR-pressure ratio; MF-mass flow (kg/s); T-time of operation (s); L- length of pipe (km); d-diameter (m); GC- gas compressor; PLaccess- Pipeline accessories

6.3 Component Cost

Methods and equations used to calculate the various costs element of the pipeline investment at baseline condition are hereby presented below.

6.3.1 Capital Expenditure (CAPEX) Calculation Method

The pipeline data is shown in Table 6-1. To evaluate the capital expenditure of this project, Figure 6-2 shows the flow framework instituted in this research in order to calculate the capital cost of GT engine, gas compressors, pipelines and all accessories. Meanwhile, the pipeline data is shown in Table 6-1

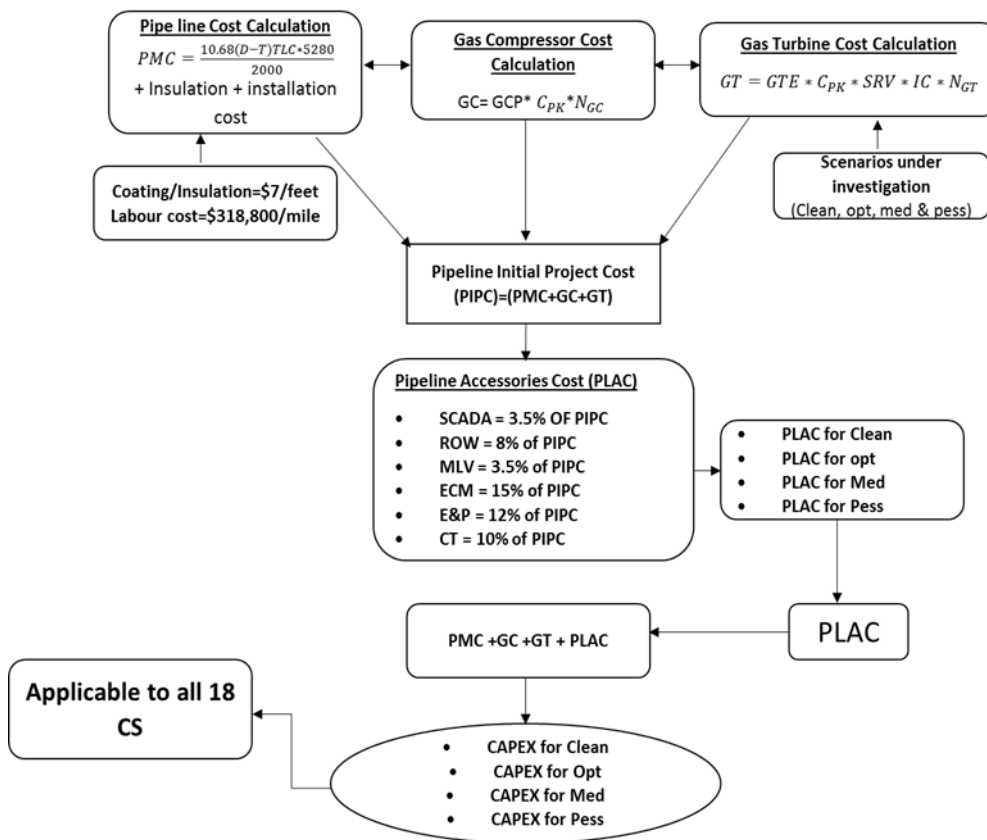


Figure 6-2 Calculation of the capital expenditure (CAPEX)

Table 6-1 Pipeline data[6]

Parameters	Values
Pipe Grade	API 5L X70
Nominal Diameter	56"
Outer Diameter	56" (1422.4mm)
Inner Diameter	54.38" (1381.2mm)
Wall Thickness	0.812" (20.62mm)
Gas flow	30bcm/y
Design Pressure	100 barg
MAOP	95 barg
Design Code	0.72
Roughness	0.003" (7.62μm)
Design Factor	ASME B31.8

Based on Figure 6-2, there are several components that comprise of the capital cost. These components are divided into two parts. One part is the pipeline initial project cost (PIPC) and the second part is the dependent components cost. The PIPC involves the costs of pipelines, GCs and GTs while the dependent cost involves the cost of all the pipeline accessories such as SCADA & telecommunication, ROW, MLV, ECM, E&P and CT. The costs are regarded as a dependent cost because their actual costs are hinged on the costs of the initial project cost as expressed in Figure 6-2. The various components costs are discussed below:

6.3.1.1 Pipeline Cost

The pipeline costs comprise of the cost of material, cost of installation, and insulation. Also, cost evaluation for pipe generally depends on the material used, the diameter of the pipe, wall thickness and the pipe length. Having known the length of the pipeline and all necessary parameters related to the TSGP, the material cost was calculated using

$$PMC = \frac{10.68(D - T)TLC \times 5280}{2000} \quad (6-1)$$

Where, PMC-Pipe material cost, D T, L, and C represent pipe diameter(in), thickness(in), pipe length (mi) and cost of material (\$/ton).The categorisation of welded and seamless pipeline used for gas pipeline and petroleum industries is given by the American petroleum institute (API). This research used AP1 51 and X70 steel grade, which is appropriate for transporting natural gas. The steel material cost was gotten from the pipeline manufacturer's website.[182]

It is important to state that the actual cost of the pipeline is the combination of the cost of pipe material, installation and insulation. In order to achieve the installation cost for a given diameter and length, the equation was obtained from a wealth of historical data offered for the construction of different sizes of pipe from the Institute of transportation study as seen in equation (6-2) and was also used by Tukur[18] in his research work

$$Y = 343.21X^2 + 2073.9X + 170013 \quad (6-2)$$

Where Y and X represent the cost in (\$/mile) and the (diameter/inches) respectively. Also, the cost for insulation was valued in (\$/foot). Therefore in this work, \$5/foot was utilised[22]. Hence, the actual capital cost for the pipeline is given as

$$PL_c = PMC + P_{INSTC} + C_{CI} \quad (6-3)$$

Where PL_c , - pipeline cost; PMC -pipe material cost; P_{INSTC} , - pipe installation cost and C_{CI} -cost of coating and insulation.

6.3.1.2 Compressor Station (CS)

It was assumed that each compressor station accommodates four GC and the required number of GTs based on the site ambient temperature and degradation. It was also assumed that the gas compressor does not degrade over time. Based on these assumptions, the initial cost of the GC was calculated using Equation (6-4)

$$C_c = \text{cost per } \left(\frac{\$}{kW} \right) * \text{Number of GC (KW) installed} * GC_p \quad (6-4)$$

\$3000 was used for the cost per kW (\$/kW). It is important to state here that gas compressor in the market that matches the calculated value of GC in each CS was used to calculate the actual cost of the GC in each CS. This is represented in Appendix F. GC_p represents GC power. The selected GC were rated in hp. The hp was converted to kW by a factor of 0.7457

The gas turbine Handbook has provided a guide on the cost calculation of GTs engines as shown in Figure 6-3, the price (\$/kW) of the engine decreases as the engine capacity increases[64].

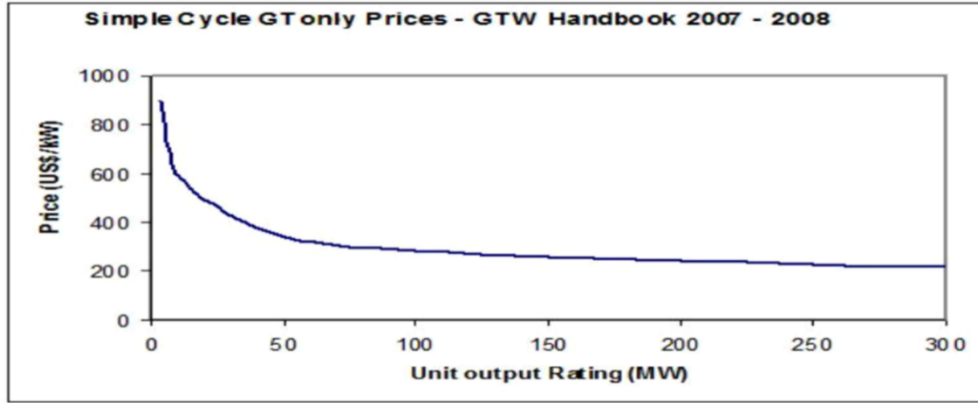


Figure 6-3 Price of gas turbine[64]

Based on Figure 6-3, web plot digitiser was used to retrieve the original values. From there, a trend line polynomial function was produced using a curve-fitting approach. This approach was adopted to accommodate the capacity of the present research engines. After that, the actual cost per kilowatt of the engines was derived based on the polynomial function. From the analysis, IGTA (13.4MW), IGTB (19.3MW), IGTC (17MW) and IGTD (15.2) used \$566.75, \$503, \$524.32 and \$543.86, respectively as the cost per kilowatt of the engines. The result of the cost per kilowatt of the mixed fleets agrees with Gas turbine handbook [64] that the price (\$/kW) of GT engine decreases as the engine capacity increases[64].

Having known the cost per kilowatt, the capital cost of GT was calculated using (6-5)

$$GT_{CC} = GTE * C_{PK} * SRV * IC * N_{GT} \quad (6-5)$$

From Equation (6-5), The capital cost for the gas turbine(GT_{CC}) operating in each compressor station comprises of installation cost (IC), shipping and regional variation (SRV) cost which is compared to the power of the GT (GTE) with its unit cost (Cost per Kilowatt, C_{PK}) and the number of engines per station (N_{GT}).

The following assumptions were utilised in calculating the GT capital cost[18]

- The GT costs were increased per kW by multiplying by 2 to accommodate the installation cost
- Multiply by a factor of 1.38 to cater for shipping and regional variation cost

6.3.1.3 Pipeline accessories calculation[22]

- 3.5% of the PIPC was used to calculate SCADA & telecommunication cost.

- 5% of the PIPC was used to calculate the cost of valve stations and 20miles was adopted along the natural gas pipeline to install the valve
- 8% of the PIPC was used to calculate the cost for ROW.
- 15% of the PIPC was used to calculate the cost associated with the ECM.
- 12% of the PIPC was used to calculate the E&P costs.
- 10% of the PIPC was used to account for contingency costs

6.3.2 Method to Estimate Fuel and O&M cost

To calculate the fuel and O&M cost of the engines used on each of the compressor stations, performance result from the TM code become very important in the evaluation process. Figure 6-4 illustrates fuel and O&M costs concepts showing the conjugate performance data flow from simulation result to estimate the fuel and O&M costs at various scenarios. Conjugate performance data here means that the performance results are coming from a different number of the engine with their respective power settings. This means that the TET and their corresponding fuel flow from 5 GT will be different from that of 6GT, hence the need to produce the conjugate performance data that will serve as the best data for the economic evaluation as shown in chapter 5. In this research, this conjugate data is regarded as the economic conjugate data.

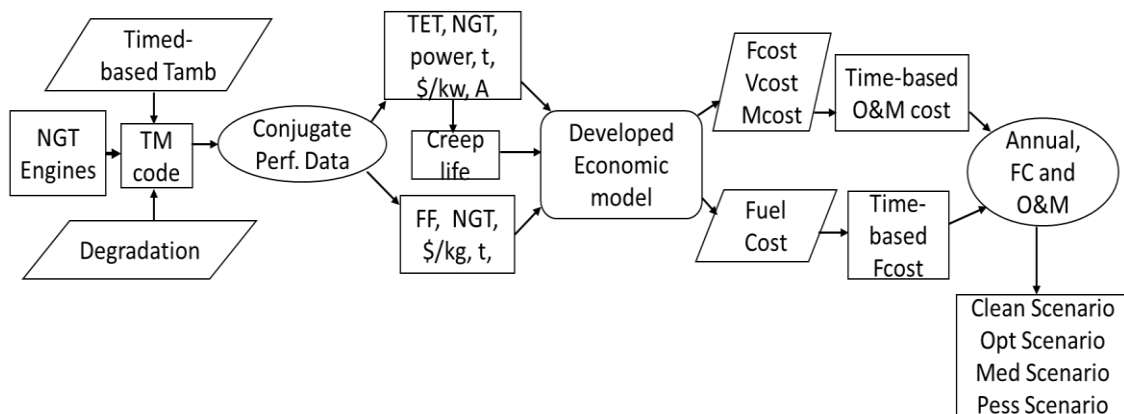


Figure 6-4 Method of calculating Fuel and O&M costs

Total number of GTs at each CS and their respective performance parameters such as TET, fuel flow and varying power settings due to TET limits were utilised by the model. Also, time-based ambient temperature and the implanted degradation for

each season of the year for thirty years were loaded in the economic model for each scenario under investigation.

6.3.3 Operation and Maintenance Costs (O&M)

The maintenance cost comprises of fixed, variable and major maintenance cost (TET-based maintenance cost). The sum of these three different costs gives the entire maintenance of the engine. Equation (6-6), (6-7) and (6-8) shows the calculation of fixed, variable and major maintenance cost respectively.

$$Fixed\ Cost = \left(\frac{\$}{kW} \right) * GT_{shaft\ power}(kW) * NGT * 1000 \quad (6-6)$$

$$Variable\ Cost = \left(\frac{\$}{kW} \right) * GT_{shaft\ power}(kW) * T\ (hours) * Availability * NGT * 1000 \quad (6-7)$$

$$Major_{maintenance\ factor} = \left[\left(\frac{\$}{kW} \right) * \frac{Baseline\ Life}{Creep\ Life} \right] * GT_{shaft\ power}(kW) * T\ (hours) * Availability * NGT * 1000 \quad (6-8)$$

Where NGT = Number of GT ; $\frac{\$}{kW}$ = unit cost per kW ; T= time (hours)

It is worth noting that the variable cost was calculated based on time of day in each season and throughout the project life while the fixed cost was calculated based on the project component without time. It was assumed here that major maintenance cost takes care of engine overhaul. Based on this, it was assumed that the first engine overhaul has 10% less in the total cost when compared with the rest of the engine overhaul. This assumption was made because it was believed that the engine had not had much degradation, and therefore, the cost was expected to be less compared with the rest of the engine overhaul as the operation continues. The baseline life of the engines used in equation (6-8) was the assumed creep life of the GTs engine at DP (25,000 hours). 5.8 \$/kW and 0.23 \$/kW was used for variable and fixed cost calculation[183]. Also, engine availability was assumed to be 100%. Since the variable and the major maintenance cost of the engine varies from season to season and year to year, Equation (6-9) becomes useful in the developed model to

calculate the total operation and maintenance cost in each season, year and entire life of the project.

$$\sum_{n=1}^{n=30} FC + VC + Major_{maintenance\ factor} \quad (6-9)$$

Where,

FC= Fixed cost

VC= Variable cost

T = Time (hours)

NGT = Number of GT

$\frac{\$}{Kw}$ = unit cost per kw

6.3.4 Fuel Cost Calculation

The fuel cost calculation was done using equation (6-10) and (6-11)

$$CS = FP * FF * T * NGT * 3600 \quad (6-10)$$

Since the fuel consumption of the GTs varies from season to season and year to year due to engine degradation and ambient temperature variations, equation (6-10) and (6-11) were used in the developed economic model to calculate the cost of fuel for the pipeline project. To have an accurate quantity of fuel consumed by the selected mixed fleets of engines, daily ambient temperature variation for each CS were segmented into three seasons and hours of the day for 24 hours. The time taken were segmented into three hour differences (0:00, 3:00, 6:00, 9:00, 12:00, 15:00, 18:00 and 21:00 hours of day). The ambient temperatures that correspond with this time of day in each of the CS were used to simulate the various GTs engine throughout the 18 CS. Also, the degradation was implanted in each season of the year, as explained in the preceding chapter.

It implies that each season of each year has different values of degradation throughout the project life. The time-based performance result from TM was used for economic analysis. It is worthy of note that the four scenarios under investigation were calculated for each season of the year for thirty years of the project. Also, the

time (T) was based on the total number of hours in each season. Each season was assumed to have an equal number of days. The sum of the result from each season with respect to their corresponding scenarios gives the annual fuel cost, and this was also applicable to the O&M cost.

The model was able to calculate the total sum up the fuel cost for each season of the individual year and finally produced the life cost of fuel throughout the project life (30 years). The calculation was done for the various scenarios under investigation (clean, opt, med, and pess)

$$\sum_{n=1}^{n=30} FP * FF * T * NGT * 3600 \quad (6-11)$$

The unit cost for fuel consumed by the GT was assumed to be 0.168 (\$/kg) [18]. It is worth noting that the GT performance parameters such as fuel flow, Turbine entry temperature, including the different shaft power from TM result in each compressor station, was connected to the developed economic model. The connection was made based on the TM result from the different scenarios under investigation to calculate the O&M and fuel cost for the entire 18 compressor station for thirty years.

6.3.5 Economic Sensitivity Analysis

In this research, 0%, 3%, and 5% escalation rate were implemented for the different scenarios under investigation. The main aim was to investigate the effects of inflation on the life cycle cost of the project

6.4 Result and Discussion

The result for the cost of the pipeline, gas compressor (GC), gas turbine (GT) and all pipeline accessories considered in this research are hereby presented as shown in Figure 6-5 to Figure 6-8.

6.4.1 The impacts of GT Degradation on the Project CAPEX

Figure 6-5 shows the impact of degradation on the cost of GT for the pipeline application. The result shows that each compressor station has their different cost for each of the different scenarios under investigation. The result is expected because of the different number of GT used per station, which also varies according

to each scenario. For instance, CS 1, 2, 3, 4, and 13 for pessimistic scenario used 5, 6, 7, 7 and 8 GTs respectively, and so also the cost varies. The reason for the variation in the number of GT used in each station has been explained in the preceding chapter. The different fleet of the engine also has its various cost associated to them due to their capacity, hence one of the reasons for the variation of cost. For instance, station 1 and 18 used 13.4 MW engine while the rest used a higher power rating. This is the reason why station 1 and 18 cost \$0.1 billion as the lowest cost while station 16 and 17 cost \$0.15 billion as the highest cost for the total number of clean engine operation while the degraded condition varies from one compressor station to another. Meanwhile, Figure 6-6 shows the total cost of GT engines at different scenarios. the result showed that the total cost of GT at clean condition is \$2.2 billion while the total cost at opt, med and pess scenarios was 10.1%, 18% and 33.7 %, respectively higher than the cost at the clean condition.

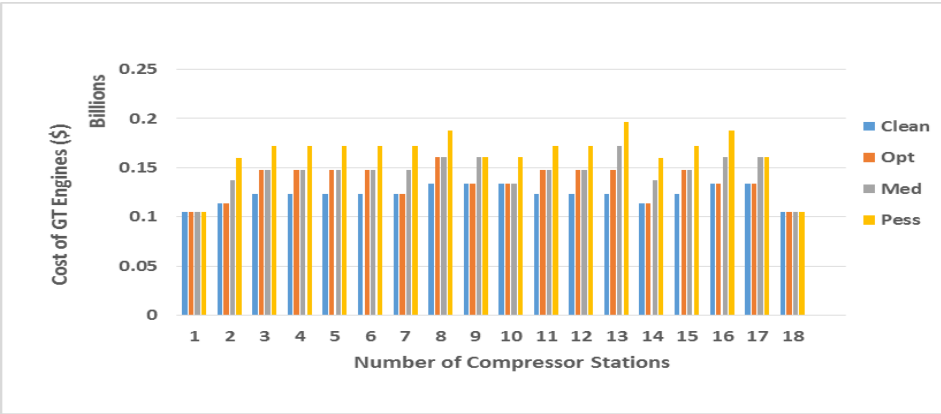


Figure 6-5 Cost of GT engines used for each compressor station

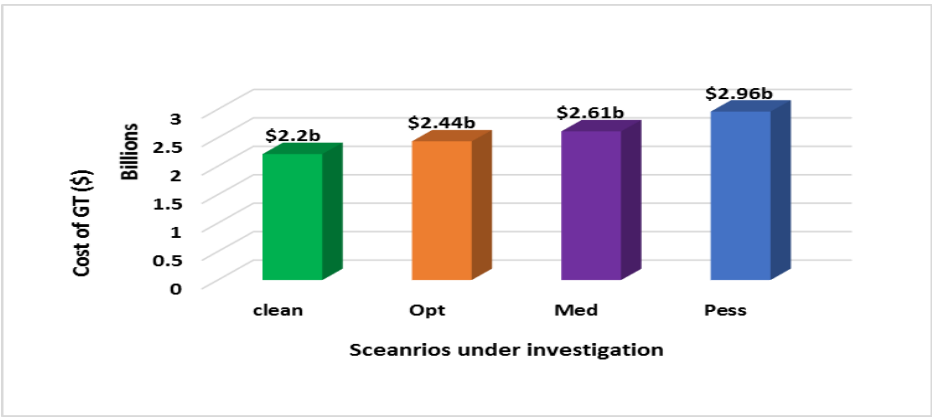


Figure 6-6 Capital cost of GT engines at various scenarios for 18 CS

Figure 6-7 and **Error! Reference source not found.** represent the cost for gas Compressor. The little variation in cost at different compressor station is as a result of different types of GC and their respective power rating cost as represented in Table F- 1 of Appendix F. For instance, station 1, 2, 14 and 18 used the same GC. Station 3, 4, 5, 6, 7, 8, 11, 12, 13 and 15 have the same GC while station 9, 10, 16 and 17 used the same class of GC. The different power rating of the GC was used based on the calculated power of the gas compressor in each CS. Furthermore, since the number of GCs was the same for all compressor station, the cost at different scenarios remain the same to a tune of \$3.195 billion, as shown in **Error! Reference source not found.**. The cost remains the same for all scenarios because there was no degradation considered in the GC. This is the reason why both the clean, optimistic, medium and pessimistic cost remain unchanged.

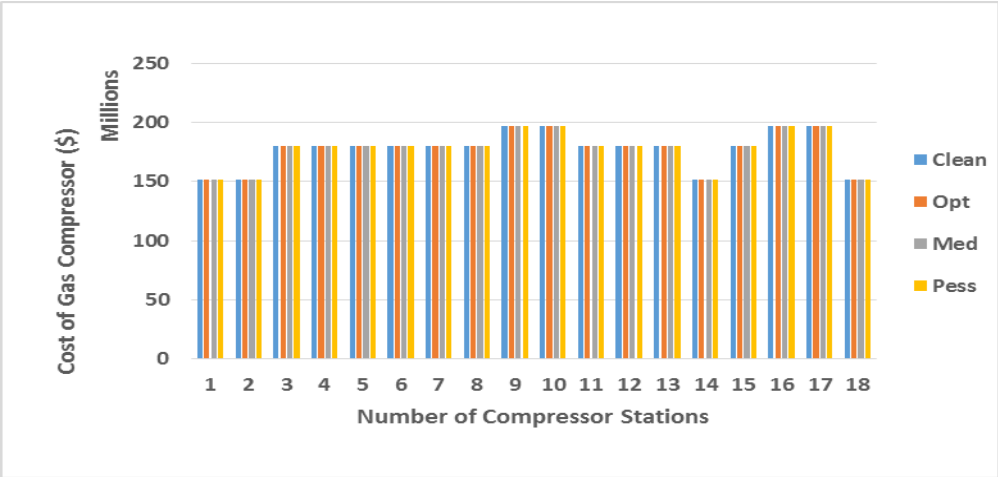


Figure 6-7 Capital cost of GC in each compressor station

The total cost of GC was \$3.195 billion and it was independent of the level of degradation of the GT. It was assumed that the GC was clean throughout the project life.

Figure 6-8 shows the cost of various pipeline accessories used for the natural gas pipeline transportation. It should be recalled that the pipeline accessories depends on the PIPC, which involves the sum of GT, GC and pipeline cost.

It was observed that ECM has the highest cost, followed by E&P and thirdly CT while SCADA and MLV have the list cost when compared with even ROW. The result for contingency (CT) shows that the clean condition has \$0.98 billion. The further result

shows that the cost for the optimistic, medium and pessimistic scenario was 2.5%, 4.44% and 8.3%, respectively higher when compared with the cost of the clean condition. The cost variation in contingency, as shown in Figure 6-8 happened because of the influence of the different cost of GT engines due to degradation on the pipeline initial project cost (PIPC). It is worth noting that the influence of PIPC affects only contingency while the cost of all other components remains the same. The result here serves as a guide to pipeline investors to know the associated cost with respect to the various pipeline accessories. Figure 6-9 represents all the components that constitute the total CAPEX. It should be noted here that the auxiliary cost consists of the cost of SCADA, ROW, MLV and E&P. The result shows that the 4,180km pipeline of 56" diameter has the highest cost, followed by the cost of GC while the contingency (CT) has the least cost in all the scenarios. All the cost are fixed on the curve. Figure 6-10 expressed the total CAPEX for the pipeline project at different Scenarios under investigation. It was seen that the total CAPEX for the clean condition is \$14,873,409,029 while the opt, med and pess scenarios are 1.8%, 3.21% and 6% higher than the clean condition when compared with the clean condition.

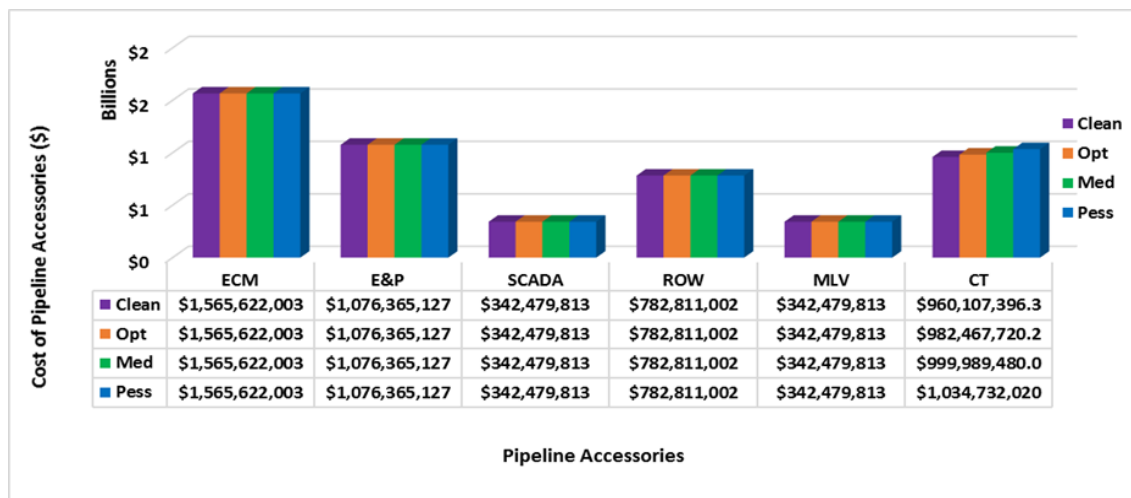


Figure 6-8 Cost of various pipeline accessories

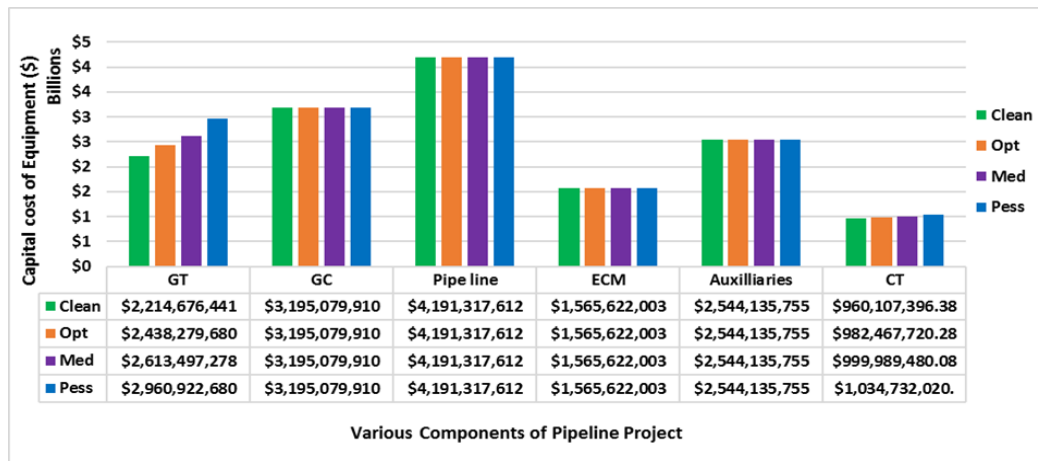


Figure 6-9 Capital cost of all pipeline components for 18 CS

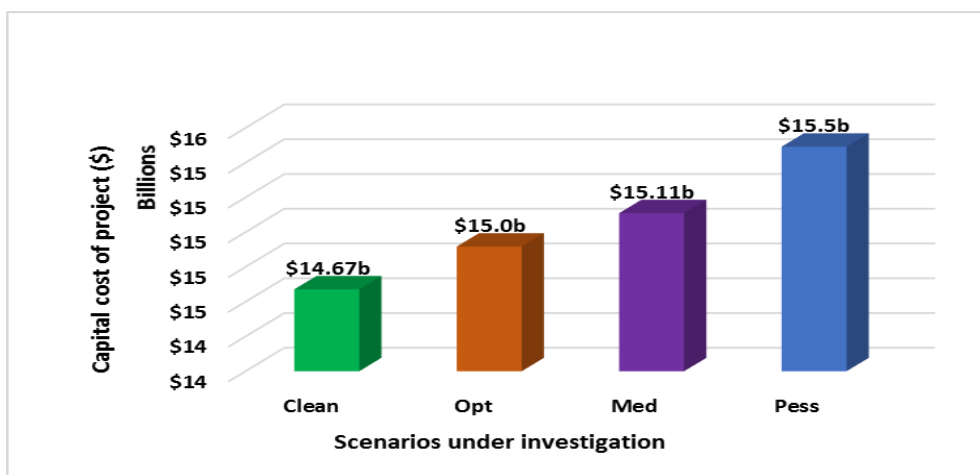


Figure 6-10 Total CAPEX for the pipeline project-Baseline case

6.4.2 Impact of GT Degradation on Fuel Cost

Since the shaft power of GT was fixed with the GC power during operation, the TET increases with a corresponding increase in the fuel flow in order to maintain the engine power for all the investigated scenarios. The increase in the fuel flow, especially at the degraded condition of the GT directly affects the fuel cost. The fuel cost increases as the year of engine operation increases. The result for the time-based fuel consumption showed that the engine (15.2MW) operation between a degradation value of 0.25% and 0.5% (reduction of engine flow capacity and isentropic efficiency of both compressor, compressor turbine and power turbine) within an average temperature of 17.12°C and 28.92°C results to a fuel cost of \$574.1 per hour.

It was generally observed that the time-based fuel cost for the various scenarios in the hot season was higher than the dry season and the dry season higher than the rainy season. The reason was that the ambient temperature and degradation increases as it progresses from rainy season to hot seasons, which implies that, both the time-based ambient temperature and degradation values for the hot season was higher than the dry seasons and the dry season higher than the rainy season. Comparing the fuel cost for the dry and rainy season together for CS 2, the variation in the fuel cost between the rainy and dry season for the optimistic scenario was within 0.35% to 1.16% depending on the time of day. That of the medium scenario was between 0.37% to 1.46% increase while the pessimistic scenario was 0.4% to 2.4% higher, all depends on the time of day when compared between rainy and dry season. This percentage increase showed how the operation of the GT engine in dry season increases in cost when compared with the rainy season. Similarly, the cost of fuel for the hot season was higher than all the seasons. The reason is that the hot season has the highest temperature and degradation among all the seasons of the year. It is important to state that the three-hourly segmented timed-based fuel cost for each scenario was added together to achieve the fuel cost for each season according to their respective scenarios as presented below.

6.4.3 Season's-based Fuel Cost

Figure 6-11, Figure 6-12 and Figure 6-13 indicate the fuel cost for winter, dry and hot season for compressor station 2. Each of these seasons has four scenarios (clean, opt, med and pess) as earlier explained. An overhaul was implemented after 3 years interval for 30 years of operation at baseline condition. Because of overhaul implementation, the cost of fuel at year 4, 7, 10, 13, 16, 19, 22, 25 and 28 assumed almost the cost of fuel at the clean condition.

The result showed that running the engine on degraded mode increases the fuel cost. For instance, The second year in the hot season of compressor station 2, the total fuel cost for the clean engine operating between an ambient temperature of 21.45°C and 31.95°C above the ISA condition was \$5.68 million. However, operating the engine at degraded scenarios with 1.5%, 2.5% and 3.5% for opt, med, and pess, respectively (reduction in flow capacity and isentropic efficiency of the compressors and turbines) within the same ambient temperature had an increase of 3.4%, 5.8%

and 10% for opt, med, and pess, respectively when compared with the clean engine operation. Further observation on the result showed that the progressive increase in fuel cost in each of the scenario after each maintenance cycle indicates the influence of unrecovered degradation on the fuel cost. Generally, the cost of fuel at the pessimistic condition was higher than all the scenarios. The reason was because of the higher level of degradation in addition to the higher number of engine usage at this scenario. Since the fuel flow is cumulative, the fuel flow goes higher at the degraded condition in order to maintain the fixed power of the GT. The same approach was applied to all the seasons of the year and in all the CS. The result also varies from one compressor station to another. It is worth noting that the fuel cost calculation, which varies from one year to another, was done with zero percent escalation rate (no inflation on fuel cost). It is also important to state that the fuel cost for the different seasons with respect to each scenario was added together to achieve the annual fuel cost as presented Figure 6-11 to Figure 6-13 for CS 2.

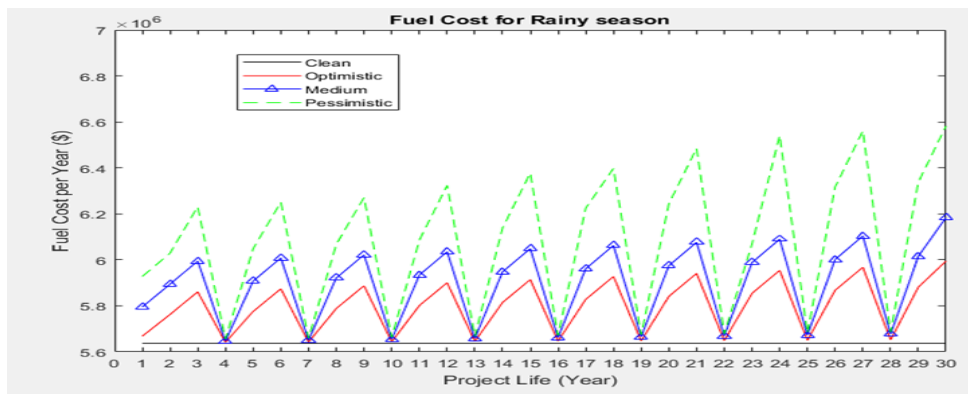


Figure 6-11 Fuel cost for the rainy season

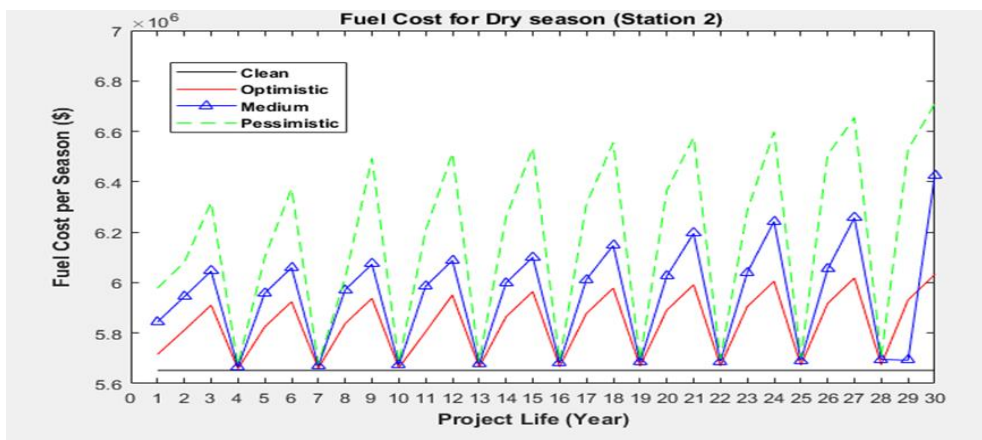


Figure 6-12 Fuel cost for dry season

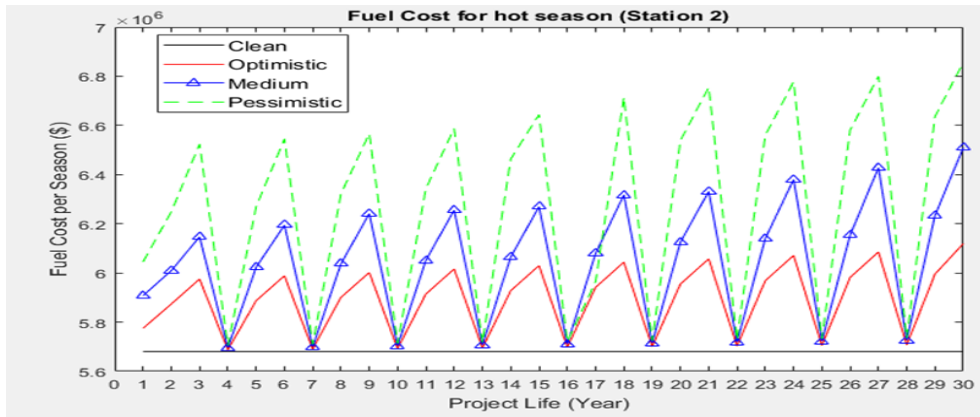


Figure 6-13 Fuel cost for the hot season

Figure 6-14 showed the annual fuel cost for all the 18 CS at 0% inflation rate. The addition of the fuel cost of the three seasons of the entire 18 CS resulted in the annual fuel costs for thirty years of operation. From the result, fuel cost for the clean condition maintained an approximate annual cost of \$0.423.8 billion throughout the life of the project. This is because the engine was assumed clean throughout the operation. The result also shows that the fuel cost for the degraded scenarios was higher as the degradation increases. This implies that the fuel consumption of the engine at degraded condition was high to maintain the power of the engine. Its Worthing noting also that the fuel cost reduces at the year of engine overhaul (Year 4, 7, 10, 16, 19, 22, 25 and 28).



Figure 6-14 Annual fuel cost at different scenarios

6.4.4 Fuel Cost at each Compressor Station

Figure 6-15 to Figure 6-17 shows the cost of fuel at each compressor station in each season of the year for thirty years of project life and with respect to the different scenarios under investigation. The aim of analysing each compressor station was to ascertain the fuel cost that can be budgeted to the various compressor station since each compressor station has different ambient temperature and altitude. Also, the number of GT engine used in each CS varies from each other. The result will also help the operators to budget appropriately to each compressor station.

Generally, the result for all the seasons shows that compressor station 10, 16 and 17 has the highest fuel cost when compared with the rest of the CS. The reason was that this CS has a lower number of GT that operate on a higher TET except for the pessimistic scenario that has a higher number of GT. Having a lower number of GT engines operating at these CS implies operating the engine at a higher TET and this result in higher fuel consumption. It should be recalled that as the number of GT engine increases the lower the operational TET of that engine, vice versa and this affects the fuel consumption, hence the reason of having higher fuel cost at that CS. Also, it was observed that the fuel cost of the clean scenario of CS 10, 16 and 17 was higher than other scenarios, this is because the clean engine at this station operates on higher TET and the fuel consumption at other scenarios does not outweigh the clean condition though at the expense of the engine capital cost. Compressor station 1 and 18 has the lowest fuel cost compared to other CS. The reason was that these two CS used an equal number of GT throughout the life of the project. The slight differences in fuel cost experienced in CS 1 and 18 were as a result of the different level of degradation in each scenario. So, as degradation increases, fuel cost increases also. This happened because the shaft power of the engines was fixed.

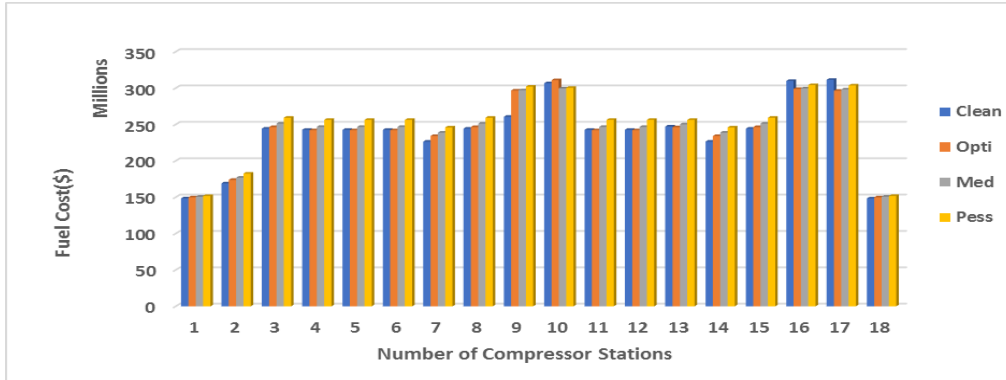


Figure 6-15 Fuel cost for each CS in rainy season for 30 years

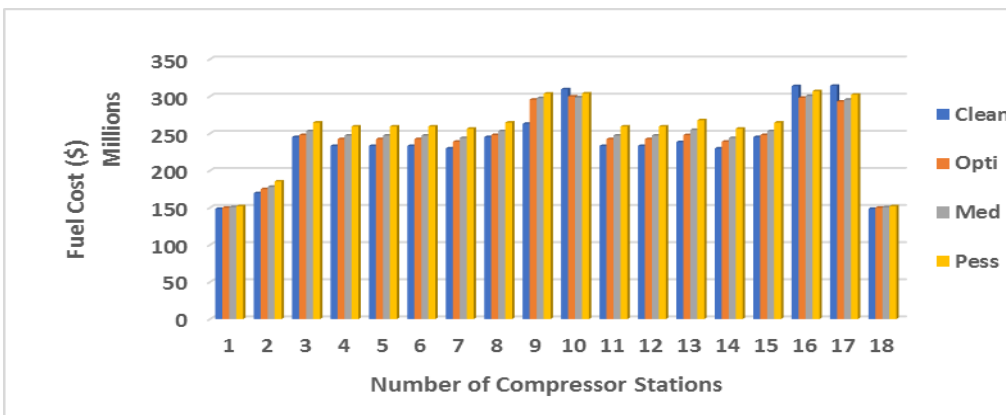


Figure 6-16 Fuel cost for each CS in the dry season for 30 years

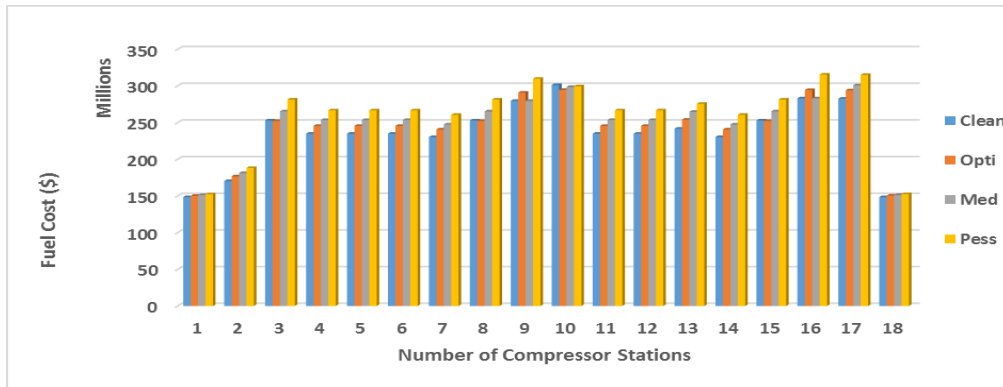


Figure 6-17 Fuel cost for each CS in the hot season for 30 years

There was a need to achieve the holistic value of the fuel cost per annum and for the entire project life having evaluated the fuel cost for different seasons of the year. Figure 6-18 and Figure 6-19 shows the total fuel cost for each compressor station and the total fuel cost for the project life, respectively. The result for the total fuel cost for each CS was achieved by adding the fuel cost of each season together with

respect to each scenario. The fuel costs in each of these stations are the 30 years operational fuel cost. Finally, Figure 6-19 represents the integrated fuel cost for the project life at different scenarios. This was achieved from the sum of the individual fuel cost at the various CS based on the scenarios under investigation. The result shows that the clean engine operation has a total fuel cost of \$12.7billion. The optimistic, medium and pessimistic scenario has an increase of 3.1%, 4.7%, and 8.7 % respectively when compared with the clean condition. These clearly show that engine degradation enhances fuel consumptions at fixed power of the engine operation.

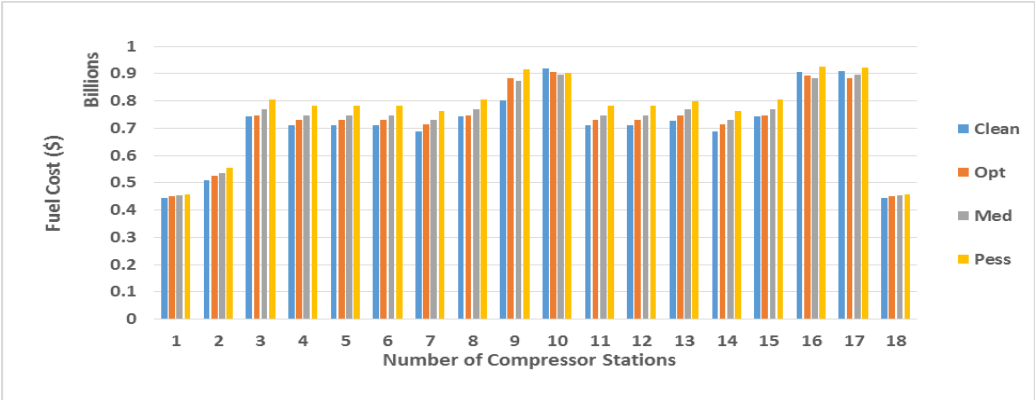


Figure 6-18 Total fuel cost for each CS in 30 years

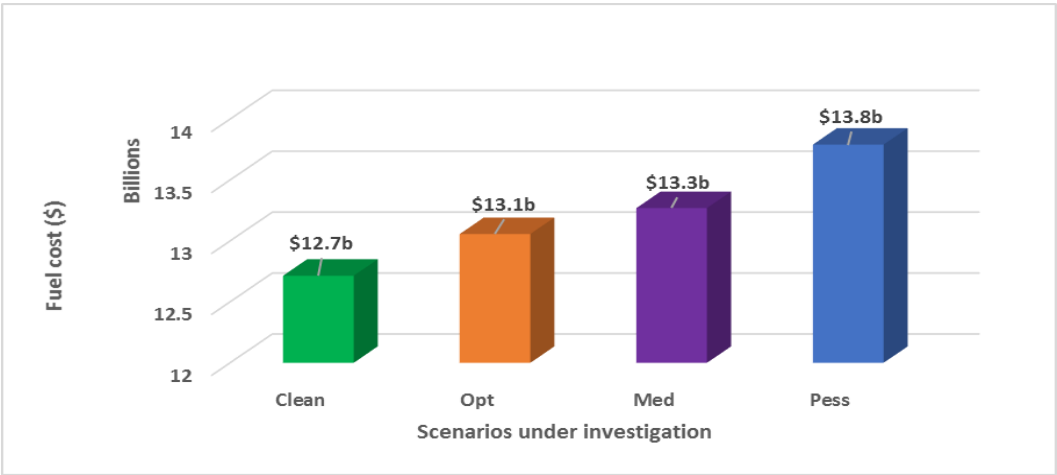


Figure 6-19 Integrated fuel cost for each scenario

6.4.5 Creep Life Estimation

Consistent operation of GT engines at a high temperature can cause high stresses that can have an adverse effect on the mechanical integrity of the turbine blades and other components. This will have a significant effect on the engine maintenance cost

due to high-pressure turbine blades replacement. Hence, the reason for analysing TET-based maintenance cost. This implies that the more the engine is prone to high temperature, the higher the cost of maintenance.

Based on the method adopted in estimating the life of the engine, Figure 6-20, Figure 6-21 and Figure 6-22 showed the estimated creep life for the rainy, dry and hot season, respectively for year 1 of CS 2. The result shows that at a particular rate of degradation corresponding to a particular ambient temperature, the creep life reduces with increasing TET. The result also shows that as engine degradation increases, the creep life reduces. This is reflected in the curves when the creep life for the pessimistic scenario was lower than all the scenario. The clean engine operates at baseload life of 25000 hours. Meanwhile, different operational engine life was obtained at different compressor stations with respect to the different level of degradation in each year of operation. However, the non-linearity in the curves shows the influence of the time-based ambient temperature.

The result presented here was when the number of GT was fixed. However, the developed simplified lifing model was able to deduce the creep life when the number of the GT increases in each season of the year in a time-based system. The alternating creep life values due to increase and decrease number of GT (effect of degradation) was connected to the economic model to calculate the O&M cost.

The result for the creep life when the number of the engines was fixed and varied was obtained for the entire project life at the different scenarios under investigations for all the 18 CS. The creep life result was used to evaluate the engine maintenance cost

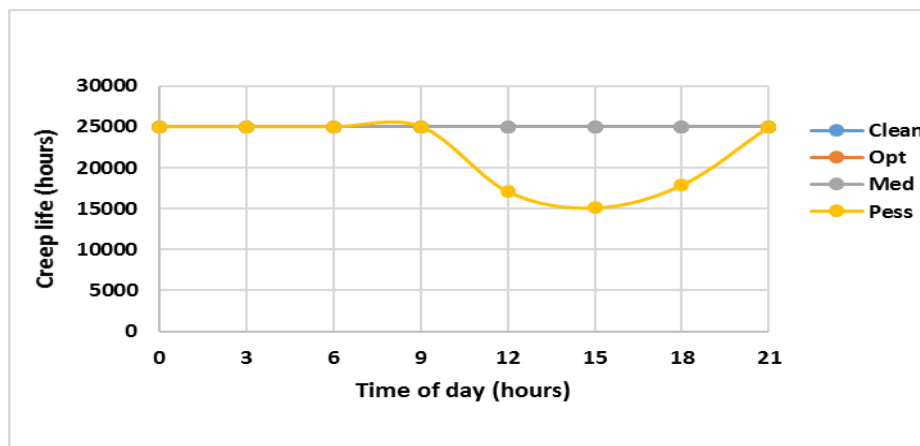


Figure 6-20 Effects of degradation on the engine creep life (rainy season)

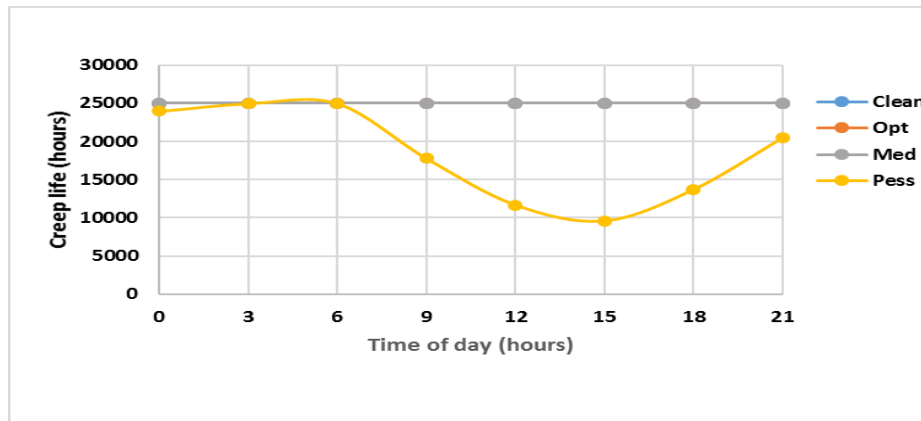


Figure 6-21 Effects of degradation on engine creep life (dry season)

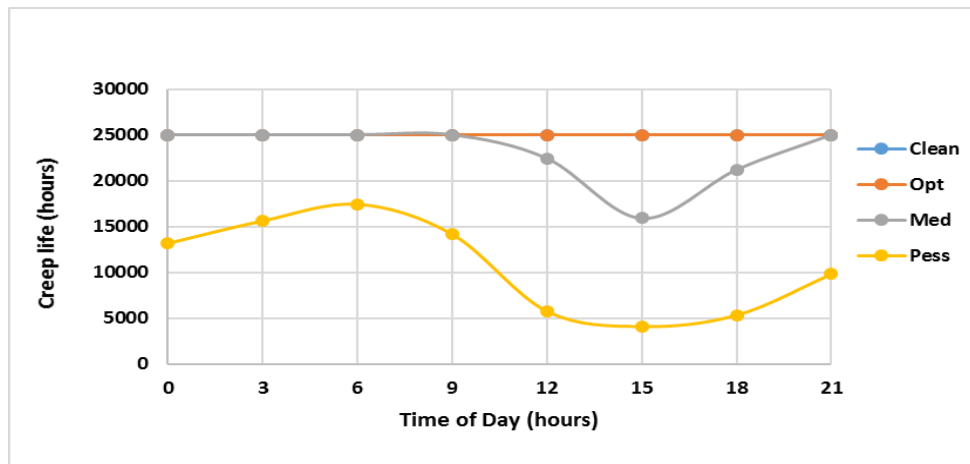


Figure 6-22 Effects of degradation on engine creep life (hot season)

6.4.6 Maintenance Cost

Operation and maintenance (O&M) cost is very important in determining the life cycle cost of this project. The major maintenance cost of the engine was hinged on the TET as the power of the gas turbine was fixed throughout the operation. Generally, as the TET goes high, the engine life reduces, which requires higher maintenance cost. This analysis was applied to all scenarios and in all seasons throughout the project life.

6.4.7 Influence of Degradation on Different Component of O&M Cost

In this research, The O&M cost of the GT engine was determined by the fixed, variable and major maintenance (TET-based maintenance). (Figure 6-23) and Figure

6-24) shows the variable and major maintenance (TET-based) costs. The addition of these three costs effectively gives the annual O&M cost of the engine for the project life with respect to the various scenarios. From the results, it was noticed that there were variations in the variable and major maintenance. These occurred as a result of engine investment and divestment due to degradation and overhauls, respectively with respect to different scenarios under investigation at each CS. The annual costs for the various components were done with respect to each season of the year as presented below

6.4.7.1 Components of O&M Costs for Winter Season

The result for the fixed cost was the same in each CS. However, the result for the variable cost as depicted in Figure 6-23 shows that the costs were time-based. for variable

Based on the time-based ambient temperature and degradation, the number of engines used at each CS varies due to TET constrain, and as the maintenance goes towards the pessimistic scenario (highest level of degradation), the maintenance cost tends to reduce. This is because as the number of engine increases, the power reduces, and this power is one of the major factors in the variable cost calculation. Based on this assertion, it was further observed that the maintenance cost at clean condition was higher than other scenarios when the number of the engine increases with time. The reduction in maintenance costs at engine degraded condition was at the expense of the high engine capital cost at this condition.

Figure 6-24 shows the major maintenance cost (TET-based maintenance). The engine degradation was high in the pessimistic scenario when considering the maintenance cost for each scenario. General observation shows that the life of the engines reduces as degradation increases and the reduction in engine life gives a corresponding increase in maintenance cost. From Figure 6-24, the result shows that the pessimistic scenario had the highest cost until year 19. The reason was that all the scenarios used an equal number of GT and the increase in the cost was as a result of the influence in the level of degradation in addition to ambient temperature effect. However, as the operation continues the number of engine increases at year 20 to 29 where the pessimistic scenario had an additional GT engine to be 6 GTs. The creep life at this year of the pessimistic scenario reduces, thereby affecting the

maintenance cost. This is the reason why at year 20 to year 29, the maintenance costs for the medium scenario was higher than that of the pessimistic scenario. The reason was that the TET of the engine while operating with 6GTs was lower when compared with the 5GTs. Increasing the number of GT at fixed power settings increases the creep life of the engine because of part-load operation, making the maintenance cost at the pessimistic scenario to be lower than the medium scenario. After that, pessimistic costs at year 30 become higher than the medium because, at year 30, both medium and pessimistic scenario used 6 GTs and the degradation effect influences the cost of pessimistic scenario to go higher than the medium scenario.

The up and down of maintenance costs in all the scenarios were based on the fact that, if the TET reduces or increases due to the impacts of degradation as a result of the number of engine usage, so also the creep life of the engine increases or reduces, respectively. Also, as the number of engines increases, the engine power settings reduces, and this power plays a significant role in the maintenance cost calculation. This is because the developed model has the capability of selecting the corresponding power of the engine as the number of engines increases or reduces with time. This evaluation was done in all the 18 CS for the three seasons of the year and with respect to all the scenarios under investigation.

Summarily, the maintenance cost reduces as degradation increases with an increasing number of GT because ; (i) The operational TET reduces as the number of engine increases, vis-à-vis. (ii) The operational life of the engines increases as the number of the engine increase, vis-à-vis. (iii) As the number of engines increases, the power reduces and the engine shaft power plays a significant role in the maintenance cost calculation. Also, the reduction in maintenance costs at engine degraded condition was at the expense of the high engine capital cost which was felt when evaluating the benefits on investment (NPV).

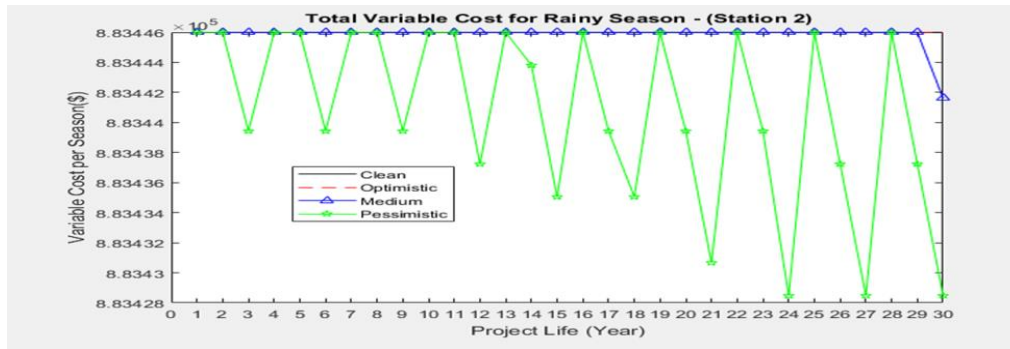


Figure 6-23 Variable costs for the rainy season

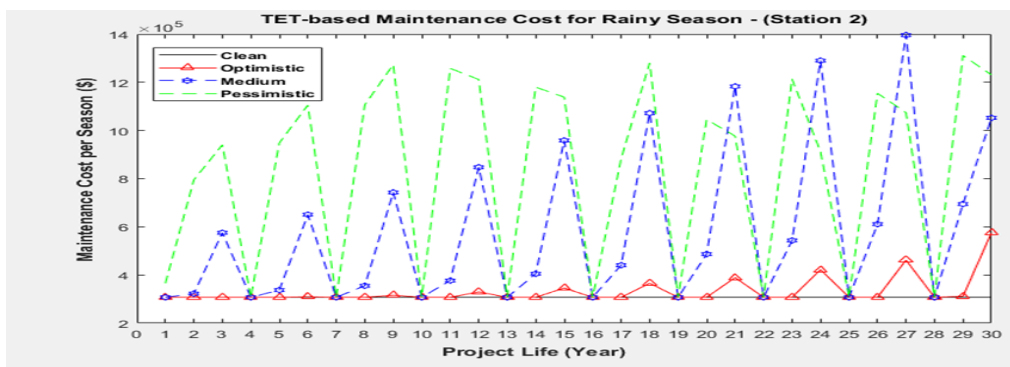


Figure 6-24 Major maintenance cost for the rainy season

The addition of the fixed, variable and major maintenance cost, as shown above, gives the O&M cost for the rainy season as represented in Figure 6-25. Meanwhile, the cost component for the dry and hot season is showed in Figure G-1 to G-6 of Appendix G-1 to G-6. The addition of the three components gives the O&M cost for each season as represented in Figure 6-26 and Figure 6-27, respectively, for the dry and hot season. The same reasons for the variation in cost, as explained above, were applicable in this result.



Figure 6-25 O&M costs for the rainy season

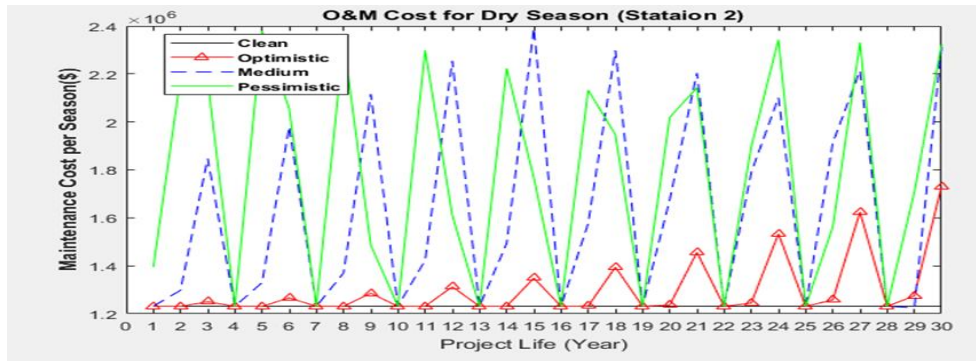


Figure 6-26 Annual O&M cost for dry season

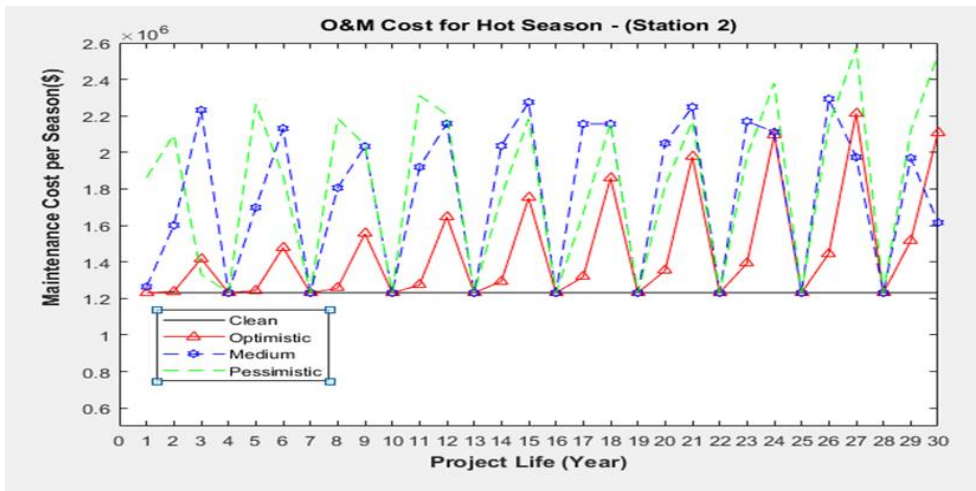


Figure 6-27 Annual O&M costs for the hot season

The above result was for CS 2. The same analysis was applied to all the 18CS, and the result varies from one CS to another. It is worthy of note that the addition of the three seasons with respect to the various scenarios for each CS resulted in the annual O&M costs for 30 years of operation.

Figure 6-28 showed the annual O&M cost when all the 18 CS was calculated. From the result, O&M cost for the clean condition remains constant with an approximate annual cost of \$0.11b throughout the life of the project. This is because the engine was assumed clean throughout the operation. The result also shows that the O&M cost for the clean condition was higher than the optimistic scenario in all the years. Also, at some years of operation, as represented on the curve, the maintenance costs for the clean was still higher than the medium. The reason was that the engine at the clean condition for those years was operating on a higher TET than the degraded scenarios. This makes the creep life of the engine to be low, thereby

influencing on the maintenance cost. The degraded (opt, med and pess) alternate in their O&M cost due to reducing and increasing number of GT before and after overhaul. This makes the operational engine life to fluctuate, thus increasing and reducing the O&M cost.

Although the degraded scenarios operate with higher creep life because of the higher number of GTs used during the operation, yet the maintenance cost was still higher. Considering the pessimistic scenario going higher in all the year except the year of engine overhaul (Year 4, 7, 10, 16, 19, 22, 25 and 28) was due to the influence of higher number of GT engines invested at those years in order not to exceed the TET limit during operation and the impacts of higher level of degradation. The same reason also affected the medium scenario where degradation and the higher number of engines also influenced the O&M costs at year 6, 9, 12, 15, 18, 21, 24, 27, and 30. This result considers all the total number of engines used for the 18 CS throughout the project life.

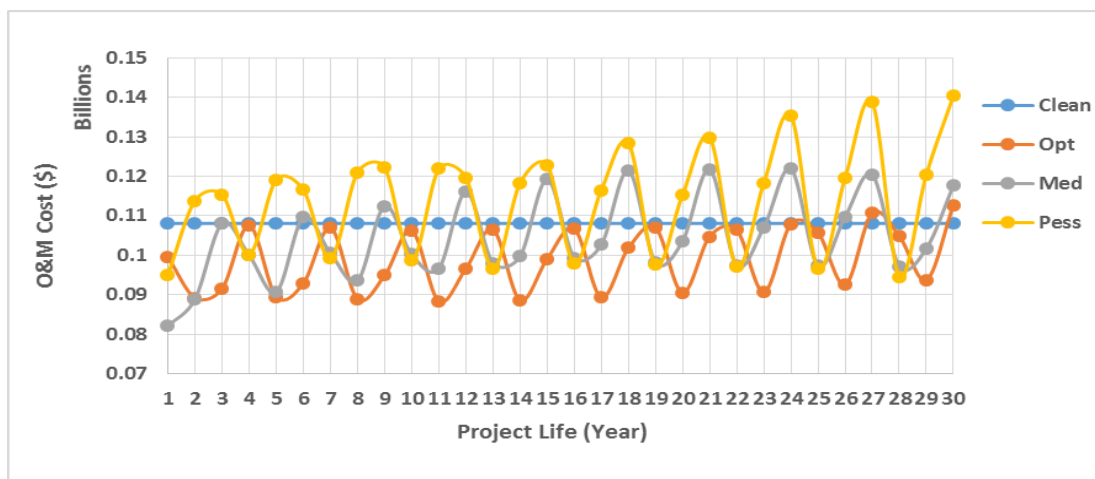


Figure 6-28 Annual O&M cost for the project's life-Baseline Case

6.4.8 Maintenance Costs of Each Compressor Station

Considering an increasingly-stringent financial budget for natural gas pipeline investors to quantify capital intensive project and the factors that influence such a project, make it more imperative to evaluate the maintenance costs of each compressor station. This is necessary because the number of engines used in each station and the ambient temperatures varies from one CS to another. Figure 6-29,

Figure 6-30) and Figure 6-31) shows the 30 years O&M costs for the individual CS for winter, dry and hot season respectively

The result in CS 1 and 18 shows that the O&M costs increases with a corresponding increase in engine degradation, this makes the O&M costs for the pessimistic scenario higher than the medium and medium higher than the optimistic while the clean scenario has the least O&M cost. However, increasing the number of GT enabled the engine to operate on a lower TET, which increases the life of the engine, thereby reducing the O&M cost. For instance, in the winter season, the O&M costs for a clean scenario for compressor station 3, 4, 5, 6, 8, 9, 11, 12, 13, 15, 16, and 17 was found higher than all the scenarios. This was because, the engine was operating on a higher TET with a lower number of GTs, while the degraded scenarios operate on a lower TET with a higher number of GTs. This implies that, as the TET increases, the maintenance cost reduces. However, the reduction in maintenance costs at the degraded condition was achieved at the expense of the engine higher capital cost at that scenario. So, the variation in O&M costs at the various CS was due to the different numbers of GT used at the different CS and the influence of the operational TET in evaluating the O&M cost. This is the reason why some CS with respect to the scenario under investigation has that the O&M cost for the clean scenarios becoming higher than other scenarios. This conditions of evaluating the O&M costs for each CS was applied to all the seasons. Further evaluation revealed that as the CS experienced higher ambient temperature with a corresponding increase in degradation, the O&M costs were high, this was found in the degraded scenario in hot season (Figure 6-27) having higher O&M cost in some of the CS that the clean scenario was high in the winter season (Station 4, 5, 6, etc). In this case, the clean condition becoming lower than the degraded condition. This clearly shows how elevated ambient temperature and degradation can affect the maintenance cost.

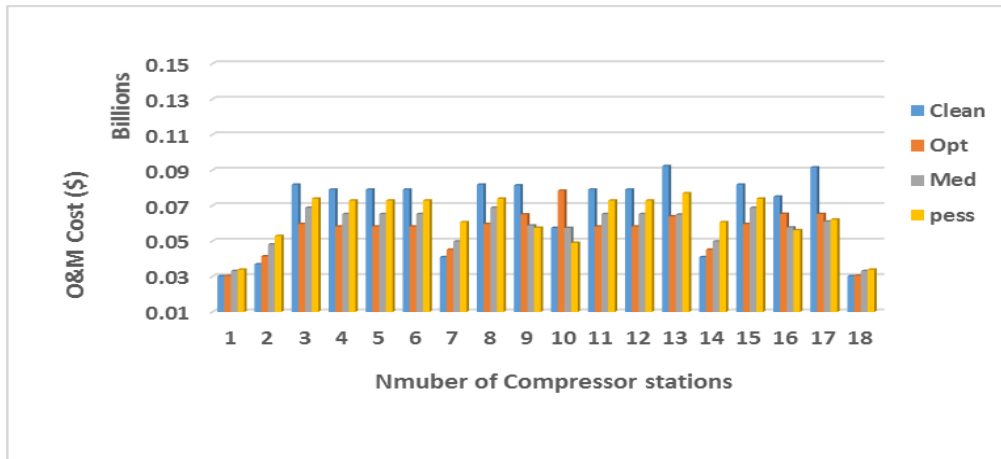


Figure 6-29 O&M costs for each CS in the rainy season

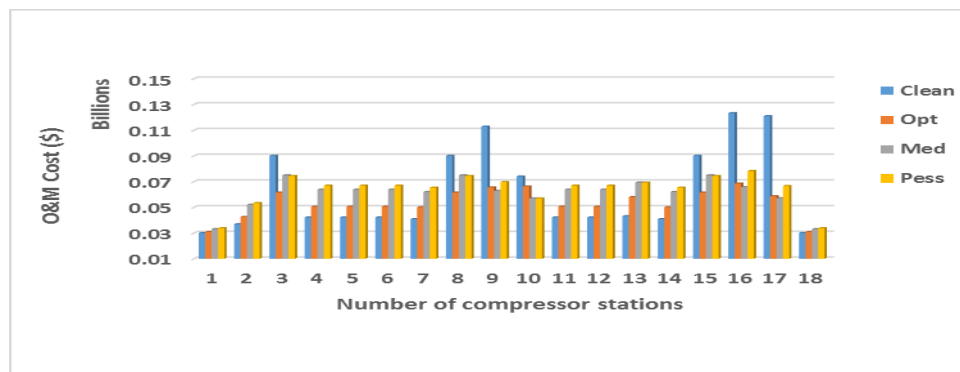


Figure 6-30 O&M costs for each CS in the dry season

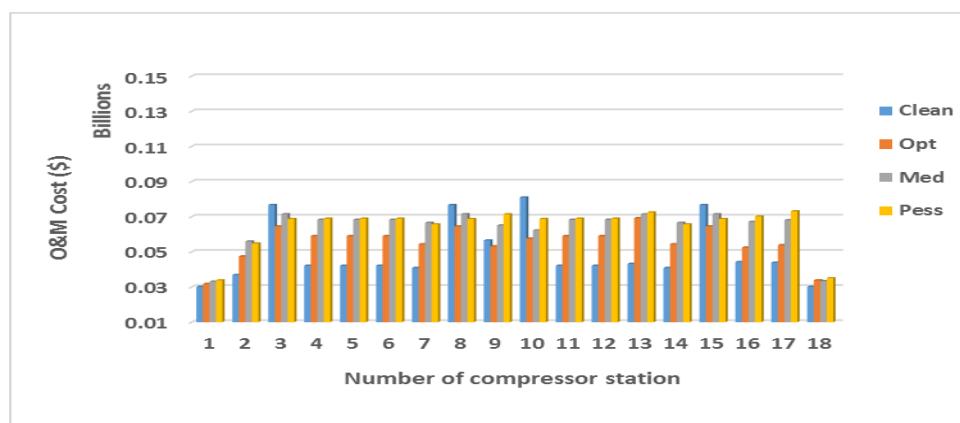


Figure 6-31 O&M costs for each CS in the hot season

Having known the O&M costs for each CS, Figure 6-32 shows the annual O&M cost for each CS with respect to the various scenarios under investigation. The addition of O&M cost of the three seasons of the year with respect to their various scenarios

in each CS resulted in the annual O&M costs for thirty years of operation as depicted in Figure 6-32.

Generally, the result shows that as the engine degradation increases, the TET also increases and this result to higher O&M cost. However, the O&M cost for the clean scenarios of some CS was higher than the degraded scenarios. For instance, Station 3, 8, 9, 10, 15, 16 and 17 have the O&M costs at clean scenarios higher than the degraded scenarios. The reason was that the operational TET of the clean scenario was higher than the degraded scenarios as the degraded scenarios used a higher number of GTs, which makes the engines to operate on a lower TET. This makes the creep life higher, thereby making the O&M cost to be lower.

Furthermore, in compressor station 11 and 12, the optimistic scenario was about 10.2% lower in maintenance cost than the clean scenario. Also, the medium and pessimistic scenarios were higher than the clean scenario. The reason was that, although the number of GTs used at medium and pessimistic scenarios was high which could have made the engine operate at lower TET so that the O&M cost can be lower than the clean condition, the effects of degradation influenced the costs since the number of GT engines used at the medium and pessimistic scenario was not too high than the clean condition. This made the medium and pessimistic scenario incurred an extra cost due to the adverse effect of higher TET as a result of the influence of degradation. Finally, the rest of the station's result shows that the O&M costs increases with a corresponding increase in engine degradation, this makes the O&M costs for the pessimistic scenario higher than the medium and medium higher than the optimistic while the clean scenario has the least O&M cost. Figure 6-33 shows the integrated (total) O&M costs of each scenario for thirty years of operation. The result showed that the clean scenario had a total O&M cost of \$3.4 billion. The result further revealed that running the engines at lower TET and higher creep life (part-load operation) for degraded scenario due to a higher number of GTs reduces the O&M costs by 8.3% and 3.4% for optimistic and medium scenario while the O&M costs for the pessimistic scenarios increases by 5.7% when compared with the cost at the clean condition. This implies that the approach adopted in evaluating the O&M cost for this projects saves \$0.27billion and \$0.19 for the optimistic and medium scenario, respectively when compared with the clean condition. While the pessimistic scenario incurred an extra cost of \$0.19 billion due to the adverse effect

of degradation when running on higher TET. However, the benefits of reducing the maintenance cost when operating degraded engines as seen in optimistic and medium scenarios happened at the expense of an increasing engine capital cost, which directly affects the NPV of the project. It is important to state that this method enables the engine to operate at the desired project life, thereby reducing the engine downtime and increasing the engine availability.

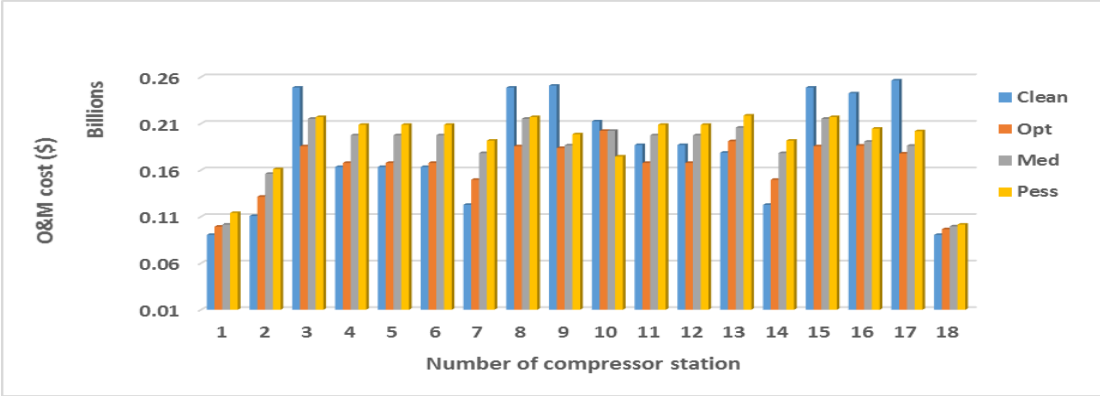


Figure 6-32 O&M costs for each scenario at the various CS for 30 years

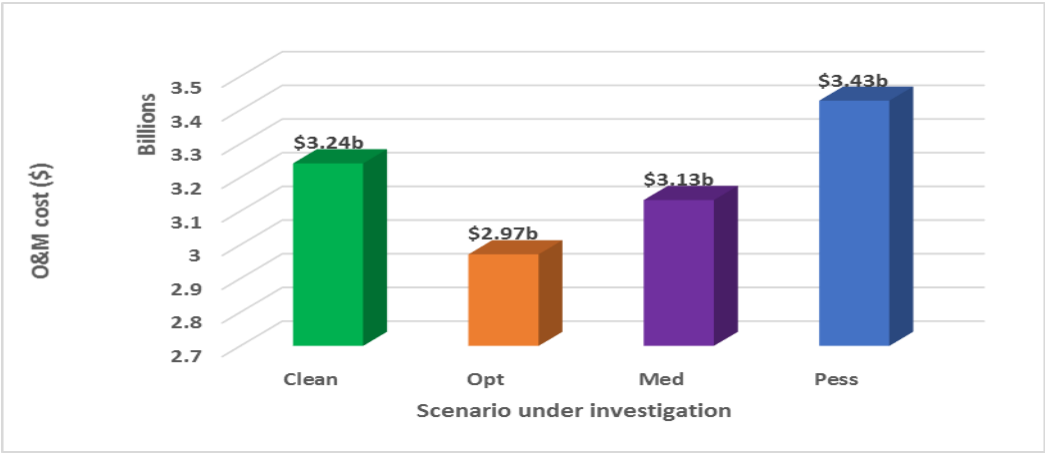


Figure 6-33 Integrated O&M cost for each scenario-Baseline case

Summary, operating the GT engines on higher TET as a result of increasing degradation results to lower life of the engines; on the other hand, lower TET operation results in higher operational engine life. So as the TET goes up, the maintenance cost goes up. vis-à-vis.

6.4.9 Life Cycle Costs (LCC) for the Pipeline Project

The life cycle cost (LCC) of the natural gas pipeline investment is represented in Figure 6-34. All the components used for economic investigation are hereby presented. These components include the total cost of fuel consumed throughout the operation, O&M cost, cost of GTs, cost of GC, cost of the pipeline with all its accessories and the emission tax from the total number of GT engines used for this investigation. The sum of these cost components made up of the LCC. The LCC result was also presented with respect to the various scenarios under investigation.

The life cost result in Figure 6-34 revealed that the fuel component in all the scenarios has the highest cost when compared with all the cost component under investigation. The result also shows that the pipeline, with addition to all the considered accessories (PL/Acess) was the second dominant cost next to the fuel cost. The result further showed that the cost of the gas compressor at clean condition was higher than the GT at the clean condition even up to the medium scenario. However, the additional engines due to degradation make the cost of GT at the pessimistic scenario higher than the GC cost at this same scenario. The O&M cost reduces at the optimistic and medium scenario due to lower TET and lower power settings of the engines while the cost of GTs increases as degradation increases. This is because the number of GT utilised at the degraded scenarios increases due to the engine operational condition. Also, emission tax increases with increasing degradation.

The result of Figure 6-35 shows the percentage breakdown of all the cost component involves in the pipeline project at clean condition only. This shows that the fuel cost has 39.6% of the total life cost while Mcost, GT, GC, PL/AC, and E_Tax has 10.1%, 6.897%, 9.95%, 28.89% and 4.57%, respectively of the total life cost. The percentage comparison for the degraded scenario is integrated into the overall cost calculation.

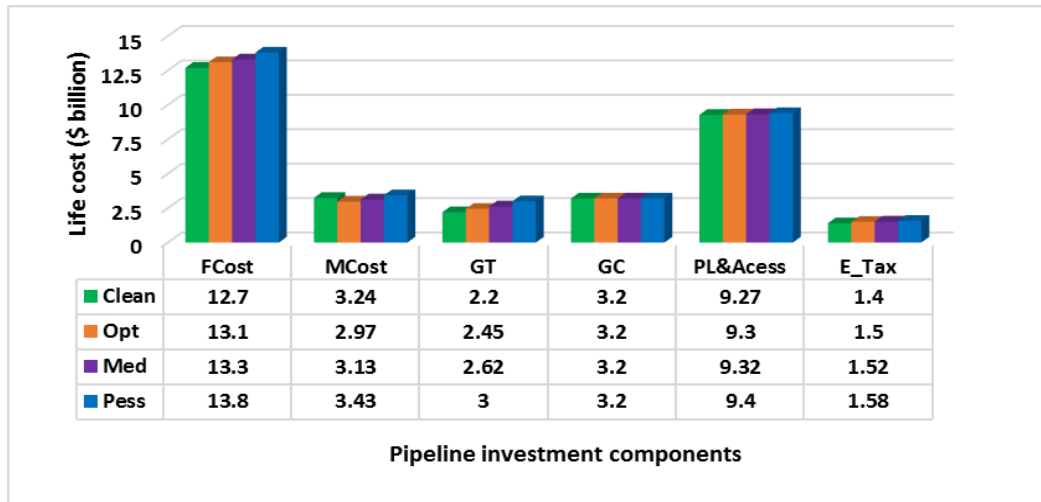


Figure 6-34 Life cycle cost (LCC) for pipeline investment

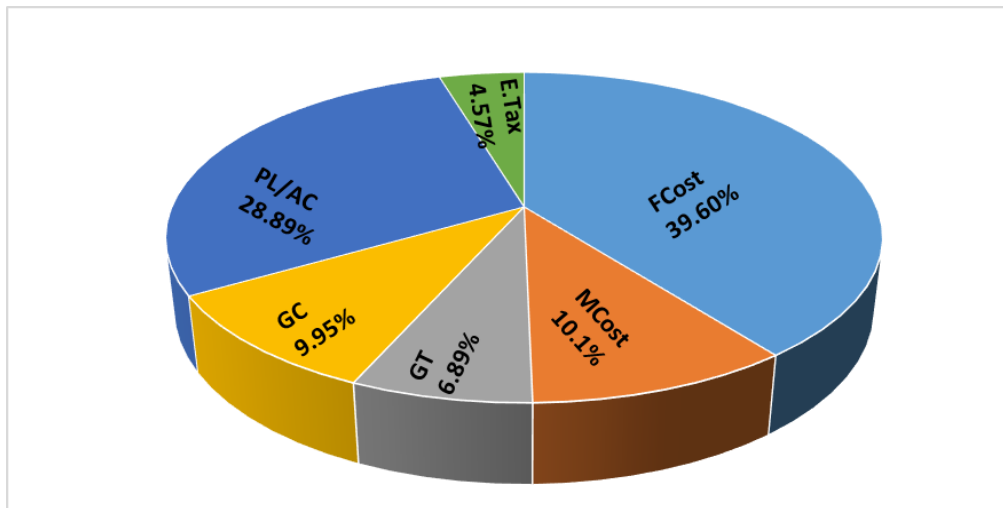


Figure 6-35 Percentage of the baseline project cost at the clean condition

With the consideration of engine degradation as represented in Figure 6-36, the total life-cycle cost for the clean condition was \$32.01 USD billion. The result for the optimistic, medium and pessimistic scenario was 1.59%, 2.94%, and 6.84%, respectively higher than the clean condition. This clearly shows the influence of running the engine on degraded mode on the overall life-cycle cost of the pipeline investment. The result shows that operating the gas turbine at the optimistic, medium and pessimistic scenarios incurred an extra cost of \$0.51, \$1.1 and \$2.4, billion respectively when compared with the clean condition

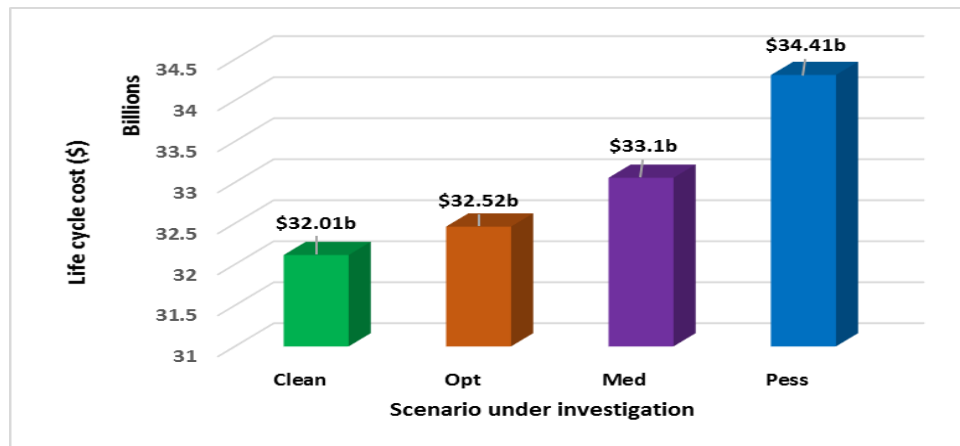


Figure 6-36 Integrated LCC for the pipeline investment

6.4.10 The impact of Escalation Rate of Fuel and O&M cost on LCC

Escalation rate here referred to the annual inflation on the O&M and fuel cost. Figure 6-37 shows the result for the effects of 3% and 5% fuel escalation rate on LCC when compared to the initial 0% escalation rate. The result obtained shows that the annual 3% and 5% escalation rate has an increasing trend as the year progresses. However, it was assumed that there was no inflation in the first year as depicted on the curve. The increase in the annual cost indicates the influence of inflation on the fuel cost except for the year of engine overhaul. This will directly affect the LCC of the project.

Figure 6-38 represents the total cost of fuel that was influenced by inflation. It should be recalled that the total fuel cost for the 0% escalation rate for the clean, opt, med and pess., scenarios were approximately \$12.7, \$13.1, \$13.3 and \$ 13.8 billion, respectively. The result obtained for the total fuel cost at 3% escalation rate at the end of 30 years for each scenario yielded 58.6%, 58.8%, 58.92%, and 59.2% increase, respectively when compared with the cost of 0% escalation rate at the end of the project. The 5% escalation rate was above 100% increase in all the scenarios when compared with the 0% escalation rate at the end of the project.

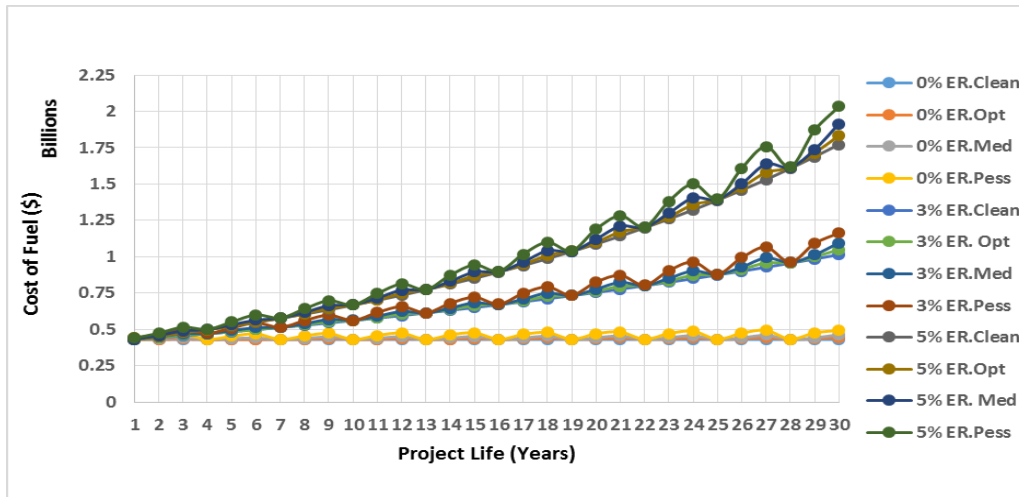


Figure 6-37 Annual escalated fuel cost for project life

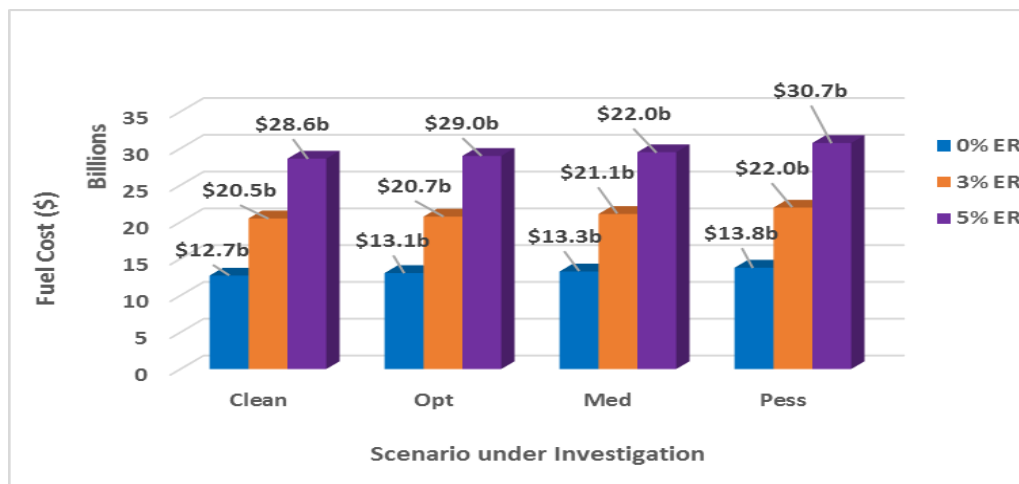


Figure 6-38 Total escalated fuel cost for the life

Due to the impact of inflation on the O&M cost when compared with the 0% escalation rate, the result obtained in Figure 6-39, shows that the O&M cost increases annually beside the effects of engine degradation.

The total O&M cost for the 0% escalation rate for clean, opt, med and pess. scenarios was approximately \$3.24, \$2.97, \$3.13, and \$3.43 billions, respectively. The result obtained for the total O&M cost at 3% escalation rate at the end of 30 years for each scenario yielded 58.5%, 59.8%, 60.3%, and 60.8% respectively when compared with the cost of 0% escalation rate. The 5% escalation of O&M cost was above 100% increase in each scenario when compared with the 0% escalation rate. The result

obtained for both fuel and O&M cost shows that the impacts of inflation will affect the LCC of the project.

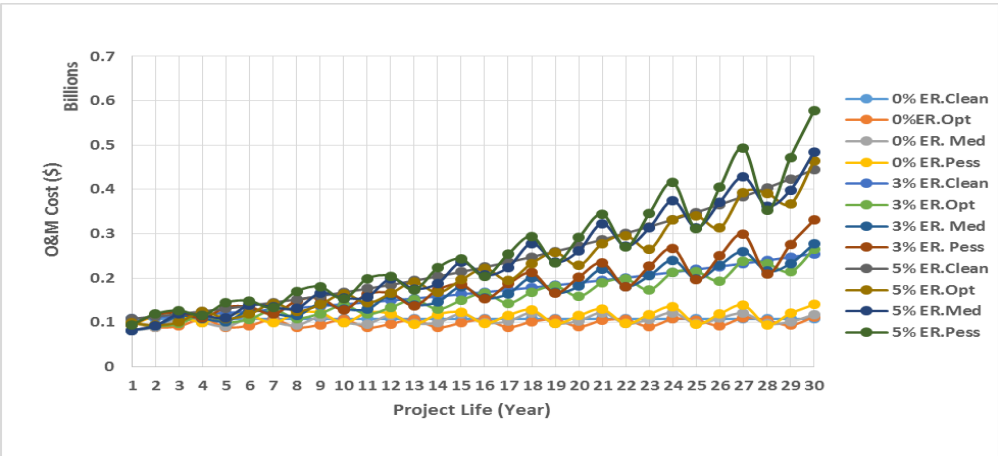


Figure 6-39 Annual escalated O&M cost for the project life

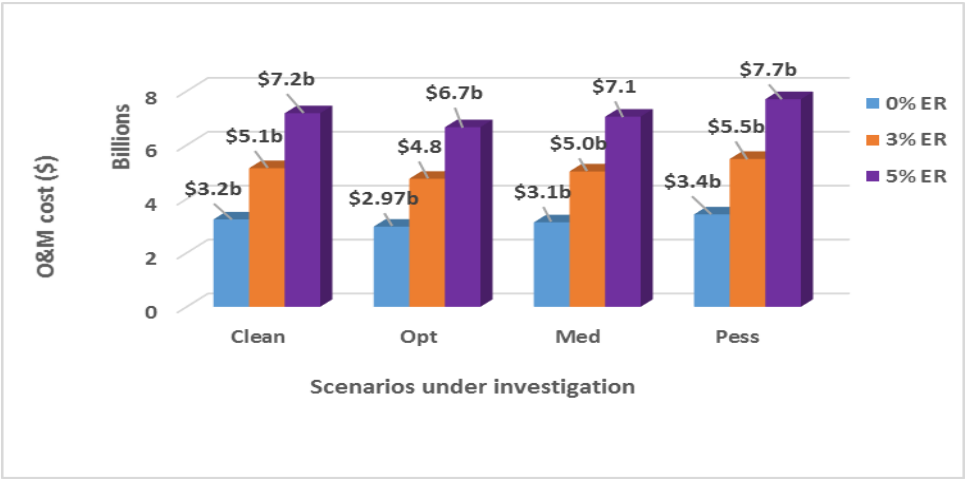


Figure 6-40 Total escalated O&M cost for the project life

Figure 6-41 shows the impacts of fuel and O&M cost escalation rate on the life-cycle cost of the natural gas pipeline under investigation. It should be recalled that the LCC for 0% escalation rate for the clean, opt, med and pess scenarios were approximately \$32.5, \$32.8, \$33.5 and \$34.9 billion respectively as represented in

Table 6-2. However, the result for the impact of 3% and 5% escalation (ER) obtained are clearly shown in the same

Table 6-2. This clearly showed that inflation has an effect on the LCC.

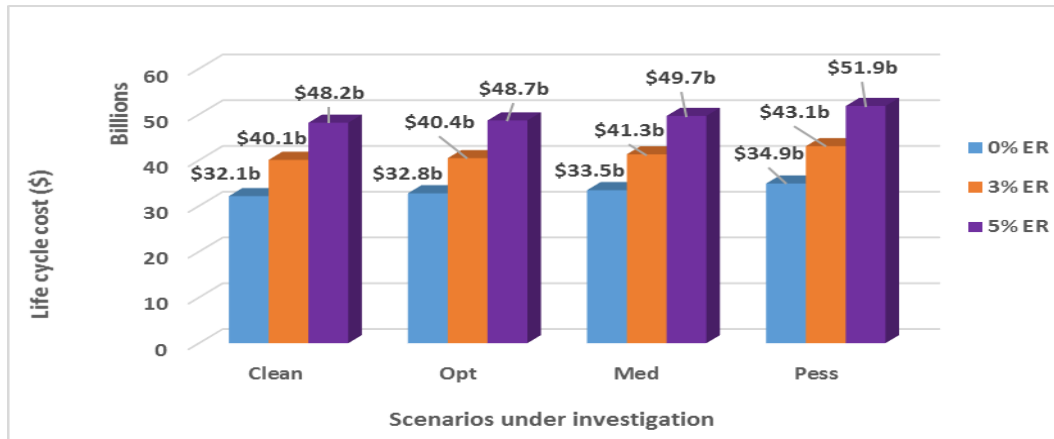


Figure 6-41 Escalation rate on the life-cycle cost of the project.

Table 6-2 Impact of ER on LCC

Scenarios	LCC at 0% ER (\$, billion)	3% ER compared with LCC at 0% ER	5% ER compared with LCC at 0% ER
Clean	32.5	23.29% above 0% ER	48.28% above 0% ER
Opt	32.8	23.43% above 0% ER	48.6% above 0% ER
Med	33.5	23.37% above 0% ER	48.48% above 0% ER
Pess	34.9	23.4% above 0% ER	48.58% above 0% ER

6.5 Evaluation of Net Present Value

Before the calculation of the NPV, relevant variables that aid in the calculation has to be evaluated in addition to the investment costs. These include revenue from the sales of gas, loan repayment schedule, Annual net cash flow, and the divestment sales on the redundant engines after overhaul.

6.5.1 Revenue from the sale of gas

The yearly revenue from the sales of natural gas was the primary source of income in this project. It should be recalled that the project was expected to deliver 30bcm/year. The value in M³ was converted to MMBTU, and 0.168 (\$/kg) was used as gas price. The gas price was also converted to MMBTU. This was done to meet the international measuring standard for the sale of gas. The revenue was evaluated using the relationship shown in

$$AR = GFP (\$/kg) * EVD (m^3/kg) \quad (6-12)$$

Where, AR= Annual revenue; GFP= Gas fuel price; EVD= Expected volume delivery.

6.5.2 Loan Repayment

The repayment schedule for the loan used for the project was estimated using Equation 6-13.

$$L_{RS} = \frac{\text{Project CAPEX} * \text{Interest Rate}}{1 - (1 + \text{Interest Rate})^{-LD}} \quad 6-13$$

Where,

L_{RS} = Loan repayment schedule

LD = Duration of loan

The following assumptions were made during the loan repayment period. Interest rate 3% [2], loan duration 15 years, and repayment holiday for 2 years. Figure 6-42 shows the 3% annual loan repayment plan on the project. The result shows that all the investment cost were settled between year 3 and 17. Meanwhile, there was no loan repayment in year 1 and 2 (repayment holiday) while year 17 to 30, loan free year.

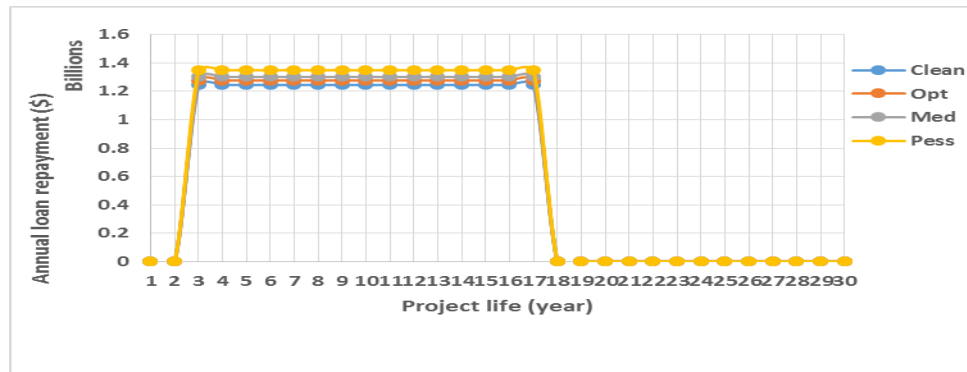


Figure 6-42 Annual loan repayment on project

6.5.3 Annual Cash Flow of the entire Project

Based on the inflow and outflow of cash for the project, the annual net cash flow was calculated. This was calculated as the summation of all cash inflow and outflow that occurs within the project. The inflow and outflow attracted positive and negative during the calculation. In this research, the annual sources of cash inflow include, revenue generated during gas sales, engine divestment and leasing due to redundant fleets of engines after overhaul. Also, the annual sources of cash outflow

include fuel cost that the engine utilised, O&M costs, emission tax, and the loan repayment.

Figure 6-43 shows the annual cash flow of the project after considering all variables as earlier mentioned. The result obtained shows that the loan repayment period in year 3 to 17 affected the cash flow thereby reducing the net cash flow.

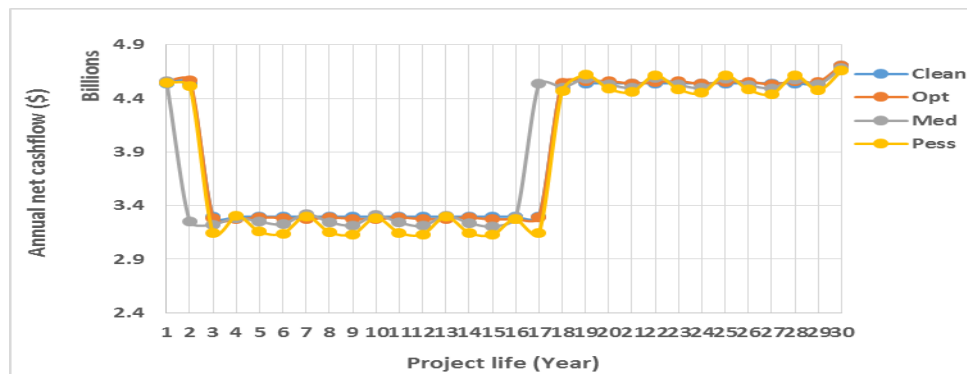


Figure 6-43 Annual net cash flow on the project

6.5.4 Engine divestment sales

The developed model to calculate the NPV has engine divestment of the redundant fleets after overhaul as one of the sources of revenue. Figure 6-44 shows the methods to calculate the benefits from engine divestment. The method shows that after engine overhaul, there will be redundant fleets, and these redundant engines have to be sold out, hence the need for engine divestment through sales. The benefits from this sales are been utilised in the economic model to ascertain the NPV.

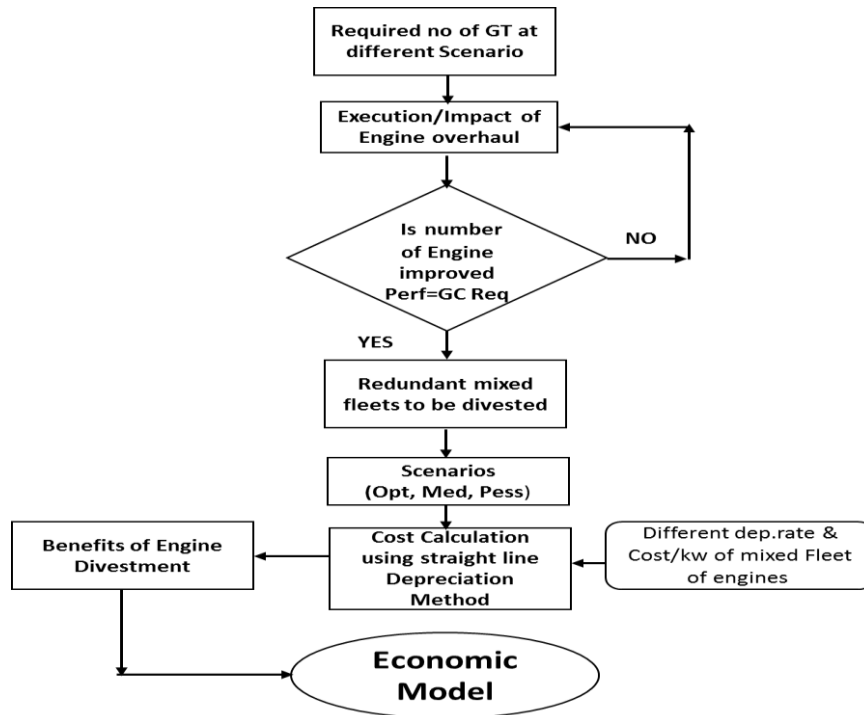


Figure 6-44 Methods to calculate the engine divestment

The engine salvage value was calculated using Equation (6-14)

$$S = P \times (1 - i)^y \quad (6-14)$$

S= Gas turbine engine salvage value in \$/kW

P = GT engine original capital cost (\$/kW)

i = GT engine depreciation rate

y = Year of gas turbine engine divestment (years)

The engine divestment sale was calculated using (6-15);

$$DS = P \times (1 - i)^y * GT_{SP} * NGT \quad (6-15)$$

Where,

DS = divestment sale of the GTs (\$)

GT_{SP} = shaft power of the Engine at DP (Kw)

All parameters remained the same as explained in Equation (6-14). 0.0516 was used as the depreciation rate for the clean engine[68]. Meanwhile, 0.0616, 0.0716, and 0.0816 were used as the depreciation rate for optimistic, medium and pessimistic, respective. These values were assumed based on the level of degradation in each scenario. This implies that the higher the degradation, the higher the depreciation rate of the engines. Straight-line depreciation rate was assumed in evaluating the engine divestment sales. This implies that the gas turbine rate of depreciation was constant in all the year with respect to each scenario.

The result obtained in Figure 6-45 shows that there was a decline in the benefits of divestment at each scenario as the year of operation increases. This means that as the year of engine operation increases, the salvage value reduces and this affected the cost value of the engine. The benefit at year 30 was the highest showing that all the engines were divested at the end of the project life. The clean becoming the highest shows that the clean condition was not affected much by the depreciation rate. Figure 6-46 shows the total benefits from divestment as indicated on the bar chart. The pessimistic scenarios becoming the highest shows that the highest number of engines was divested in this scenario. This influenced the cost calculation, thereby making it higher than all of the other scenarios. These benefits go into the economic module

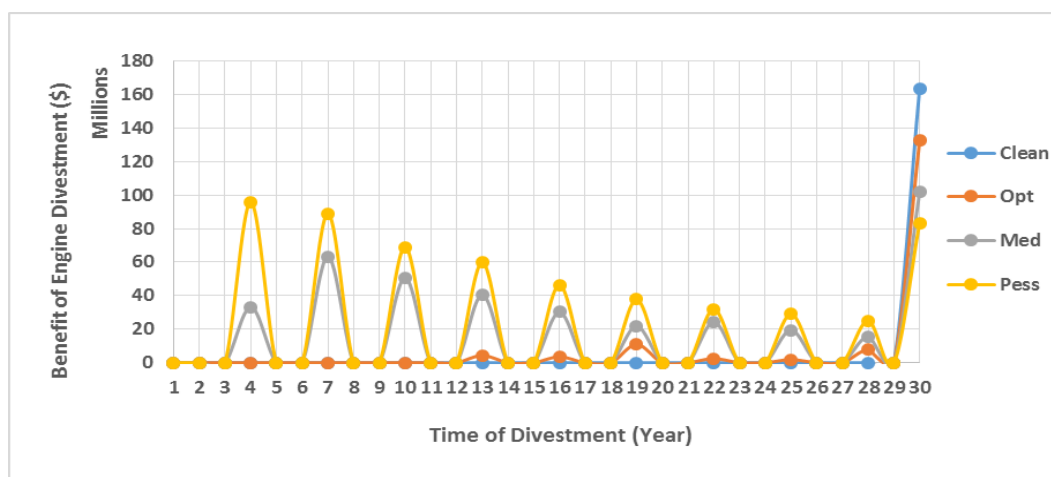


Figure 6-45 Divestment sale due to overhaul

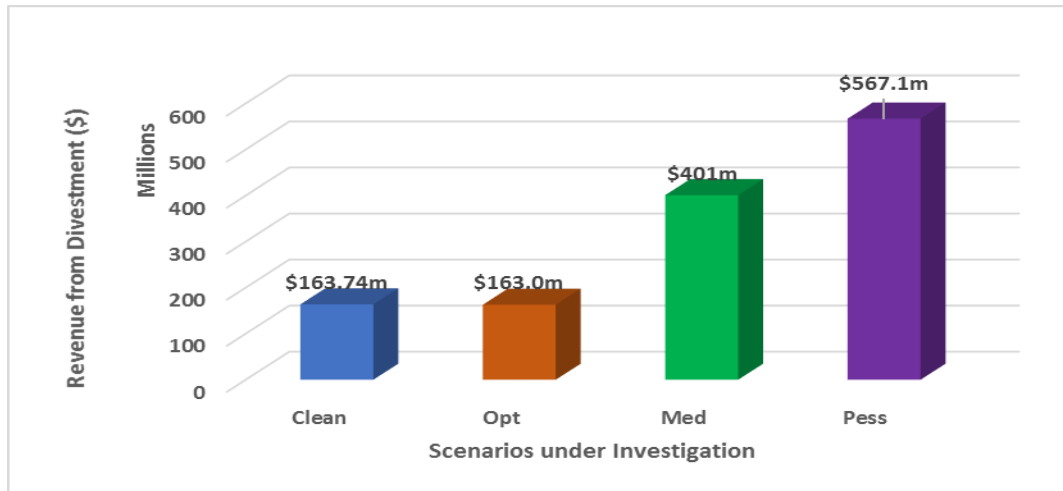


Figure 6-46 Total benefits from engine divestment

6.5.5 Net Present Value (NPV) Calculation

Figure 6-47 illustrates the method adopted in evaluating the NPV of the pipeline project under consideration. The following assumption was made; the total project life was 30 years, the production starts from the first year and 0% equity, which implies that the total capital cost of the project was funded by the debt borrowed from the bank.

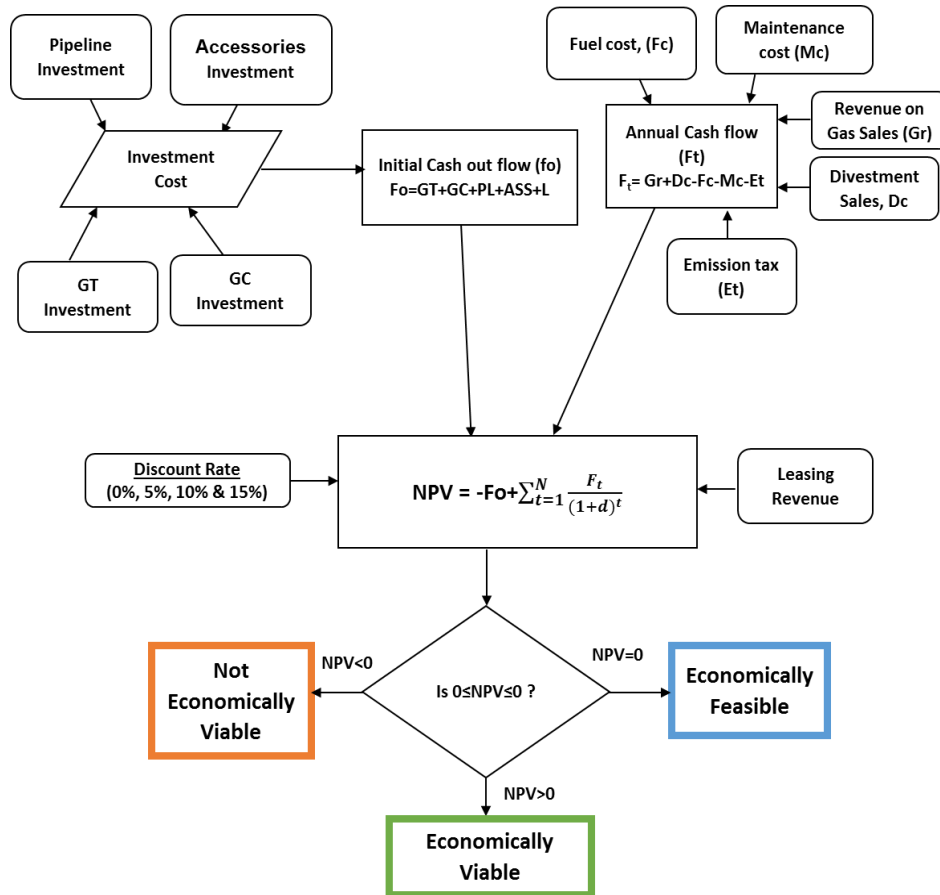


Figure 6-47 Method to evaluate the NPV

Concerning Figure 6-47, the NPV was calculated using

$$NPV = \sum_{t=1}^N \frac{F_t}{(1+d)^t} - Fo \quad (6-16)$$

Where, F_t is the net cash flow for the project life, $-Fo$ is the initial investment cost, which is also the loan taken, d is the discount rate. 0%, 5%, 10% and 15% discount rate. These values were assumed in this research, and t is the project year. It is important to state that the discount rate up to 15% was chosen arbitrarily but reasonable. The aim was to investigate the worst-case scenario in terms of benefits for investing in the TSGP project.

Figure 6-48. shows the result of the NPV with engine divestment at a different discount rate (DR) with respect to the various scenarios under investigation. Considering the clean condition of the engine operation, the NPV at 0% discount rate (DR) was \$103 billion. The result obtained for the opt, med and pess scenarios shows

that the NPV reduces by 1.2%, 2% and 3.9%, respectively when compared with the 0% discount rate. Again, for the 5% DR, the NPV at clean operation was \$43.80 billion. Accordingly, the result obtained for the opt, med and pess scenarios shows that the NPV reduces by 4.8%, 7.3% and 9, respectively when compared with the clean condition. Furthermore, for the 10% DR, the NPV at clean condition was \$21.50 billion. The result for the degraded condition shows that the NPV for the opt, med and pess reduce by 8.84%, 14.42% and 20.5 %. Finally, for the 15% DR, the NPV at clean condition was \$10.10. The result for the degraded condition also shows that the NPV for the opt, med and pess sceanrios reduced by 20.8%, 29.7% and 37.6%.

Generally, the results obtained show a declining rate in the NPV from the clean condition to the pessimistic scenario. This is showing in Figure 6-47 when the NPV at 0% DR was \$103 billion and ended up with \$99 billion at pessimistic. This is also seen in all the DR applied in this research. This clearly shows that, as engine degradation increases, the benefits on investment reduces. Conversely, if the degradation reduces, the benefits will improve, hence the need for maintenance activities that can reduce the level of degradation. However, the result obtained shows that the project was economically viable. It is essential to state that the various escalation rate as used in section 7.4 for the evaluation of the life cycle cost was not considered in the NPV calculation. However, only 0% escalation rate on the fuel, O&M cost and revenue were used for the NPV analysis. This is because the inclusion of the discount rate in NPV evaluation compensated for the escalation rate.

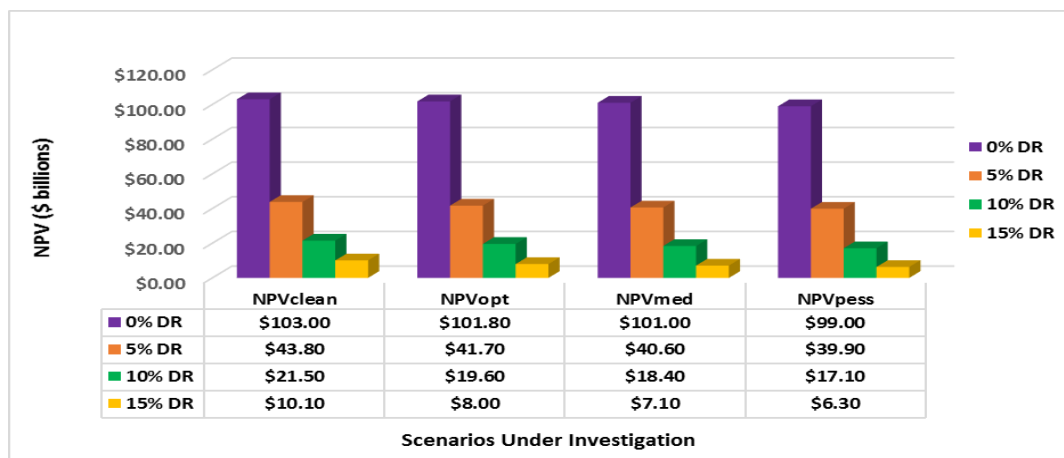


Figure 6-48 The influence of degradation and DR on NPV

6.5.6 The Influence of Engine Leasing on NPV

Apart from the outright sales (divestment) of the redundant engines after overhaul, another method to acquire the benefits from the redundant fleet is leasing. Leasing of GT becomes imperative as some engines become redundant after overhaul. These redundant fleets of engines are been leased and reused when needed. Figure 6-49 shows the method adopted in achieving the benefits of leasing.

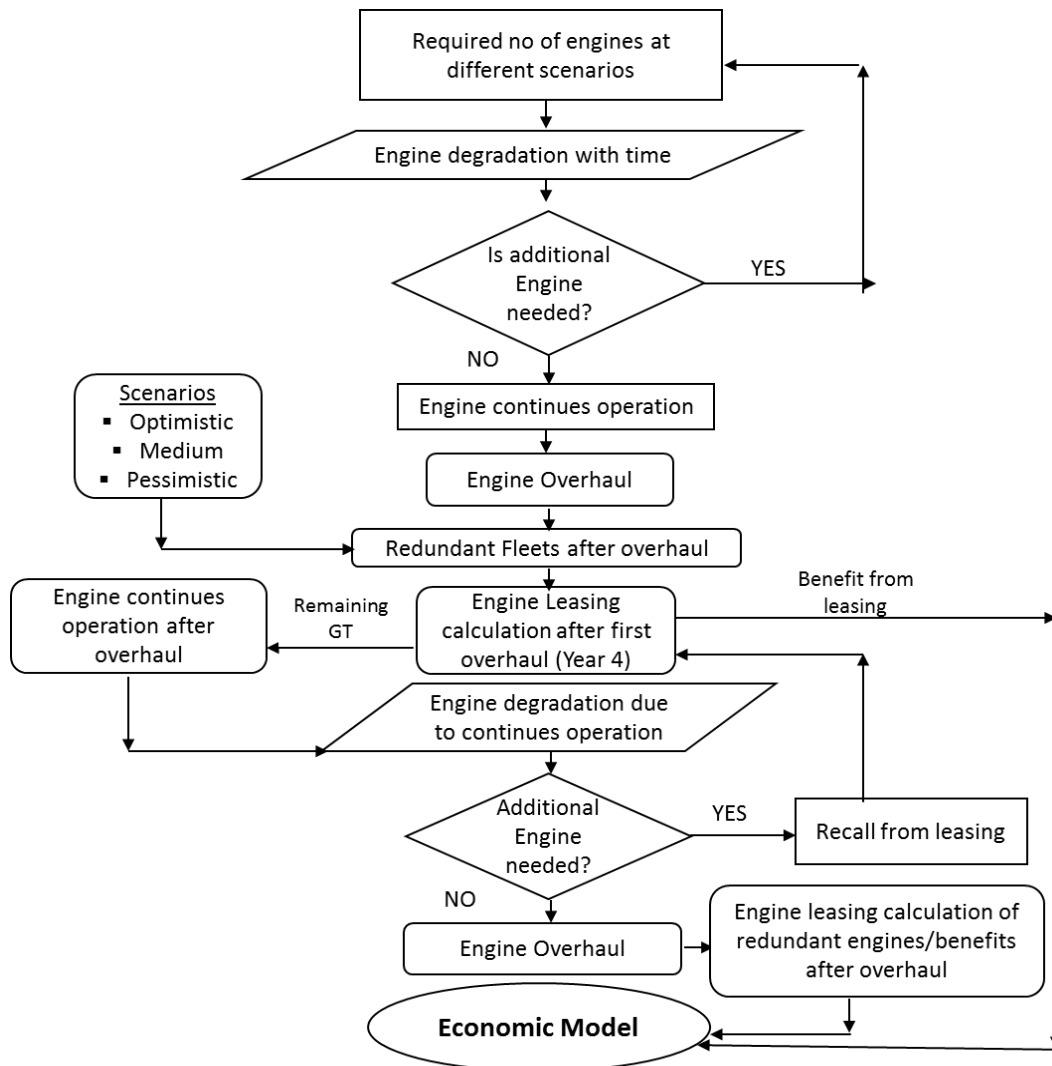


Figure 6-49 Methodology of leasing calculation

From the method above, as the degradation increases, more engines will be needed because of the limit set on the engines. After the first engine overhaul, the redundant engines will be leased to the lessee. The leasing of the engine depends on the remaining number of engines in each scenario that can give the power requirement to the GC. The remaining engines after overhaul will continue in service until the time

when additional engines are needed due to degradation. This happened because of the TET limit set on the engine operation. At this time, instead of procuring a new gas turbine as in the case of engine divestment, the engine will be recalled from the lessee based on agreement and reused in the process. The process will continue until all the leased engines are retrieved from the lessee if need be. It is important to state that new engines can only be purchased when all the leased engines have been retrieved from the lessee. New engines can only be procured if all the leased engines of a particular year can no longer satisfy the condition of operation.

The following are the underlying assumption in calculating the benefits of leasing

- (i) 3% of GT capital cost per month was used to calculate the cost of leasing
- (ii) The leased engines are recalled from leasing and re-used instead of buying a new engine when there is a need for a new engine
- (iii) Degradation does not affect engine leasing cost as the engines was assumed almost clean after overhaul. It only affects the number of engines to be leased after overhaul
- (iv) All engines are divested through sales at the end of the project (at year 30)

The Lessor's cost of leasing was estimated using Equation (6-17)

$$L_C = GT_{Capex} * NGT_{SC} * T \quad (6-17)$$

Where,

L_C = Cost of leasing

GT_{Capex} = Capital cost of GT engines

NGT_{SC} = Number of GT engines for each scenario

T = Time of the engines with the Lessee (Year)

Figure 6-50 shows the number of engines to lease, the number of engines to be recalled from leasing, and the total number of engine to be divested at the end of project life. The leasing and the recalled time of the engines are all represented on the curve

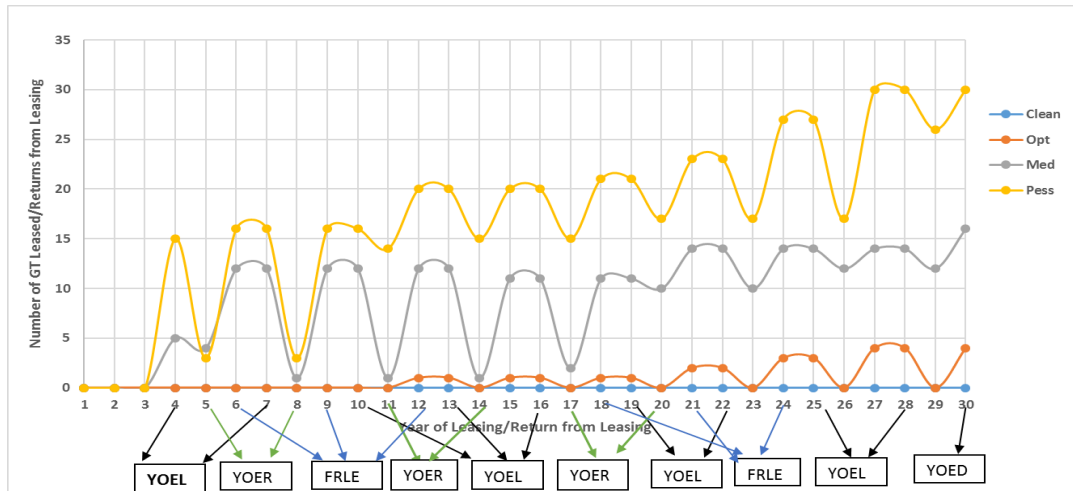


Figure 6-50 Engine leasing and return from service

From the curve above,

YOEL= Year of Engine Leasing

YOER = Year of Engines Return from Leasing

FRLE = Full Returns of Leased Engines with additional procured engines

YOED= Year of engine divestment

Figure 6-51 shows the revenue from engine leasing in all the scenarios. The result obtained shows that there was no leasing in the clean scenario. Leasing of engine for the opt scenario start at year 13 and progresses till year 30 of engine divestment. Also, the leasing of engines at medium and pessimistic scenario starts at year 4. At year 10, it was observed that the revenue for the medium with \$203 million was higher than the pessimistic scenario with \$160.8 million. The reason was that only one engine was recalled from leasing for the medium scenario while 14 engines were recalled from the lessee into year 11 for the pessimistic scenario. This is reflected in Figure 6-50. The recall of 14 engines reduces the revenue in the pessimistic scenario while the engine at medium scenario continues to remain with the lessee. The trend of leasing and recalling continues in all scenarios until the entire engines were divested at year 30. It was observed that the revenue from the divested engines reduces as the level of engine degradation increases. This is evidently shown as the total revenue from the clean, optimistic, medium and pessimistic are \$163.7 million,

\$ 131.2 million, 101.9 million and \$83.5. This is expected because all the engines apart from the clean engines were sold in their degraded condition.

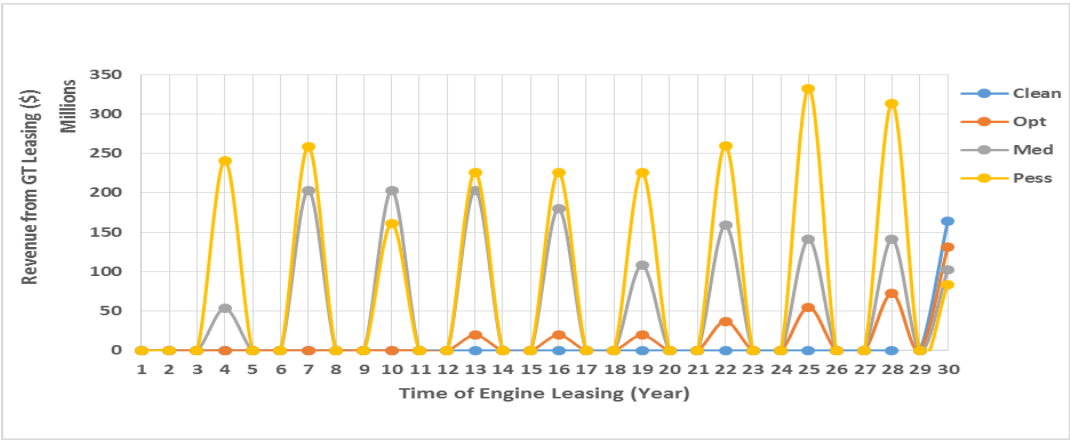


Figure 6-51 Revenue from Engine leasing and year 30 divestment

Having known the revenue generated at different leasing time, Figure 6-52 shows the total revenue generated from the engine leasing in addition to that of divestment at year 30. It is important to state that the total revenue showing on the clean condition indicates the divestment only as there was no leasing at the clean condition. The total revenue from in each of the scenarios as indicated on the curve shows both revenues accruable from leasing and divestment sale. Leasing from the pessimistic scenario was the highest because a higher number of engines were leased at the pessimistic scenario after overhaul

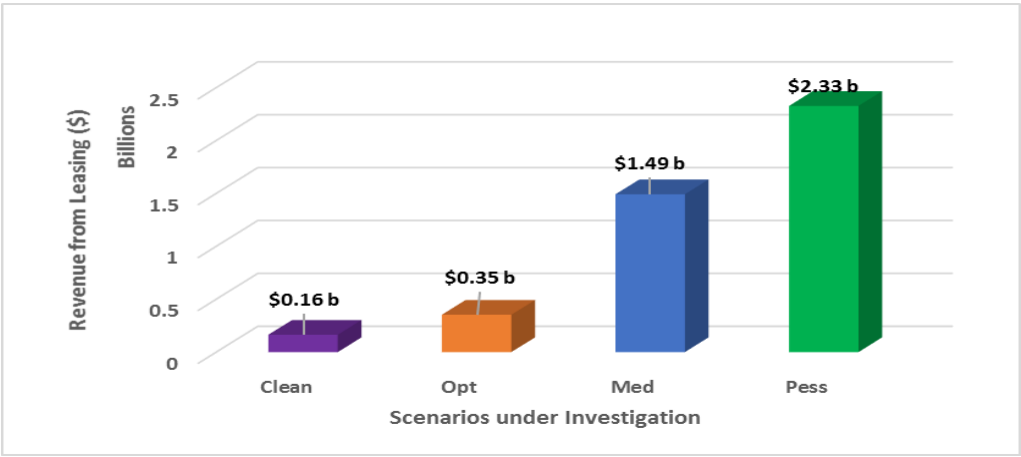


Figure 6-52 Total revenue from engine leasing and divestment at year 30

It is worth noting that the revenue from engine leasing goes into the economic model as an integral part of the calculation of NPV. Figure 6-53 shows the influence of engine leasing on the NPV at a different discount rate (DR). The result obtained shows that the NPV value of \$103.0, \$43.80, \$21.5 and \$10.10 billion for 0%, 5%, 10% and 15% discount rate respectively for the clean engine operation remain the same when compared with the NPV that was influenced by engine divestment option. The reason remains that there were no redundant engines at the clean engine operations.

Generally, There was an improvement in all scenarios for the NPV with leasing when compared with the divestment case. For instance, at 5% DR for opt, med and pess., scenario, the result showed that there was an increase of 2.6%, 3.6% and 3.9% for opt, med and pess scenario when compared with the NPV for divestment sale under the same discount rate. The result here shows that there was a reduction in NPV under engine divestment when compared with that of engine leasing.

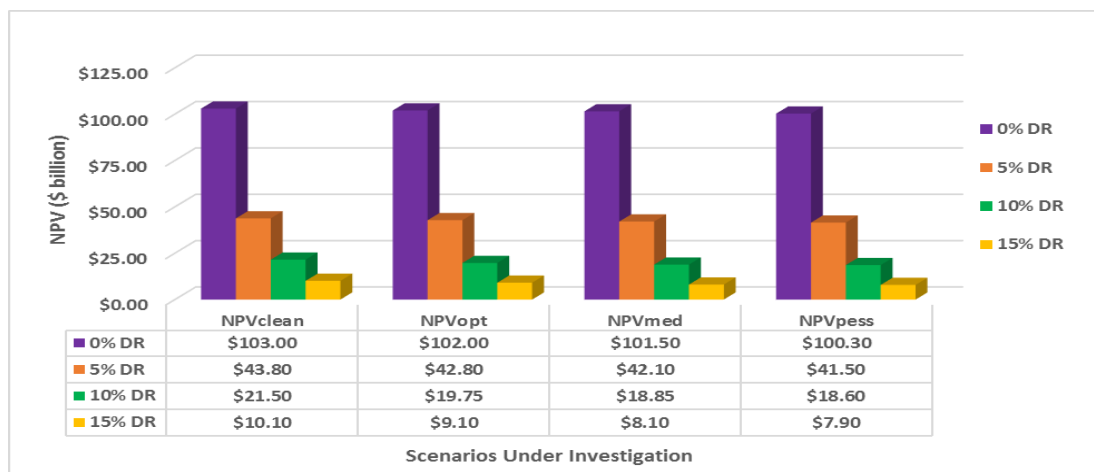


Figure 6-53 The influence of engine leasing on NPV

6.6 Conclusion

The economics of investing in the natural gas pipeline has been analysed and presented. The result obtained from the NPV analysis shows that the project is economically viable. Comparing the benefits on investment when applying engine divestment and leasing of the redundant engines after overhaul, the result shows that, the NPV achieved at optimistic scenario under engine divestment condition was 0.2%, 2.6%, 5.8%, and 12.1% corresponding to 0%, 5%, 10% and 15% DR, respectively lower

when compared with that of the engine leasing condition. The same trend was achieved for the medium scenario. Finally, that of the pessimistic scenario under divestment condition was 1.3%, 4%, 8.1%, and 20.3% lower corresponding to 0%, 5%, 10%, and 15% discount rate, respectively when compared with the engine leasing

The result obtained shows that the NPV with leasing has more promising result than that of engine divestment through outright sales. This is because in divestment after the engine was sold with less cost because of the depreciation rate of engines due to degradation, new engines were procured when the need for another engine arises. The purchase of new engines increases the capital cost, thereby affecting the overall benefits (NPV) of the project. However, in leasing, the same engine that was leased was recalled from leasing and re-used when the need arises. New engines were only procured when all the leased engines cannot satisfy the number needed at each compressor station. Conclusively, based on the above analysis, engine leasing improves the NPV than engine divestment. Hence, it is more beneficial to adopt the engine leasing approach than the engine divestment method

7 MAINTAINANCE ACTION

7.1 General Overview

The maintenance action adopted in this thesis is the implementation of a re-scheduled overhaul and compressor washing. Therefore, this chapter is divided into two sections. Section one deals on the impact of rescheduling of GT engine overhaul on the techno-economics of natural gas pipeline investment while section two deals with compressor washing as one of the maintenance action for the GT engines. The two sections evaluated their impacts on the techno-economic of the natural gas pipeline project. Based on this division, each section shall be discussed one after another

7.2 The Impacts of GT Re-scheduled overhaul on the Techno-economic Assessment of Natural Gas Pipeline Project

7.2.1 Overview of section one

This section presents how rescheduling of the gas turbine (GT) overhaul from the baseline case of three years to two years interval amid the different level of degradation affects the number of the gas turbine to be used in a compressor station without altering the original pipeline design for a particular amount of gas throughput. Also, the research further investigated how this phenomenon influenced the total capital expenditure (CAPEX) of the project, its impacts on the fuel, O&M, life cycle cost, and the NPV. Eighteen compressor stations the same with the baseline case was used for the investigation. TURBOMATCH was used to develop the engine models at a controlled Turbine Entry Temperature (TET) with fixed shaft power. The performance results were further used for economic investigation using a developed model in MATLAB. Three seasons (rainy, dry and hot seasons) were also considered in this analysis based on the location of Trans-Saharan gas pipeline being investigated. Compressors and turbines were degraded (fouled) as a single type of degradation producing three performance scenarios (opt, med and pess) in comparison with the clean conditions. The proposed approach will enhance engine life-extension strategies that engine life-cycle managers or natural gas pipeline investors may adopt to cost-effectively manage their engines while ensuring reliability and safety on the pipeline business.

7.2.2 Introduction

How long the Gas turbine (GT) engines can be kept in service safely during the project life in the midst of harsh ambient temperature and degradation, has become a growing concern to engine life-cycle managers. Also, the need to balance risk and cost of replacing completely deteriorated engines explains the growing interest in the application of life extension technologies or strategies for safely extracting maximum usage of the procured engines on the natural gas pipeline. The reason is that the engines are allowed to run for an extended period due to the economic benefits of the natural gas business. GT engines that work for an extended period need overhaul to revamp the engine and to keep the engine working to its useful life.

The baseline model instituted three years interval for the engine overhaul in chapter 6. However, the time interval between overhaul varies with engine make, model and condition of operation which may extend or reduce to longer or shorter service intervals as established in chapter 2. The present study under this chapter considered engine overhaul after two years interval different from the baseline case. This is regarded as a re-schedule overhaul (ROH) The main aim was to investigate how this idea of reducing the time between overhaul will affect the fuel cost, O&M cost, life cycle cost and by extension the net present value (NPV) of the pipeline project using the TERA concept with the developed economic model

7.3 Methodology

Figure 7-1 represents the developed method for implementing the concept of a re-scheduled overhaul (ROH) for the natural gas pipeline project. Having achieved the gas compressor power requirement and the modelled engines, the overhaul was implemented after the two years interval of each season of the year. The engine model with the implementation of the re-scheduled overhaul was developed in TURBOMATCH. This was done amid time-based ambient temperature and the different level of degradation (opt, med and pess) for each season of the year for thirty (30) years. The limit of the operational TET not exceeding 60k above the DP was also considered irrespective of the level of the engine degradation.

This implies that if the operational TET exceeded the limit, an additional engine of the same capacity with lower power settings would be added to the compressor

station. The implementation of two years interval for overhaul enabled the engine not to undergo much degradation as in the case of three years interval.

Based on the use of the different number of engines with their respective shaft power, conjugate performance maps of the various parameters such as the fuel flow and TET will be produced. These conjugate performance maps will form the economic data that was used for the economic investigation of the project.

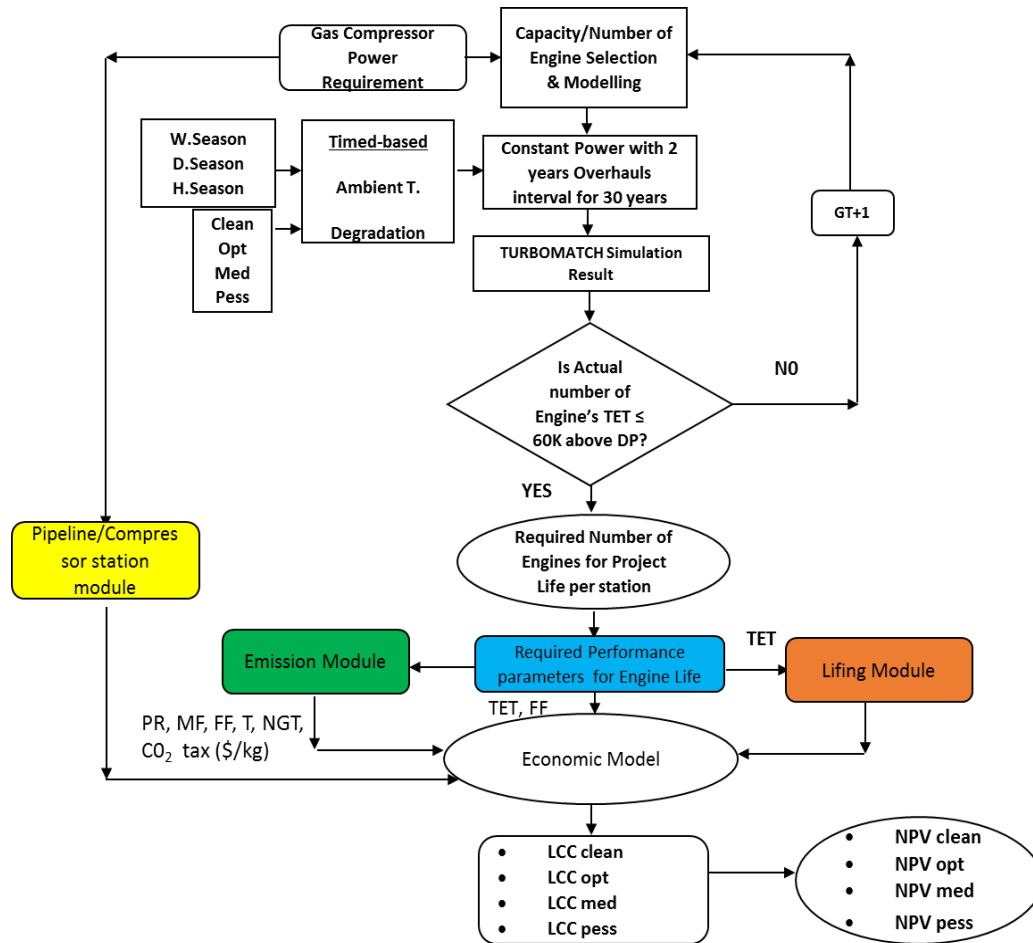


Figure 7-1 Methodology for the re-scheduled overhaul

The proposed approach will enhance engine life-extension strategies that engine life-cycle managers or natural gas pipeline investors may adopt to cost-effectively manage their engines while ensuring reliability and safety on the pipeline business.

7.3.1 Consideration of the Implanted Degradation base on ROH

Three degraded scenarios were also examined on the engines model, for the three scenarios in comparison with the clean condition. The three scenarios including the

clean condition were investigated in each season (winter, dry and hot season) of the year with the implementation of a routine overhaul after two years interval for the project life of thirty (30) years different from the three years overhaul of the baseline.

In the developed engine model and for each season of each year throughout the project life, different values of degradation were implemented with consideration to the assumed two years routine overhaul. The percentage of values used was implanted with respect to industrial practice and literature. The same pattern of the baseline case in modelling and degradation implantation was adopted but different time of overhaul implementation was considered in this case. All the scenarios were modelled for 30 years.

Figure 7-2, Figure 7-3, and Figure 7-4 showed the percentage values of degradation implanted for each scenario for winter, dry and hot seasons, respectively throughout the project life. In the winter season, as shown in **Figure 7-2**, opt ranges from 0.25% to 2.4%, med ranges from 1.25% to 3.4% and pess ranges from 2.25% to 4.4%. In the dry season, the implanted degradation for opt ranges from 0.5% to 2.65%, med ranges from 1.5% to 3.65% while pessimistic ranges from 2.5% to 4.65%. Finally, in the hot season, optimistic ranges from 0.75% to 2.9 %, the medium was between 1.75% and 3.9% while pessimistic ranges from 2.75% to 4.9%. An interval of 0.75% between each year of the first two years of operation was assumed for all the seasons. Also, an interval of 1% between each scenario per season was assumed to account for the different level of degradation. These are shown in **Figure 7-2 to Figure 7-4**. The implanted degradation is also presented in Table F 1 of Appendix H for more clarity.

The years of the overhaul are year 3, 5, 7, 9, 11, 13, 15, 17, 19, 21, 23, 25, 27, and 29. These years were chosen based on the assumption that the engine overhaul was implanted after two years interval for the project life. To account for the successive unrecovered degradation after each routine overhaul, the percentage from the specified range of values to be used depends on the scenario under investigation with their respective seasons. Therefore, 0.025% to 0.435% was implanted for the winter season to account for the years of the overhaul. To account for the unrecovered degradation after each overhaul for the opt scenario, 0.01% interval

between each overhaul was adopted while 0.03% was used as the interval for both medium and pessimistic scenario in the winter season.

A similar approach was adopted for the dry and hot season with alternating values. The main aim was to show that after each maintenance cycle, the engine goes into operation with some percentage of unrecovered degradation. These values were used to reduce the flow capacity and isentropic efficiency of both the compressor, compressor turbine and power turbine. This assumption was made based on literature and current industrial practices.

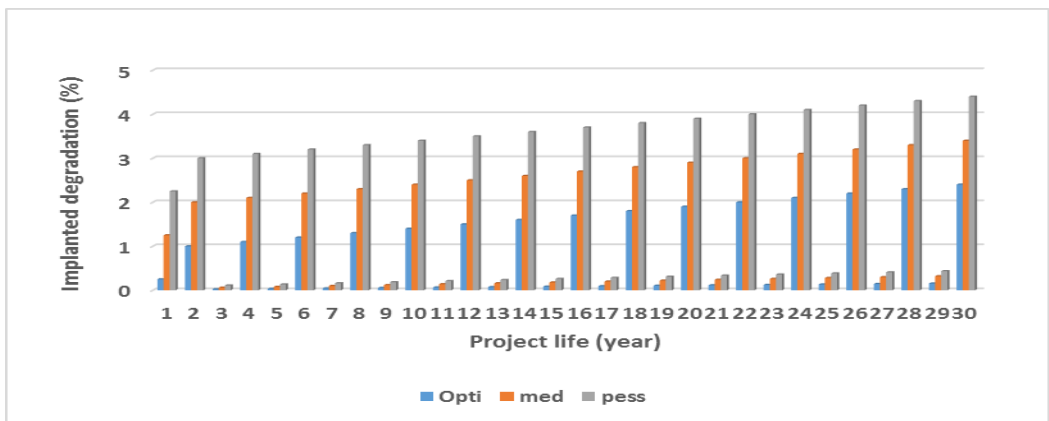


Figure 7-2 Implanted level of degradation for the rainy season

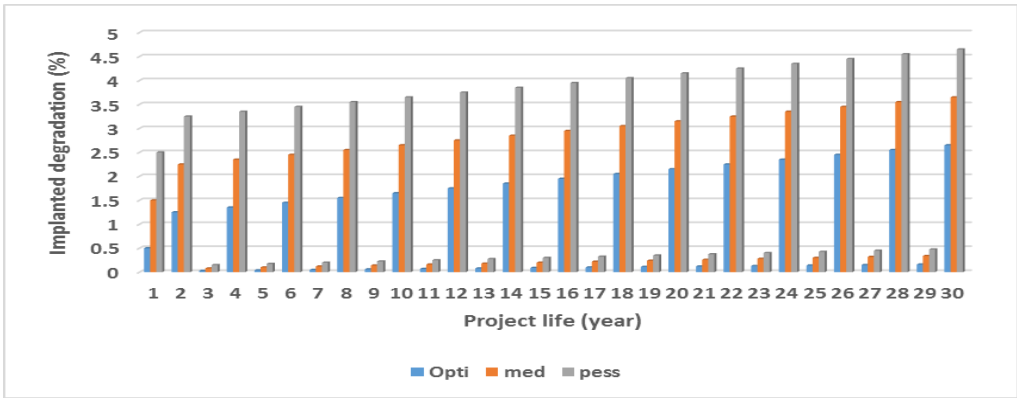


Figure 7-3 Implanted level of degradation for dry season

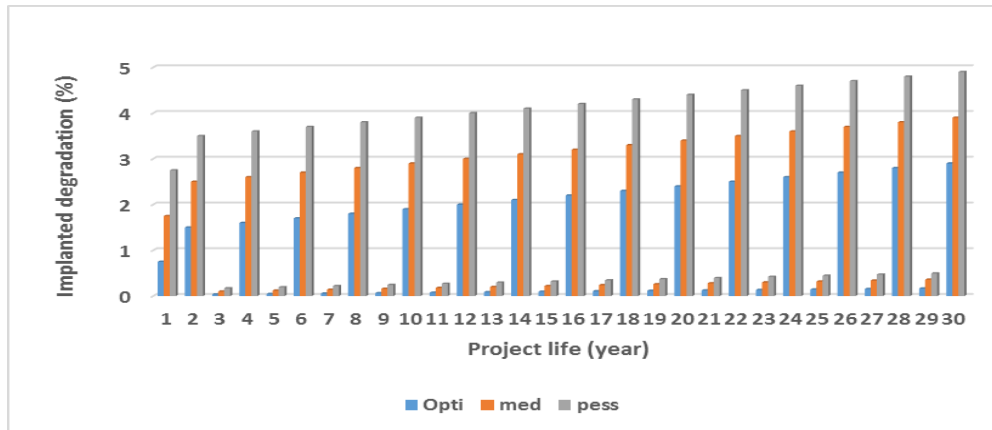


Figure 7-4 Implanted level of degradation for the hot season

7.4 Consideration of the Number of GT used for the Re-scheduled Overhaul

To ascertain the number of GT used for each compressor station under this case study, the method used in section 5.4 of chapter 6 to investigate the number of GT engines usage in each CS for the baseline condition was adopted. However, the difference here was the implementation of engine overhaul after two years interval, and this resulted in the engine having a lower level of degradation.

7.4.1 Result and Discussion

Based on the operational limit set on the TET of the engines, the TET profile has been analysed in addition to other performance parameters in all the 18 compressor stations (CS). For instance, Figure 7-5 and Figure 7-6 showed the time-based TET profile map for CS 2 in the rainy season for the baseline and ROH condition, respectively. The result shows a reduction in TET for the ROH when compared with the baseline condition. The same trend was achieved in all the CS and all seasons with respect to the various scenarios under investigation. The baseline and ROH result for the TET profile showed the influence of degradation on engine operation at constant shaft power. So the higher the degradation, the higher the TET, vice Versa. The analysis will determine if an additional engine is needed for other scenarios to reduce the TET or not based on the engine condition of operation. The engine below was expected not to exceed TET of 1610K since the DP TET was 1550K

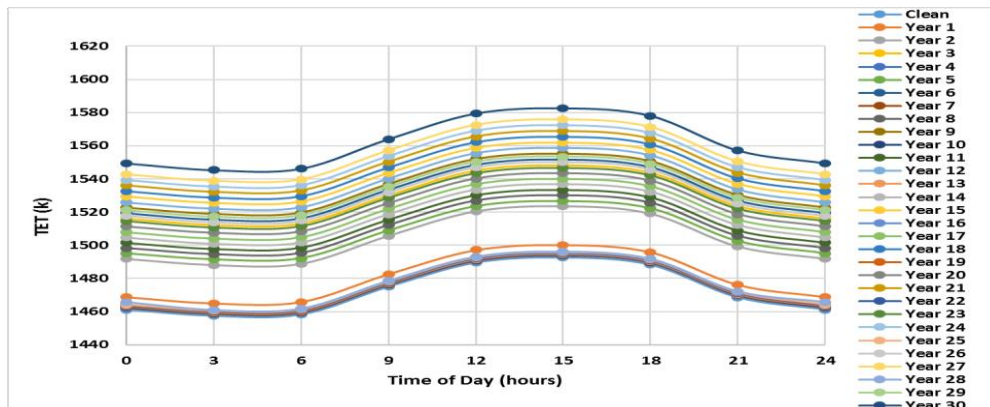


Figure 7-5 Time-based opt TET profile for the baseline condition -Rainy season

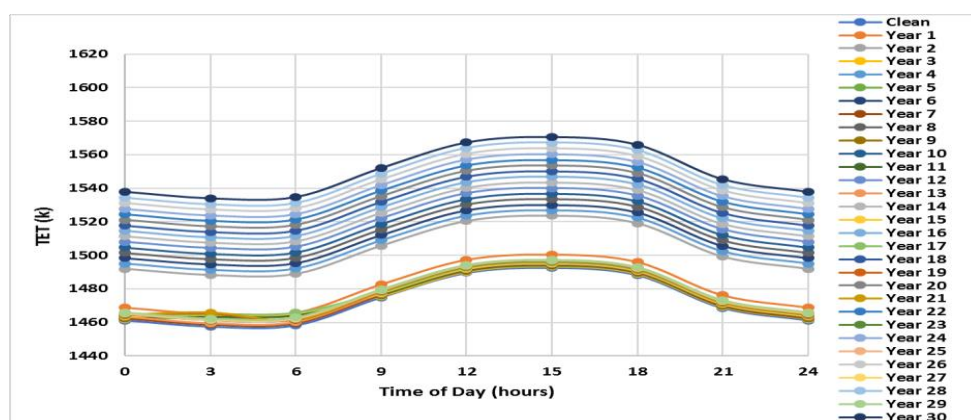


Figure 7-6 Time-based TET profile after ROH- Rainy season

Figure 7-7 shows the influence of the re-scheduled overhaul in determining the number of engines usage in each year. The result indicated that the re-scheduled overhaul reduces the number of GT used when compared with the baseline. This is evidently shown in Figure 7-8, where only the pessimistic scenario is represented. The reason was that the ROH technique reduces the level of degradation of the engine. This means that at the 3rd year where degradation would have been on the increase, engine overhaul took place to hinder a further increase in degradation.

With this technique, an additional engine that would have come in as a result of an increase in degradation was hindered by the introduction of the overhaul. This technique helps to reduce the number of engine usage in each of the compressor stations. Figure 7-9 shows the reduced number of GT used in each of the CS. It should be recalled that at baseline condition, CS 13 used 7 and 8 GTs for the medium and pessimistic scenario, but at the implementation of ROH, the number of GT

reduced to 6 and 7 GT engines at the same CS 13. The reduction of GT was also experienced by other CS. Meanwhile, the clean engine maintains the same number of engines throughout the project life in a similar way with the baseline case. This is because there was no degradation and overhaul for the clean condition.

It is necessary to state also that the outcome of the total number of GT used in each year for each of the CS was based on the power settings of the engines that will drive the GC. The operational TET of the engines was within the set limit.

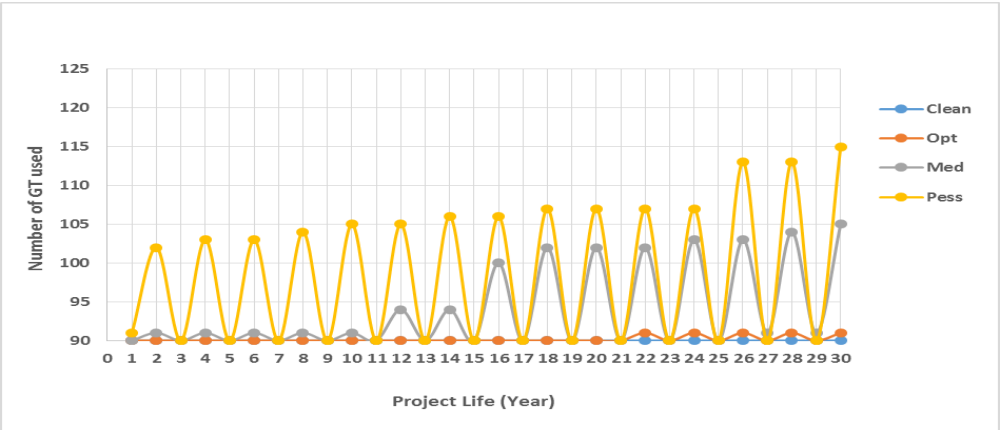


Figure 7-7 GT investment for the re-schedule overhaul

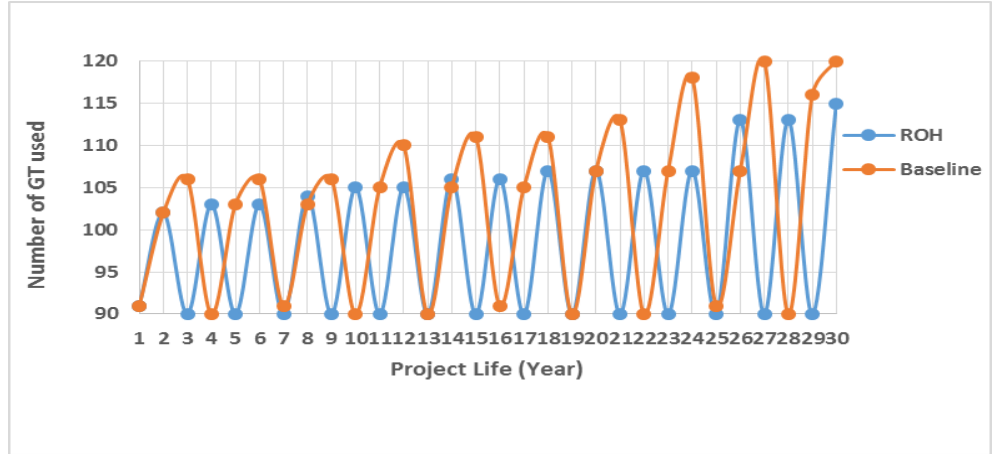


Figure 7-8 Number of engines used in the pessimistic scenario

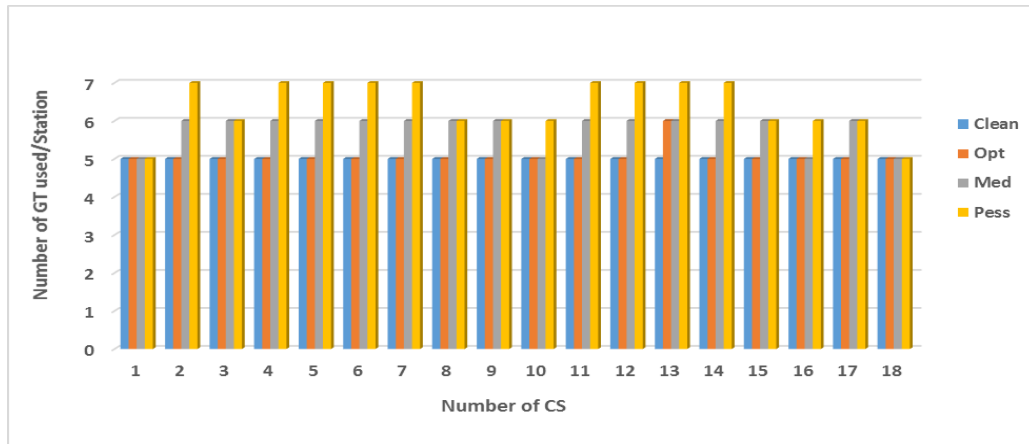


Figure 7-9 Number of GT used for each CS at different scenarios

Having evaluated the required number of GT in each CS, Figure 7-10 shows the total number of GT used in each scenario in comparison with the baseline case. The result shows that, the opt, med and pess scenario that used 99, 106 and 120 number of engines throughout the life of the engines for 18 compression stations at baseline reduces to 91, 104 and 115 respectively for the same number of compressor station and at the same operating conditions to deliver the same amount of .gas as in the baseline case. The opt scenario had the highest reduction in the number of GT because the opt has a lower level of degradation. Having a lower level of degradation enabled the system to quickly get rid of the degraded condition at the introduction of the overhaul after two years of operation when compared with the waiting time of 3 years before overhaul. The number of engines was also reduced in other scenarios based on this phenomenon. Figure 7-11 shows the number of different engines used in the mixed fleets.

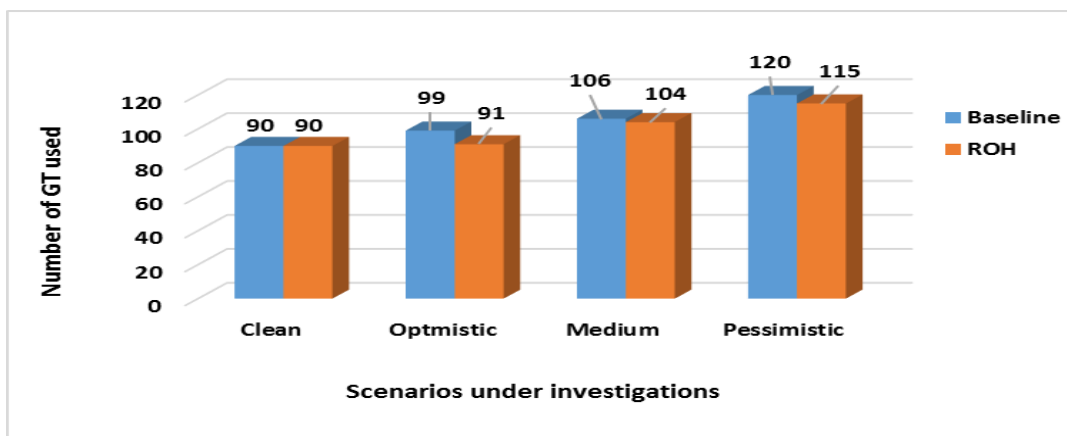


Figure 7-10 GT used for ROH in comparison with the baseline

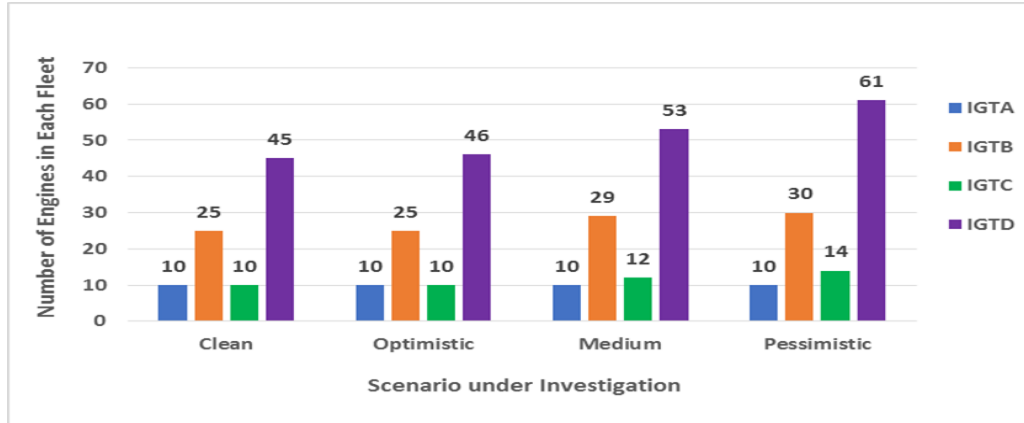


Figure 7-11 Number of each GT used in the mixed fleets for ROH

7.5 The Influence of Re-schedule Overhaul on the Economics

This section presents how rescheduling of gas turbine overhaul due to degradation affects the entire natural gas pipeline investment cost when compared with the baseline case. This cost includes the capital, the O&M cost (fixed, variable, TET based maintenance) and fuel cost., Its influence on the emissions and CO₂- tax was also evaluated. The investigation in this section extended to the evaluation of the life cycle cost and the net present value (NPV). This will show the viability of the natural gas project when compared with the baseline case. Three(3) degraded scenarios were also examined.

It is important to state again that, the implemented methodology to evaluate the economics of the pipeline project for the baseline case in chapter 6 was adopted in this section. These methods involve GT, GC, pipeline and its accessories calculation. The methods also include the fuel, O&M cost, LCC and NPV. Finally, the developed TERA framework, the developed emission, and economic model, as explained in chapter 3, 5 and 6 respectively were used for this section.

7.6 Result and Discussion

The result shows the influence of the re-schedule overhaul on the capital expenditure (CAPEX), the life cycle cost, and the net present value of the project. The results are shown in comparison to the baseline case.

7.6.1 The Influence of Re-schedule overhaul on the CAPEX

The capital expenditure (CAPEX) of the natural gas pipeline comprises of the cost of the pipeline, gas compressor, gas turbine and all the relevant pipeline accessories as considered in this research. Figure 7-12 shows the cost of GT in comparison with the baseline cost. The result revealed that the opt, med and pess reduced by 8.2%, 2% and 4.3%, respectively when compared with the cost of the respective scenarios in the baseline cost. Figure 7-13 shows the total CAPEX for the pipeline project when compared with the baseline case. The result obtained indicated that the opt, med and pess scenarios reduce by 1.5%, 0.38% and 0.9% when compared with the baseline case. The differences in the CAPEX for the ROH were attributed to the reduced number of GT and the influence of the contingency cost.

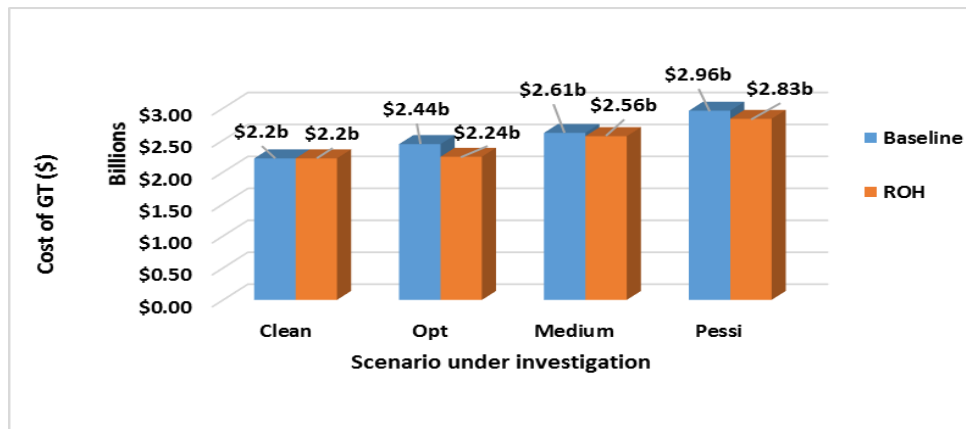


Figure 7-12 Cost of GT in comparison with the baseline

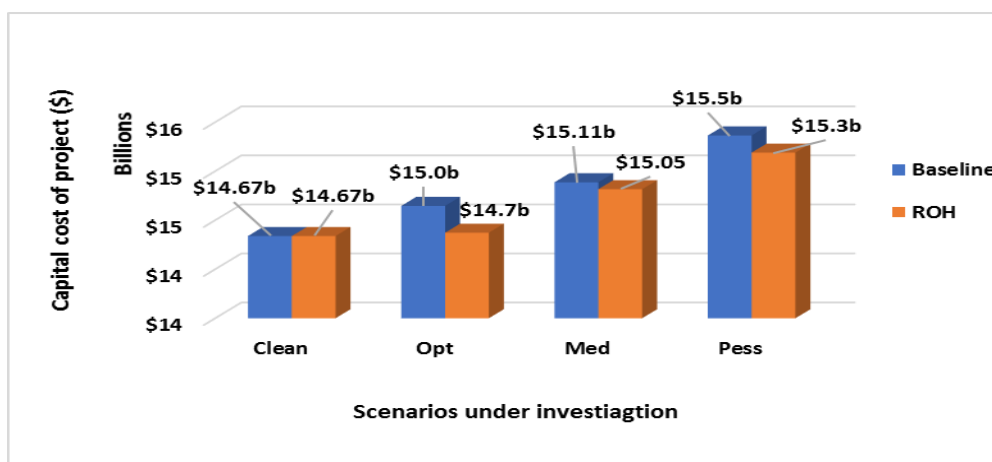


Figure 7-13 Total CAPEX of the project-ROH

7.6.2 The Influence of re-scheduled overhaul on the Fuel and O&M costs of the Pipeline Project

Figure 7-14 shows the annual fuel cost under the res-scheduled overhaul (ROH). The clean condition for both the baseline and ROH maintained an annual cost of \$0.424 billion for the 18 CS. However, the degraded condition (opt, med, and pess) varies from year to and assume almost clean at the year of engine overhaul. The increasing trend on the curve as the year increases revealed the influence of unrecovered degradation after each maintenance cycle which increases with the year of operation. This affected the cost of fuel as the operation continues

Figure 7-15 shows the total fuel cost alongside with the baseline case for all the scenarios under investigation. The result shows that the total fuel cost for the baseline case and clean condition remained approximately \$12.7 billion for thirty (30) years of operation. The result obtained also shows that the total fuel cost for the opt, med., and pess., scenario reduces by 2.8%, 3.01% and 4.7% respectively when compared with the baseline case.

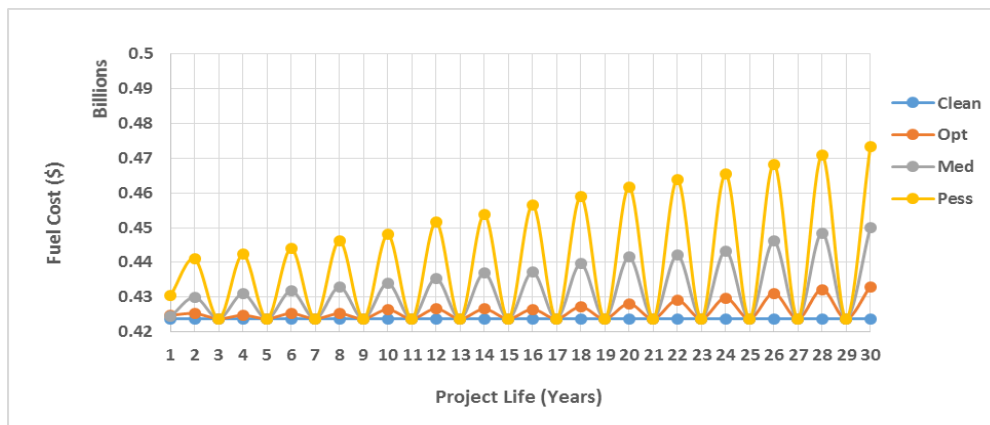


Figure 7-14 Annual fuel cost for the project life-ROH

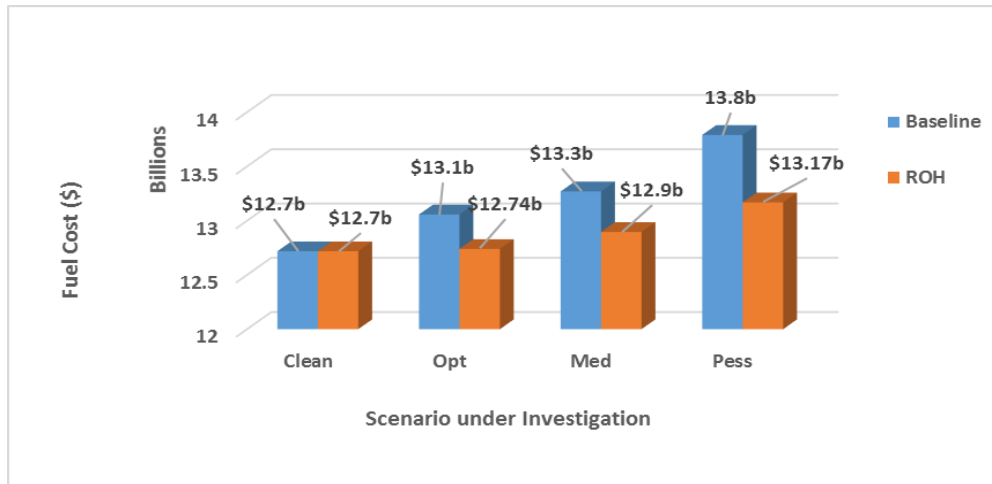


Figure 7-15 Total fuel cost for project life-ROH

Figure 7-16 showed the annual O&M cost when all the 18 CS was calculated. From the result, O&M cost for the clean condition maintained an approximate annual cost of \$0.11billion throughout the life of the project as it was in the baseline case. The result still showed that the O&M cost for the clean condition was higher than the opt., scenario in all the years. Again, apart from the year of engine overhaul, the clean condition was higher than the med. from year one until year 17. At year 18 upward, the medium was higher than the clean condition. This is because the engine at the clean condition was operating on a higher TET than the degraded scenarios for the years that the O&M cost was higher than the opt and medium.

It should be recalled that the maintenance of these engines was TET-based maintenance. The degraded condition operates on lower TET as a result of part-load operation due to a higher number of engine usage. This makes the creep life of the engine at the clean condition to be low, thereby influencing the maintenance of the engines. The implication of this is a direct effect on the maintenance cost, thus making the maintenance of the engine at the clean condition to be high. The alternating values on the curves for the degraded condition (opt, med, med) indicates the influence of the number of engine usage in each scenario before and after each maintenance cycle (overhaul). The degraded scenario alternate in their O&M cost due to reducing and increasing the number of GT before and after overhaul. This makes the operational engine life to fluctuate, thus increasing and reducing the O&M cost. Although the degraded scenario operates with a lower TET with a corresponding higher creep life of the engines, the higher O&M cost in the pessimistic

scenario was due to the influence of the higher number of GT used in this scenario and a corresponding value of degradation.

Generally, the O&M cost for the re-scheduled overhaul was lower than the baseline case. This is shown in Figure 7-17, where the total O&M cost is expressed in comparison with the baseline case. The result obtained showed that the total maintenance cost for the clean engine operation remained approximately \$3.24 billion. The further result revealed that the opt., med., and pess., scenarios had a reduction of 4.03%, 1.9% and 5.1%, respectively in their O&M cost when compared with the baseline case. The medium scenario having a lower value was as a result of the lower number of GT reduction between the baseline and the ROH condition

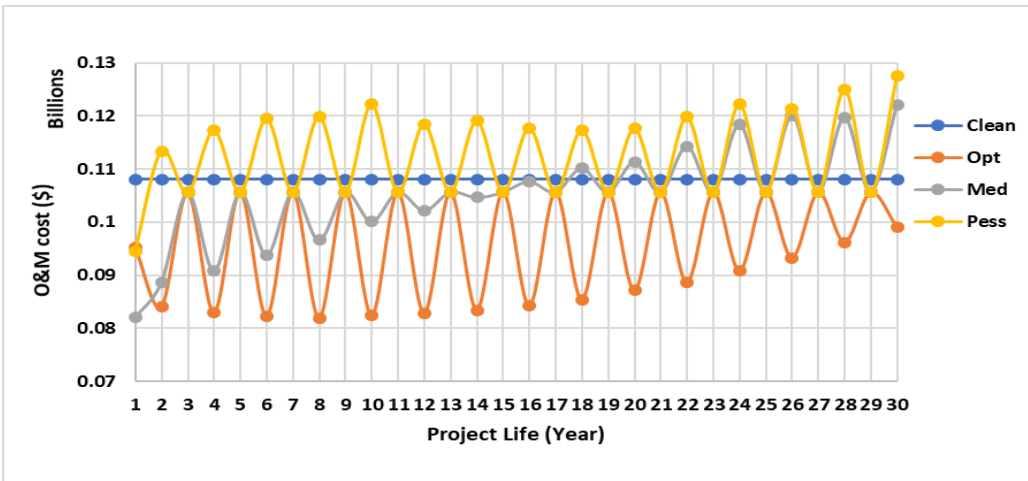


Figure 7-16 Annual O&M cost for the life-ROH

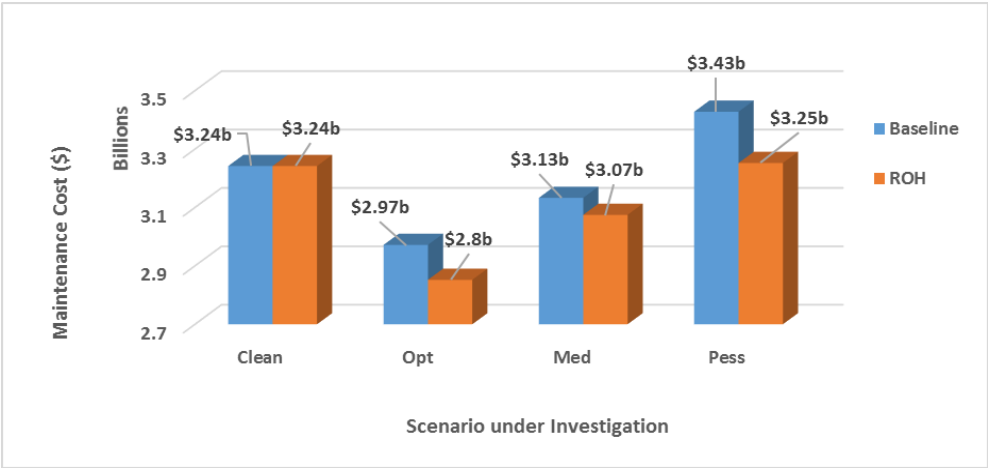


Figure 7-17 Total O&M costs for ROH in comparison with the baseline

7.7 The Impacts of ROH on GT Emissions

The emission considered under this section is the CO₂ because it is only the CO₂ tax that was considered in this research. Therefore, the impacts of the ROH on the CO₂ formation and tax was investigated and presented.

It is important to establish that the developed methodology and model in chapter 5 for the evaluation of emission for the pipeline project is adopted in this section. The time-based performance result from TURBOMATCH based on the ROH of the engines was linked with the developed emission model to evaluate the impact of the ROH on emission generation as well as the emission tax. Figure 7-18 shows the annual emission generated throughout the 18 CS under the ROH. The total annual CO₂ emitted was the summation of the time-based emission for each season of the year. So the addition of the three seasons makes up the annual CO₂ as represented in Figure 7-18. The result shows that the clean condition maintained an annual CO₂ emission of 2.45 billion kg same as the baseline. The increasing trend in each of the scenario before and after overhaul as the year increases indicates the accumulation of the unrecovered degradation after each maintenance cycle.

Furthermore, Figure 7-19 shows the total emission generated throughout the project life. Based on the total number of GT used for for the 18 CS, the clean condition maintained a total of 73.4 billion kg of CO₂, the same as the baseline case. The further result obtained for the emission showed that the opt., med., and pess., reduced by 1.6%, 2.1% and 3.3% respectively when compared with the baseline condition. Generally, ROH reduces CO₂ emission.

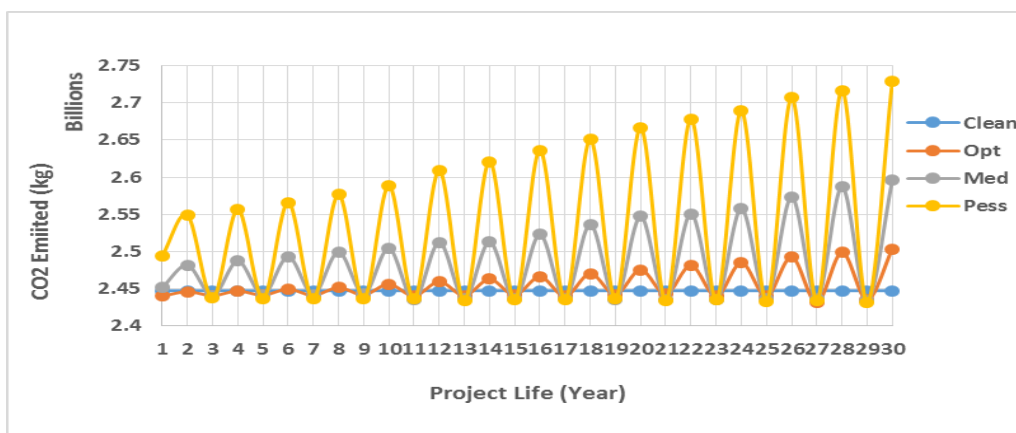


Figure 7-18 Annual CO₂ emission for the project life

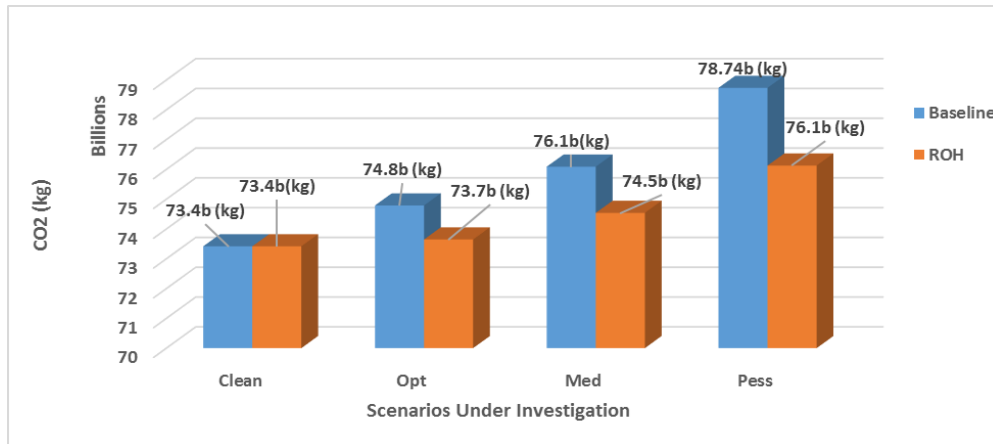


Figure 7-19 Total emission generated for the ROH throughout the project life

Based on the amount of CO₂ emission generated, Figure 7-20 shows the annual emission tax produced throughout the project life. The result shows that at the clean condition, a constant annual emission tax of \$49.26 million was maintained as it was in the baseline condition, while the degraded condition increases with increasing degradation with time. Figure 7-21 shows the total CO₂ emission tax throughout the project life. The result revealed that the opt., med., and pess., reduces by 1.6%, 2.1% and 3.7% when compared with the baseline condition.

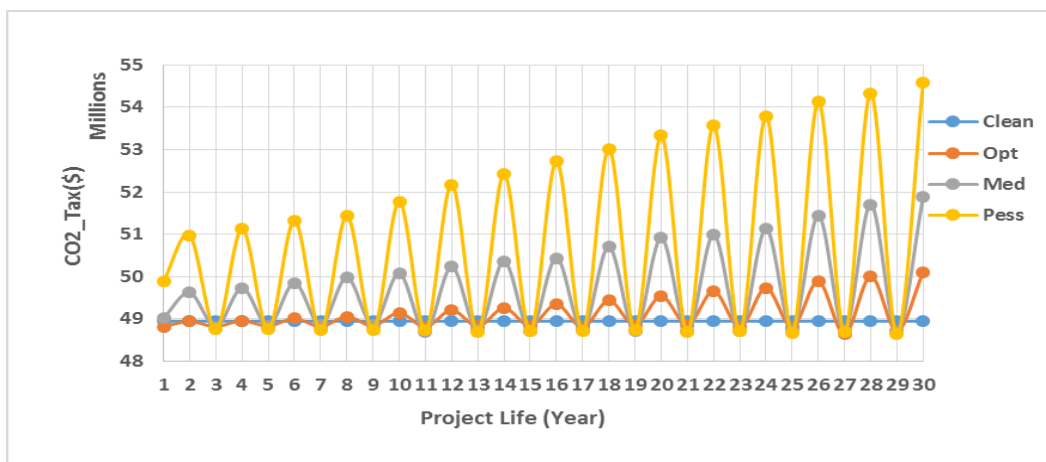


Figure 7-20 Annual emission tax for ROH throughout the project life

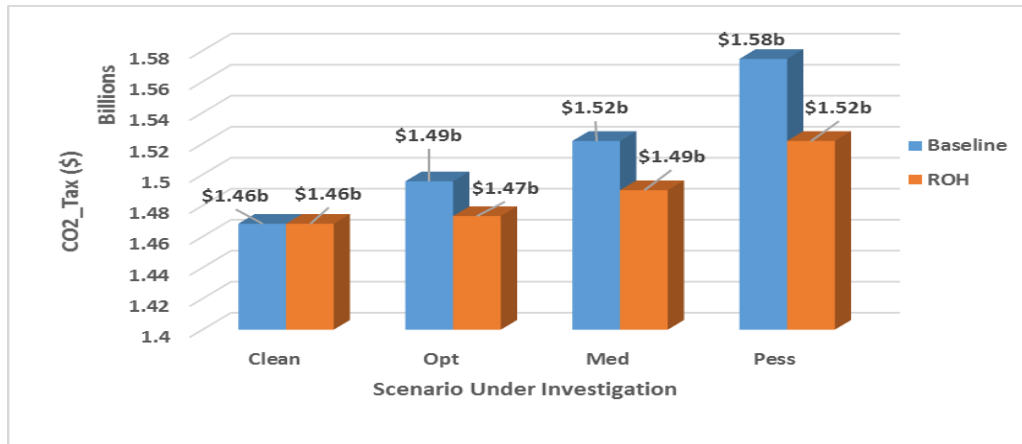


Figure 7-21 Total costs of CO₂ emission for ROH throughout the project life

7.7.1 The Influence of ROH on the Life Cycle Cost (LCC)

The life cycle cost consists of the CAPEX, fuel and O&M cost, including the emission tax. Figure 7-22 presents the LCC for the ROH when compared with the baseline case. The result shows that approximately \$32.01 billion remain the LCC for the clean condition for both the baseline case and ROH. The result further showed that the opt, med, and pess scenario experienced a reduction of 2.1%, 1.6% and 3.0% respectively when compared with the baseline LCC. This implies that implementing ROH saves \$0.7, \$0.54, \$1.01 billion for opt, med and pess, respectively when compared with the baseline case. The variation in the LCC was attributed to the reduced number GT, which influenced the CAPEX, emission tax, fuel and O&M cost. The benefits here indicated that, apart from saving some amount of money, implementing ROH enabled engine availability despite the level of degradation. This shows clear benefits of running the engines at lower TET which improve the creep life of the engines. It also shows that the technique enables the engine to operate throughout the desired project life despite the level of degradation.

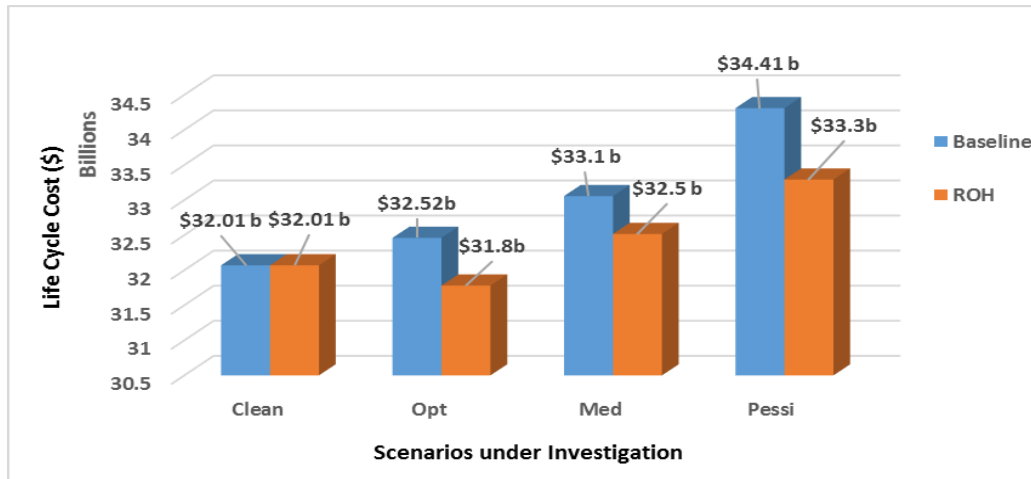


Figure 7-22 Life cycle cost-ROH

7.7.2 The Impacts of Reschedule Overhaul on NPV

Under this sub-heading, two cases shall be addressed. This include:

- i) The influence of ROH on engine divestment for NPV evaluation
- ii) The influence of ROH on engine leasing for NPV evaluation

7.7.2.1 The Influence of ROH on engine divestment for NPV Evaluation

Figure 7-23 shows the annual benefits from engine divestment for the ROH. Generally, the benefits of divestment in the baseline case was higher than that of the ROH basically because less number of engines were used in the later than the baseline as previously presented. The lower divestment benefit was caused by the depreciating rate of the engines at the degraded condition as was applicable to other degraded scenarios. The higher divestment benefits in medium and pessimistic scenarios were as a result of the influence of the higher number of engines that were divested at their respective years. Generally, the result also shows that as the year increases, the divestment benefits cost reduces due to the engine deterioration with time.

Meanwhile, at the end of the project, all the engines were divested through sales as represented in Figure 7-23. However, the increase in the medium scenario from year 13 to 19 was due to the higher number of engines used in the preceding years before overhauling. Therefore, the higher number of redundant engines influenced the benefits of divestment in those years, thus making it go higher instead of coming down. After year 19, it started coming down due to the influence of engine

depreciation as the divested engine stabilised. Figure 7-24 showed the total benefits from the engine divestment for the ROH in comparison with the baseline case. The result obtained showed that the benefit from the clean condition remained approximately \$163.74 million as it was in the baseline condition. The benefits from the clean engine show the benefit of divestment at the end of the project life. The further result shows that the benefit from divestment for opt., med., and pess., reduces by 22.7%, 32.5% and 28.4, respectively when compared with the baseline line condition. The non-linearity in the percentage shows the relationship with the baseline values as indicated on the bar chart.

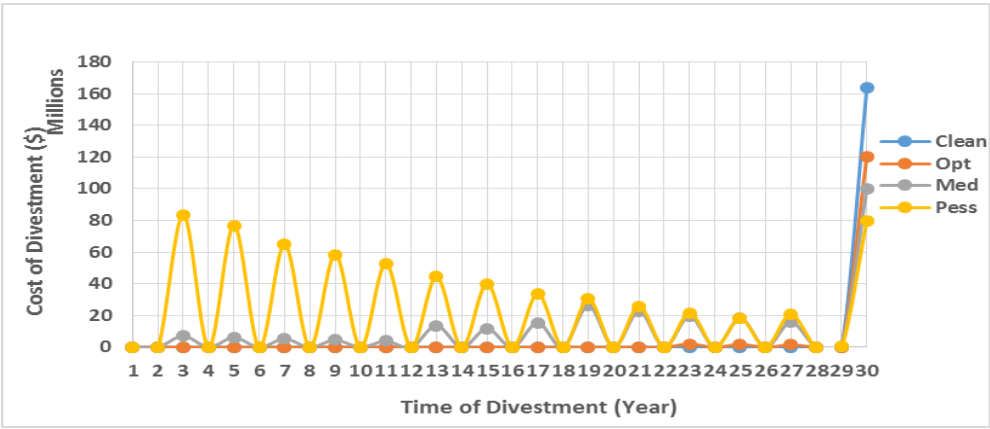


Figure 7-23 Annual benefits from divestment

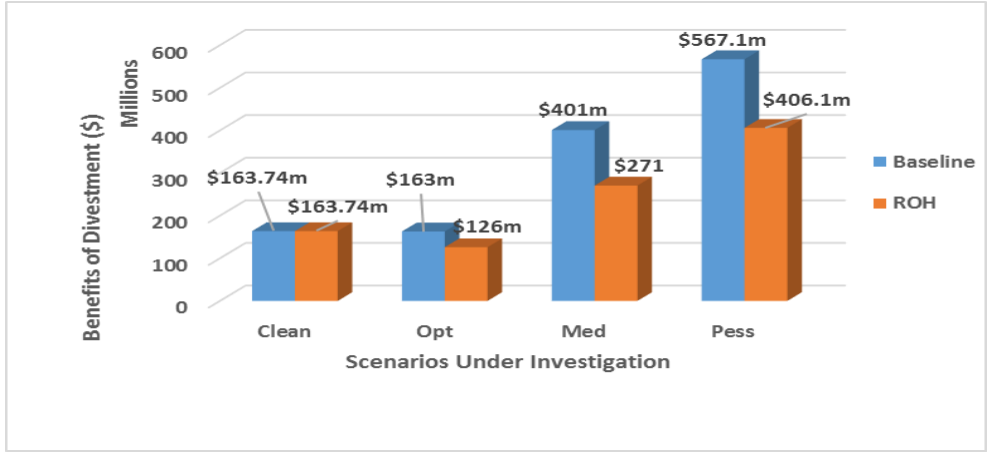


Figure 7-24 Benefits from engines divestment –ROH

Discounting technique and the addition of “ divestment term” based on NPV calculation method is at this moment presented to allow initial assessment of the likelihood of various scenarios being attractive for the pipeline project. Figure 7-25

shows when divestment is accounted for during NPV calculation under ROH. The result shows that as the discount rate increases, the NPV reduces. For a specific discount rate (DR), the NPV reduces as the degradation increases. For instance, at 0% DR, the NPV for the clean, opt., med., and pess., had \$103.0, \$102.5, \$101.90 and \$100.6 billion, respectively. Also, at 10% DR, the NPV for the clean, opt., med., and pess., had \$21.5, \$19.86, \$18.80 and \$18.0 billion, respectively. This implies that at 10% DR under ROH, there was an increase of 1.3%, 2.2% and 5.3% for opt., med., and pess respectively, in NPV when compared with the baseline. Generally, the NPV result under ROH for the degraded conditions shows promising benefits when compared with the baseline case. The reason was that the reduction in the number of GT usage, fuel cost, O&M cost and emission tax makes the NPV for the ROH higher than the baseline case. The result generally shows that the project is worth investing.

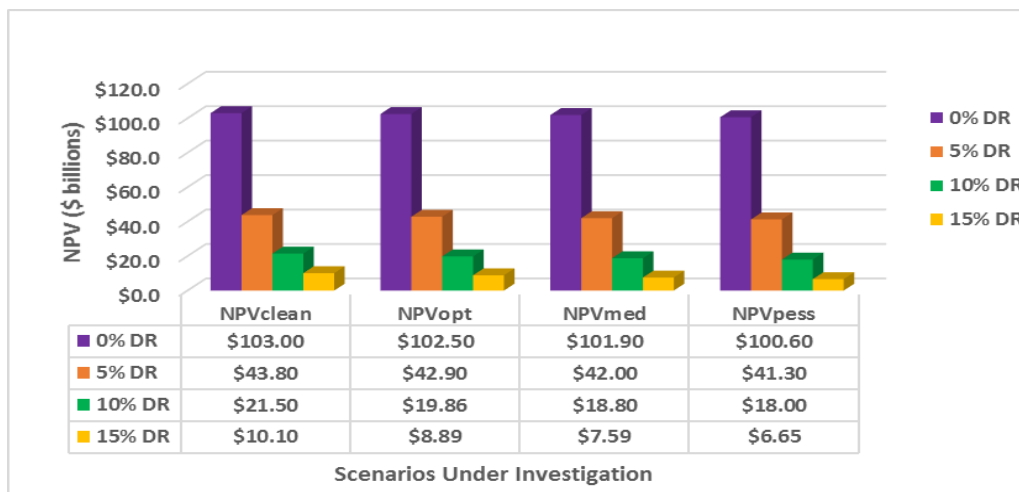


Figure 7-25 The Influence of engine divestment on NPV for ROH

7.7.2.2 The influence of ROH on Engine Leasing for NPV Evaluation

Another case study was the consideration of engine leasing for the redundant fleets after overhaul. The result from the leasing of engines is presented in Figure 7-26 where the total benefits is a representation of the benefits of engine leasing and that of the divestment of all the engines at the end of project life.

The result obtained shows that the benefits from the clean condition remained \$163.74 million a value which only shows the benefit of all the clean engine from divestment at the end of the project life. The further result obtained showed that,

there were 55.9%, 34% and 9% reduction in the benefits for the opt., med., and pess., respectively from leasing when compared with the baseline case. This is expected because of the reduction in the number of engines under the ROH. The med., and pess having a lower percentage value compared to the opt scenario shows that more engines were used in this scenario.

Furthermore, Figure 7-27 shows the influence of leasing on the NPV for the ROH. The usual trend of decreasing DR, resulting in a decreasing NPV was obtained. Although the benefits from engine leasing for the baseline case was higher than the ROH, the NPV achieved in ROH was better than the baseline case. This is because the yearly maintenance cost, fuel cost, emission tax as well as the CAPEX were reduced in the ROH. This influenced the NPV calculation, thereby making the NPV higher than the baseline case. From the result obtained, it was observed that the NPV obtained when considering engine leasing was higher than the divestment condition. For instance, at 10% DR, the opt., med., and pess., was 0.2%, 3.7% and 5.6% higher when compared with the NPV under engine divestment. Generally, in all the discount rate under consideration, the NPV for engine leasing was higher than divestment.

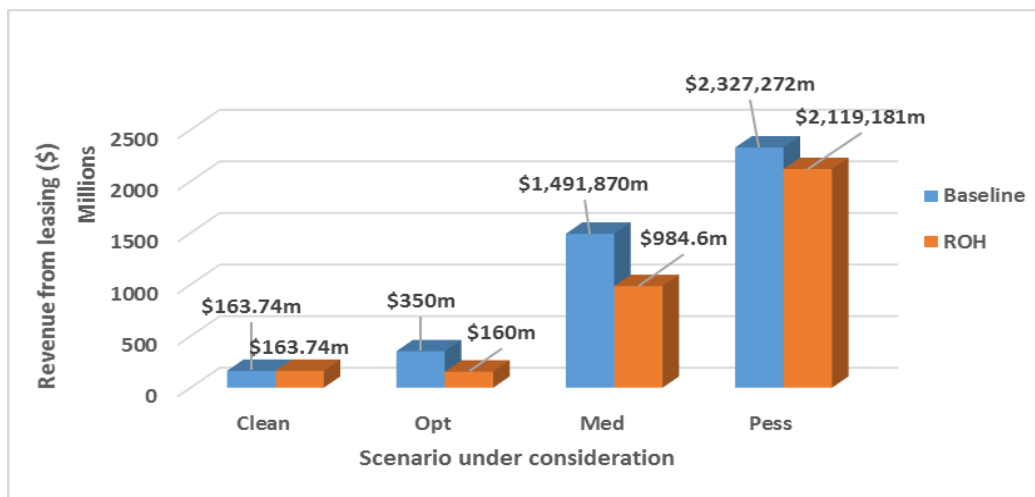


Figure 7-26 Revenue from leasing –ROH

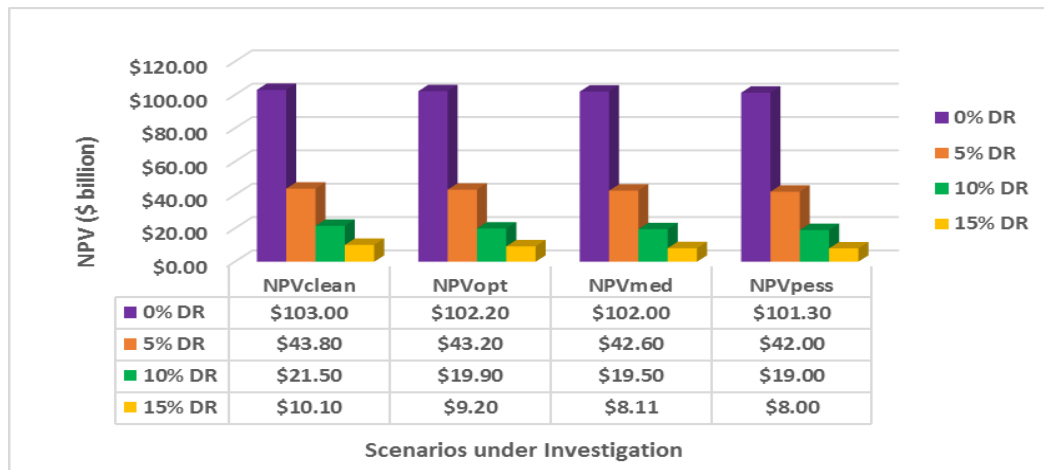


Figure 7-27 The influence of engine leasing on NPV for ROH

7.8 Compressor Washing: Maintenance Action –Section 2

7.8.1 Overview

Application of compressor washing for a mechanical drive on pipeline application is scarcely seen in the literature, especially an operation when the engine power is fixed throughout the operation. The main aim of this section is to investigate the impacts of on-line compressor washing for natural gas pipeline application at a fixed power.

Among the various types of compressor washing, on-line compressor washing was adopted in this research. The reason was that the gas turbine for the mechanical drive in the oil and gas needs online compressor washing to help continue production and to avoid loss of power requirement for the driven equipment. So, from a practical point of view, a short period of downtime in gas turbine used on pipeline application can result in a loss of money equivalent to, or more than the cost of the GT engines. Based on this reason, the present study considers online washing and the GT engines to run on a constant shaft power irrespective of the condition of operation.

This section considers the impacts of compressor washing on the following areas applicable to natural gas pipeline transportation system:

- (i) Performance improvement through Compressor washing
- (i) Economic enhancement through compressor washing
- (ii) Emission reduction through compressor washing

7.8.2 Methodology

Compressor washing practice in gas turbine plant is proportional to the rate of engine degradation. Figure 7-28 shows the flow chart or methodology for the implementation of the on-line compressor washing taking into account typical operating conditions. Three scenarios (opt. med and pess.) which defined the level of degradation were investigated in comparison to the clean engine condition. Each of these scenarios was investigated in each season of the year using TURBOMATCH at fixed shaft power. The baseline-degraded model in chapter 6 was used to ascertain the level of engine degradation before compressor washing based on each of the scenarios. The engine was model with fixed shaft power irrespective of the ambient temperature and degradation using TURBOMATCH. This setting implies that the engine will be operating at the expense of the TET and fuel flow. From the flow chart, TET, fuel flow, mass flow and pressure ratio are the performance parameters to be achieved before and after compressor washing

As the rate of degradation increases and becomes the worst, the frequency of washing (washing interval) is expected to increase also. Similarly, when the degradation rate is less, the washing frequency is less [23]. Therefore, it was assumed that as the degradation increases from optimistic to pessimistic, the volume of washing fluid (demineralised water and the concentrate) also increases accordingly. 5% and 10% increase in washing fluid above the optimistic scenario were used for the medium and pessimistic scenario, respectively. The reason for the assumption was that, as the level of degradation increases, it is expected that the washing fluid increase also to achieve the performance recovery in all the scenarios.

Furthermore, the trend line equation to determine the volume of concentrate and demineralised water by Musa [23] was utilised in this thesis. The trend line makes use of the mass flow from the engine to ascertain the volume of washing fluid to be implemented at different scenarios under investigation.

The assumed washing time interval was twice every week, while the recovery rate was assumed to be 50%. The assumption of the 50% recovery was due to the assumption of continual usage of the engine on the natural gas pipeline. In this case, power loss was not an issue rather TET and fuel flow. So, implementing on-line compressor washing for pipeline application at fixed shaft power will improve on the

engine life (TET) and fuel flow. The 50% recovery rate was implemented on the degraded model to evaluate the impact on the performance parameters and subsequently the economics and the emission. The performance result before and after compressor washing was used in the developed economic model to evaluate the impacts of compressor washing on the economics and emission

It is important to state here that the compressor washing investigation was done in only one compressor station (Station 18), and it was assumed that the same approach could be replicated in all the compressor station. The engine capacity for this investigation was IGTA 13.4 MW

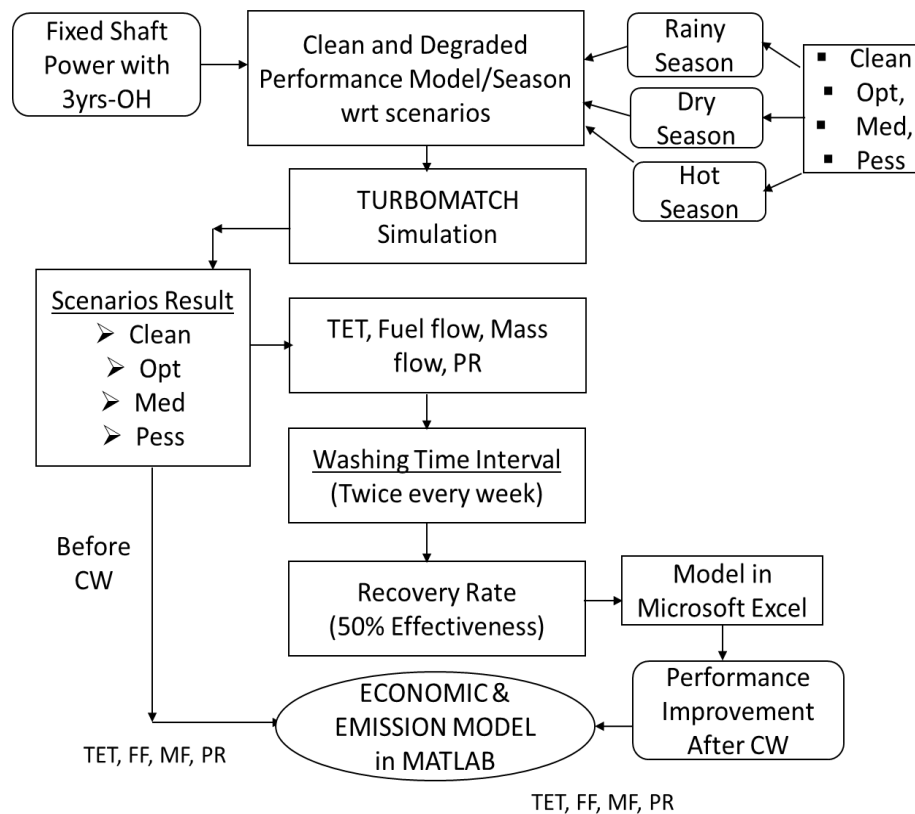


Figure 7-28 Methodology for compressor washing

7.8.3 Performance Improvement through compressor washing

Figure 7-29 to Figure 7-32 shows the performance recovery after on-line compressor washing. The result depicts the pessimistic scenario of the winter season of the first year. Meanwhile, the same method was implemented in all the years except the year of engine overhaul. The first year of the engine operation in the rainy season had

2.25% of engine degradation (reduction in flow capacity and isentropic efficiency). As the impact of the degradation varies due to the influence of the time-based ambient temperature for each season, so also the percentage of recovery after compressor washing varies according to the time of the day and from one season to another. Generally, the percentage of increase or decrease of the performance parameters depends on the level of engine degradation. Also, the percentage of performance recovery depends upon knowing the amount of performance lost due to engine degradation

Figure 7-29 showed the mass flow recovery after compressor washing. For instance, at 9:00, which corresponds with the ambient temperature of 21.42°C, a mass flow of 1.101kg/s was lost due to degradation when compared with the clean condition, but after compressor washing, 0.549kg/s was recovered to improve on the mass flow. Meanwhile, the non-linearity in the recovered mass flow shows the influence of the time-based ambient temperature. The time-based ambient temperature indicates the variation in the ambient temperature according to the time of the day. Figure 7-30 shows the influence of the compressor washing on the engine TET. The TET continue to be high as the time of operation continues. The reason was that the shaft power of the engine was fixed, and the continuous operation of the engine was at the expense of the engine TET and fuel flow. Considering the engine operation at 9:00 that corresponds with the ambient temperature of 21.42°C, the TET increased by 38.9K due to degradation when compared with the clean condition. However, after compressor washing, the TET was reduced by 19.8k from the degraded condition. The variation in the increase in TET indicates the influence of the time-based ambient temperature.

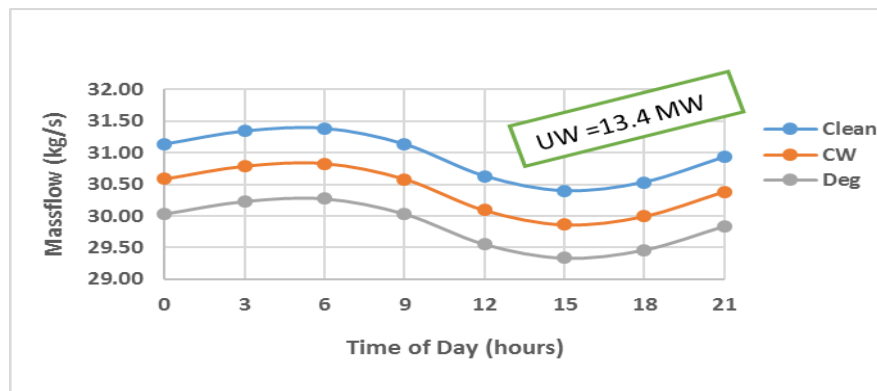


Figure 7-29 Massflow recovery through CW

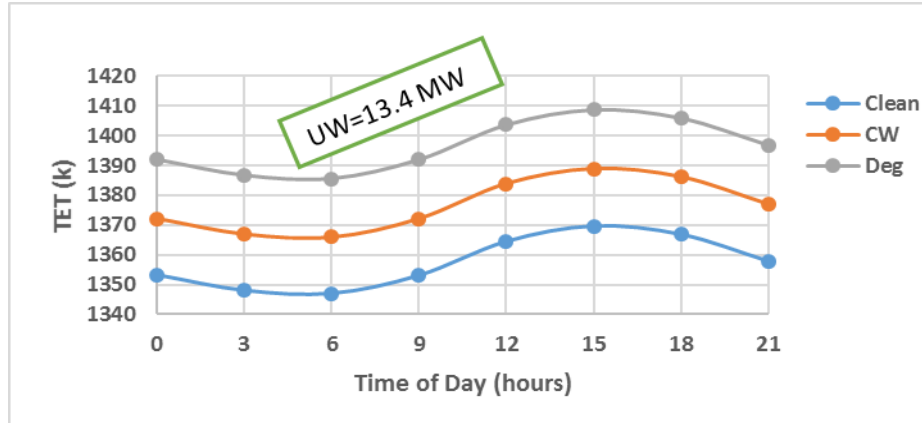


Figure 7-30 TET reduction through CW

The fuel flow of the engine increases as engine degradation continues with time. The percentage of increase depends on the level of engine degradation. Also, the percentage of fuel consumption recovery depends upon knowing the amount of fuel consumes as the engine degrades. For instance, Figure 7-31 shows the impacts of compressor washing in fuel flow reduction. From the curve, at 9:00, which corresponds with 21.42°C, the fuel flow increased by 0.01kg/s due to degradation when compared with the clean condition. However, after compressor washing, 0.003kg/s of fuel flow was recovered from the increased value from the degraded state. Similarly, The non-linearity in the fuel flow indicates the influence of the time-based ambient temperature. Finally, the pressure ratio (PR) of the engine reduces as engine degradation continues with time. The percentage of reduction depends on the level of engine degradation. While the extent of recovery depends upon knowing, how much of the PR is lost. From the curve of Figure 7-32 which shows the increase in PR after compressor washing, at 9:00, which corresponds with 21.42°C, the PR reduced by 0.05 due to degradation when compared with the clean condition. However, after compressor washing, 0.025 was recovered to improve on the PR from the value due to degradation. The non-linearity in the PR shows the influence of the time-based ambient temperature

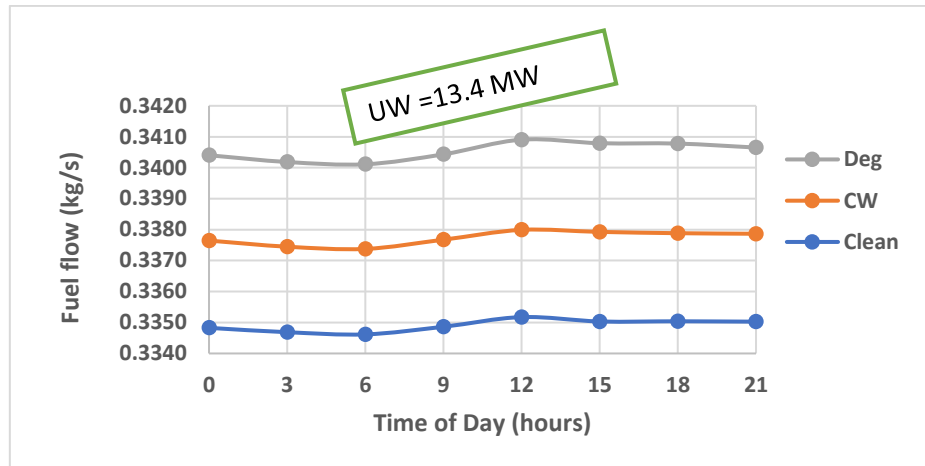


Figure 7-31 Fuelflow reduction through CW

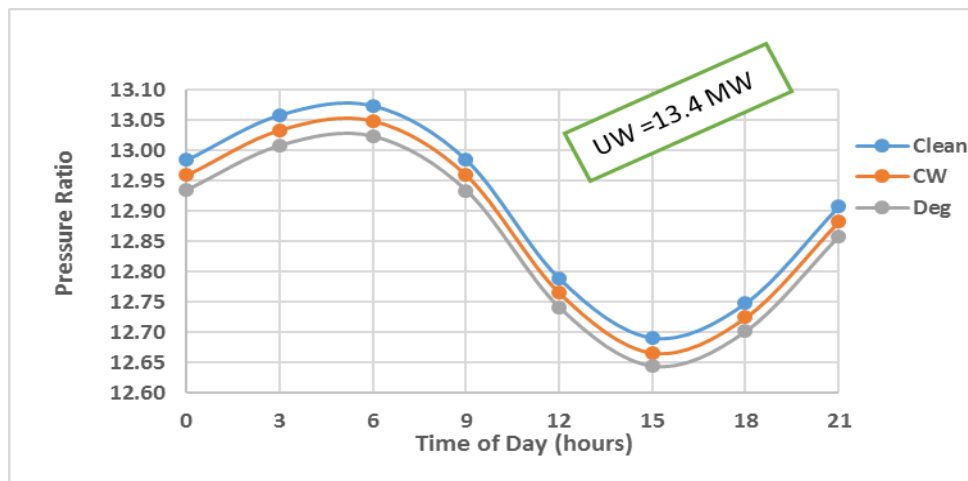


Figure 7-32 Improvement in the PR after CW

7.8.4 Economic Enhancement after Compressor Washing

The total cost of washing with respect to the frequency and recovery rate has been implemented in this research. Net profit after washing cost has been deducted, and the benefits due to TET and fuel recovery are analysed.

With the understanding that the benefits of on-line washing can be site-specific, cautious assumptions were made to provide outcomes that are more attainable in most operations. Table 7-1 represents the economic assumption implemented to evaluate the economic benefits of compressor washing. Also, The developed baseline economic model was used to evaluate the economic benefits of compressor washing using the performance parameters before and after compressor washing

Table 7-1 Economic assumptions for compressor washing

S/No	Assumed condition	Values
i	Time per wash	10 minutes
ii	Mixture (Fluid: water)	1.4
iii	Percentage of water to air	0.02%
iv	Cost of washing fluid (concentrate)	\$3.9/liter
v	Cost of demineralised water	\$0.065/liter
vi	No washing at the year of overhaul	Null
	Beginning of Production year	Year 1

7.8.4.1 Estimation of Washing fluid for Project life

The amount of liquid used for the washing cycle was based on the water-to-air ratio by mass flow. The demineralised water (DH20) and concentrate (CONC) were calculated based on the stated assumption in Table 7-1, engine mass flow of 36.64kg/s and the engine capacity of 13.4MW. The knowledge of the volume of the washing fluid will help in calculating the cost associated with it. The compressor station under consideration has five gas turbine (GT) engines throughout the project life. The trendline polynomial equation as proposed by Musa[23] for the evaluation of DH20 and CONC for compressor washing was used to ascertain the amount of DH20 and CONC for the GT engines compressor washing.

Figure 7-33 and Figure 7-34 shows the annual and total volume of both DH20 and CONC used for the washing period, respectively. The evaluation was done for the various scenarios under consideration. The result shows that 175.9, 184.7 and 193.5 litres of DH2O were used per wash for opt, med and pessi, respectively. Correspondingly, 43.97, 46.17 and 48.37 litres of CONC were used per wash for opt, med, and pess, respectively. The variation in the result per wash for the various scenarios was because of the assumption that 5% and 10% increase for med and pess scenario, respective in washing fluid above the opt scenario was considered. The reason for implementing the assumption was that the engine degraded condition increases from the opt (normal case) to med (middle) and finally to pess (worst case) scenario. So, the assumption was made because it was believed that the volume of DH20 and CONC to be used should be proportional to the level of degradation in order to achieve engine effectiveness.

The further result shows that an annual volume of 18342.5 litres 19259.63 litres and 20176.75 litres of DH20 for opt, med, and pess, respectively were used. Also, the

annual volume of CONC was 4585.23 litres, 4814.5litres and 5043.76 litres for opt, med and pess scenario, respectively. These quantities were constant throughout the year, as shown on Figure 7-33

Generally, the volume of DH20 was higher than the CONC. Meanwhile, there was no washing at the year of engine overhaul, and therefore, no washing fluid was evaluated. Figure 7-34 shows the total volume of DH20 and CONC for project life. The result affirmed that the volume of washing fluid for the med and pess were 5% and 10%, respectively higher than the opt scenario.

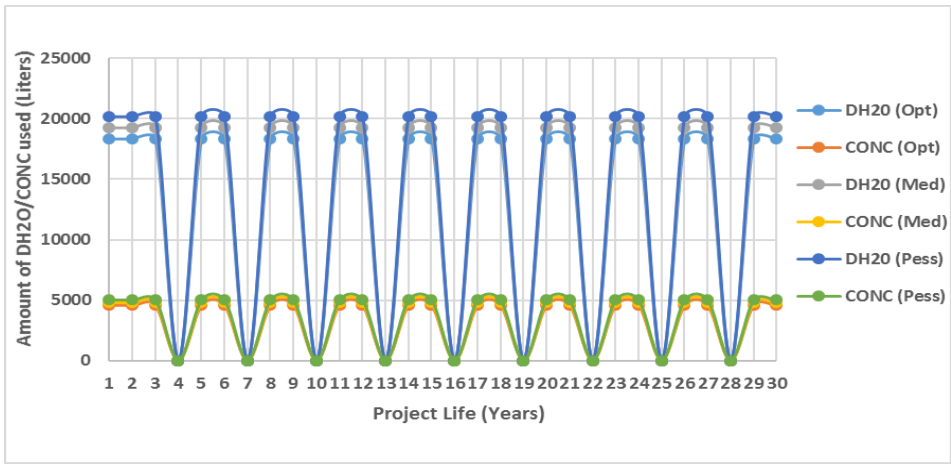


Figure 7-33 Annual litres of DH20 and CONC used for the washing period

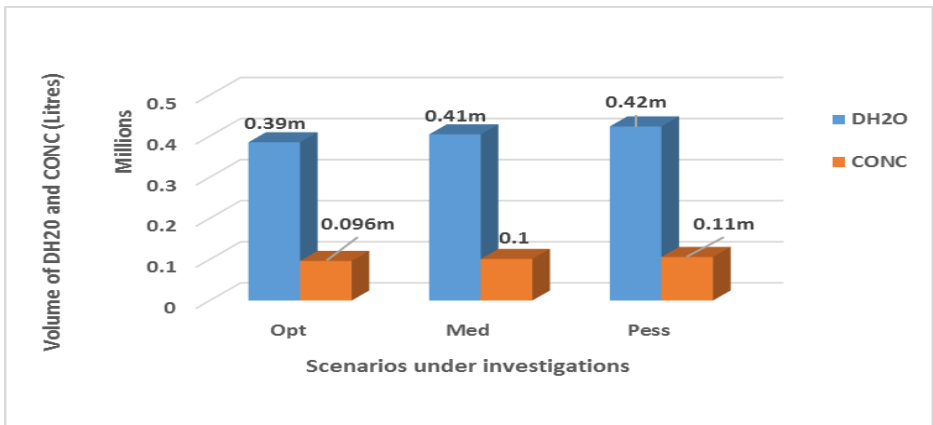


Figure 7-34 Total litres of DH20 and CONC used for project life

7.8.4.2 Estimation of cost of Washing

The washing cost was hinged on the volume of DH20 and CONC used per wash. The costs of the washing fluid (DH20 and CONC) per wash and the annual cost for the 13.4 MW is shown in Table 7-2. Meanwhile, the total costs of the washing fluid

are represented in Figure 7-35. The result shows that the cost for the opt scenario was \$0.4 million while the med and pess scenarios have 5% and 10%, respectively higher than the opt scenario. The variation in the cost of the various scenarios is an indication that the more an engine degrades, the higher the cost of washing.

Table 7-2 Cost of washing fluid

	Cost of washing fluid per wash		Annual cost of washing fluid	
	CONC (\$)	DH20 (\$)	CONC (\$)	DH20 (\$)
Opt	171.48	11.43	17882.41	1192.26
Med	180.05	12.00	18776.53	1251.88
Pess	188.62	12.58	19670.66	1311.49

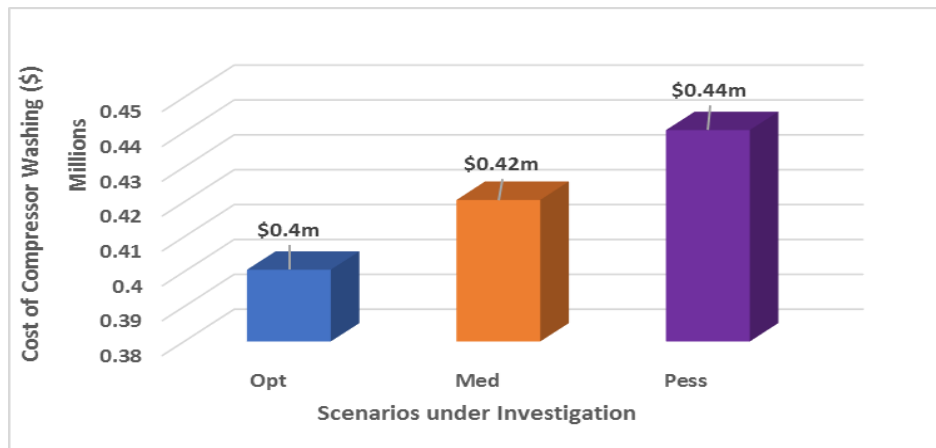


Figure 7-35 Total costs of washing

7.8.4.3 Evaluation of Fuel cost after compressor Washing

Having known the fuel flow before and after compressor washing using TURBOMATCH; the developed baseline economic model was used to evaluate the fuel cost in each season of the year with respect to the various scenario.

The time-based fuel cost after compressor washing for the various scenarios in each season of the year throughout the project life has been analysed. The addition of the various scenarios with respect to their respective season of the year makes up the fuel cost for each season of the year. Figure 7-36, Figure 7-37 and Figure 7-38 show the annual fuel cost for the rainy, dry and hot season after compressor washing

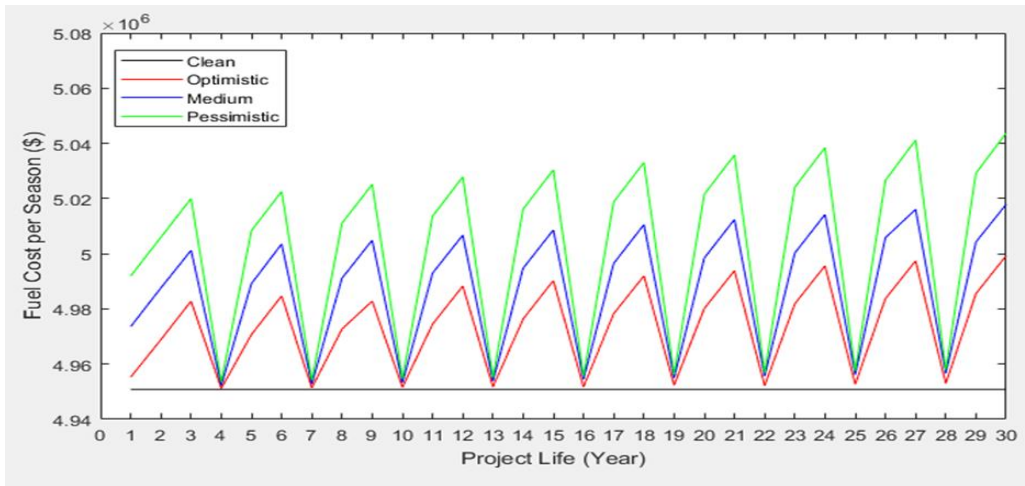


Figure 7-36 Annual fuel cost after compressor washing (rainy season)

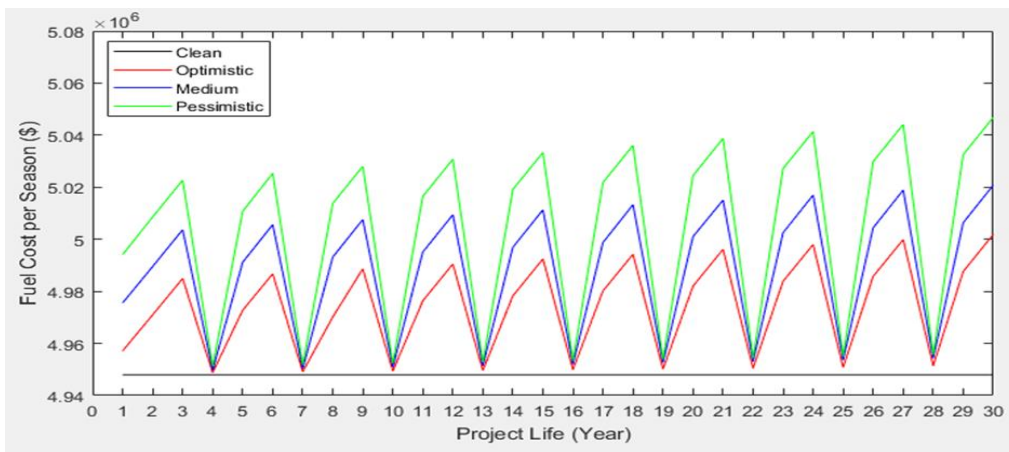


Figure 7-37 Annual fuel cost after compressor washing (dry season)

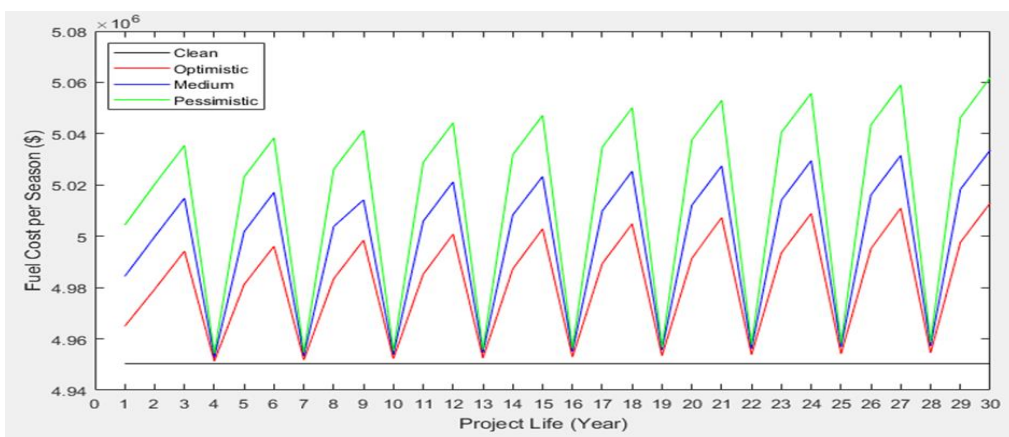


Figure 7-38 Annual fuel cost after compressor washing (hot season)

The addition of the three-season fuel cost with respect to their various scenario gives the annual fuel cost. Figure 7-39 and Figure 7-40 showed the annual fuel costs before and after compressor washing, respectively. Both results showed an annual fuel cost of \$14.85 million. Generally, the result showed a reduction in fuel cost due to the influence of recovered degradation after compressor washing. Figure 7-41 shows the total fuel cost after compressor washing in comparison with the baseline case. The baseline here means the total fuel cost before compressor washing. The result shows that the total baseline fuel cost for the clean, opt, med and pess scenario are \$445.4, \$450.4, \$453 and \$457 million, respectively. The total fuel cost after compressor washing shows that the opt, med and pess reduces by 0.6%, 0.9% and 1.42% when compared with the baseline cost. Meanwhile, the cost for the clean condition remain the same as there was no compressor washing in the clean engine condition

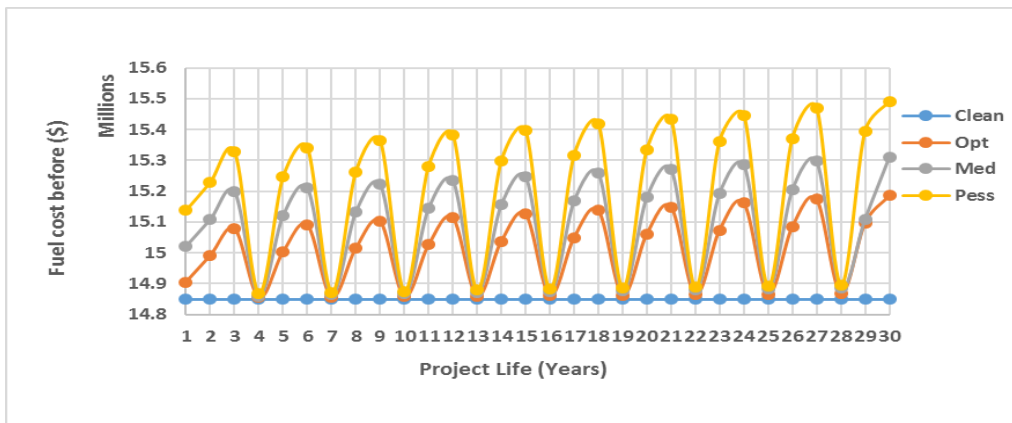


Figure 7-39 Annual fuel cost before compressor washing

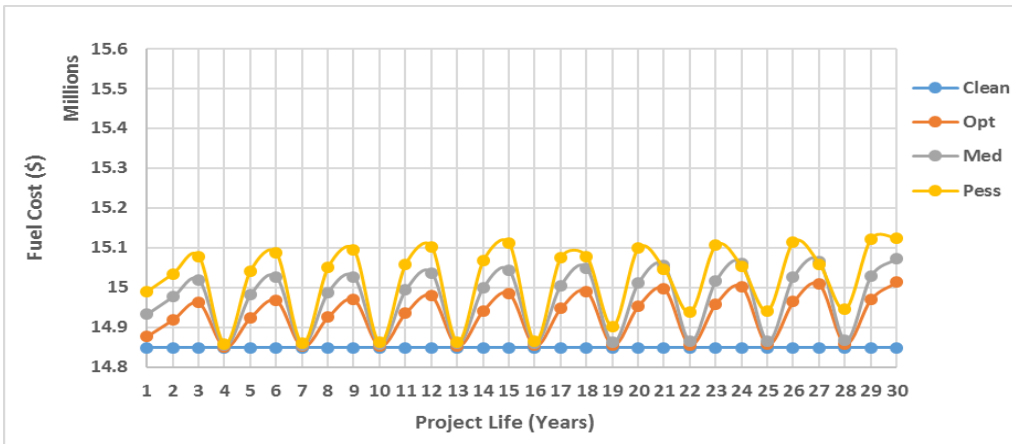


Figure 7-40 Annual fuel cost after compressor washing

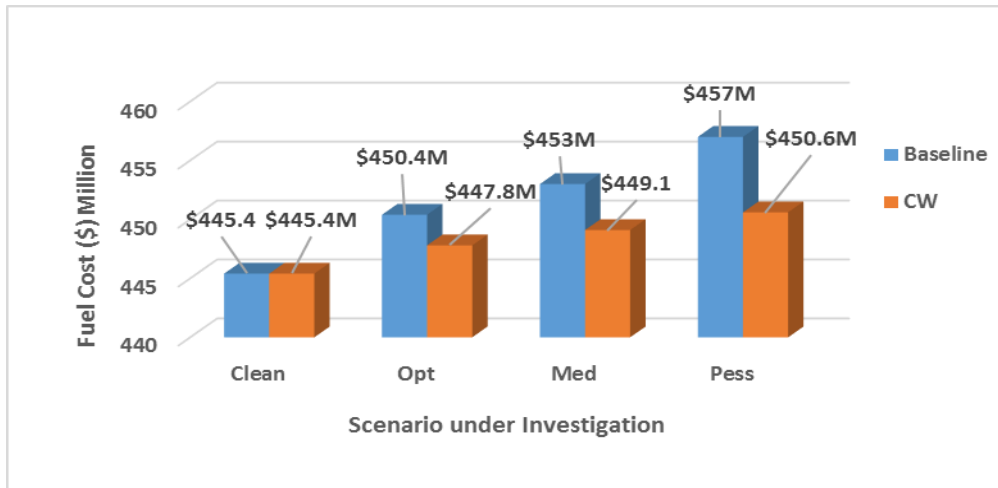


Figure 7-41 Total fuel cost after compressor washing with the baseline

7.8.4.4 Evaluation of Operation and Maintenance (O&M) cost under ROH

The O&M cost of the engines becomes necessary to analyse the benefit of compressor washing on investment. The method used in chapter eight for the baseline case was adopted here to evaluate the O&M cost. The only difference was the addition of compressor washing cost. Figure 7-42 showed the O&M cost before and after compressor washing (CW). The result revealed that there was a little variation in the O&M cost between the opt, med, and pess scenario for the baseline case to a value of \$90.46, \$90.8 and \$96.6 million, respectively. However, the result after compressor washing showed that the opt, med, and pess scenario had 0.07%, 0.4% and 5.6% reduction in maintenance cost when compared with the cost of the baseline case.

The result is in line with the fact that an increase in degradation result to increase in the engine TET vice versa. As the TET increases, the life of the engine reduces also. So, the cost increase from opt to pess scenario in the baseline case was as a result of the variation in the TET due to the fixed shaft power as the level of degradation increases. This makes the life of the engine at each scenario to varies. However, after compressor washing, there was a considerable improvement in the engine TET, which directly improves the engine creep life. Hence, the model calculation maintained a constant creep life of 25,000 hours for all the scenarios, thereby making the O&M cost after the compressor washing to be slightly low when compared with the baseline case.

It is important to state that increase or reduction in TET reduces or increases the engine creep life, respectively. Hence, the reduction or increase in the engine creep life reduces or increases the O&M cost, respectively.

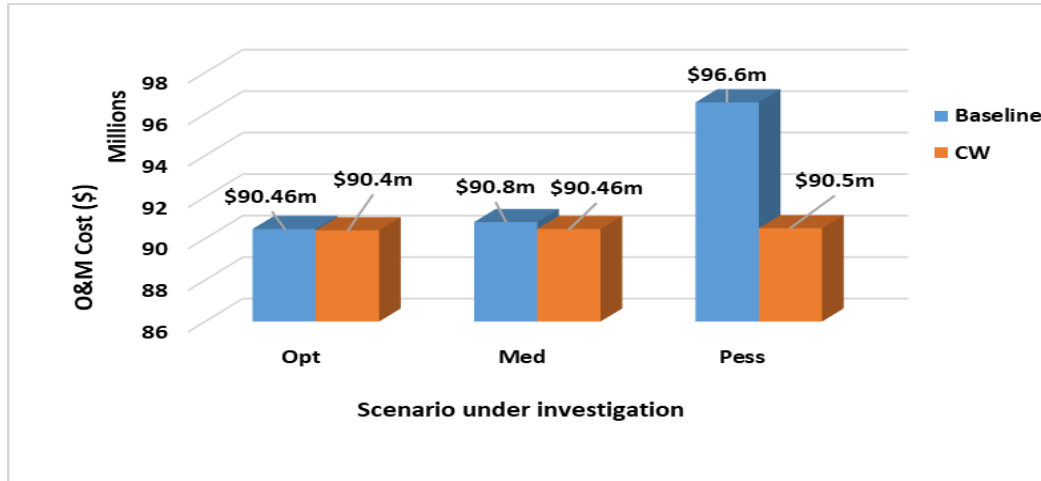


Figure 7-42 Total O&M cost after compressor washing with the baseline case

There is a need to evaluate the impacts of compressor washing on emission reduction to achieve the real benefits of applying the online-compressor washing to the pipeline project under consideration. The result will determine the emission tax on the amount of CO₂ generated before and after compressor washing.

7.8.5 Emission reduction through compressor washing

7.8.5.1 Overview

All prime mover, including gas turbine engines, generates emission during operations. These emissions range from one type to another and are caused by different sources with their corresponding health and environmental impacts. Some researchers have looked into several ways of reducing emission in gas turbine operation. Tian-gang and Xiao-Yun [184], investigated “new engine core configuration with heat management and an active system as well as advance combustor technology” to reduce NO_x formation during engine operation, Jung and Becker [185] introduce Platinum-Rhodium catalysts for carbon monoxide and hydrocarbon removal which decreases the removal of sulphur trioxide. Toyoaki et al. [186] explained the fuel conversion method into hydrogen-rich fuel gases with less CO₂ emissions. This method aimed to reduce CO₂ emission. Research has not been explored in the application of compressor washing in GT emission reduction. Musa

[23] in his on-line and off-line compressor washing combination recommended investigating the impacts of compressor washing in emission reduction. Therefore, this section presents the impacts of compressor washing in reducing gas turbine emission and CO₂ tax on the natural gas pipeline network.

The method outlined in Figure 7-28 was used to deduce the performance parameters after compressor washing to evaluate the emission before and after compressor washing. The performance results before and after compressor washing were connected to the developed emission model to quantify the emission before and after compressor washing. It should be noted that the percentage of engine degradation for the baseline case was also implemented for the degraded model with 3 years overhaul interval

7.8.5.2 Result and Discussion

The analysis of the time-based emission for the various scenarios was done in each season of the year. The addition of the emission in the three seasons made up the annual emission generated by the GT engines. Figure 7-43 and Figure 7-44 shows the annual CO₂ emission and emission tax, respectively, throughout the project life. Although the level of degradation in each year varies from one another, the result generally shows a reduction in the generated emission after compressor washing as a result of the reduction in the CO₂ emission due to TET reduction, further result shows that the annual emission tax reduces between 5.3% and 7.6% within the degraded scenarios.



Figure 7-43 Annual CO₂ emission after compressor washing

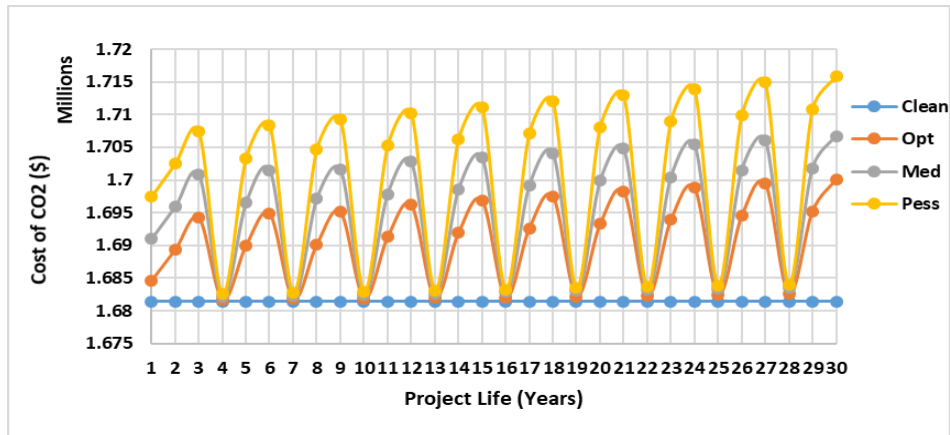


Figure 7-44 Annual CO₂ cost after compressor washing

Figure 7-45 shows the annual NO_x emission. The NO_x increases with an increase in degradation due to the increasing TET. The general trend of reduction was observed in all the generated emission based on the present operating condition of the engine after compressor washing.

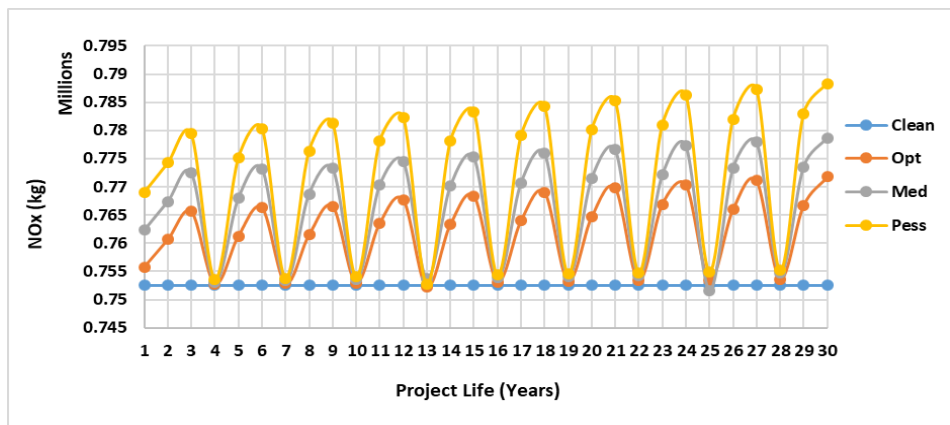


Figure 7-45 Annual NO_x emission after compressor washing

The comparison between the total emissions and CO₂ tax before and after compressor washing are hereby presented. Figure 7-46 shows the total CO₂ emission before and after compressor washing the result shows that the opt., med., and pess scenarios experienced a reduction of 5.8%, 6.1% and 6.5% when compared with the baseline condition. Similarly, the CO₂ tax in Figure 7-47 also experiences a reduction of the various scenarios within the same range.

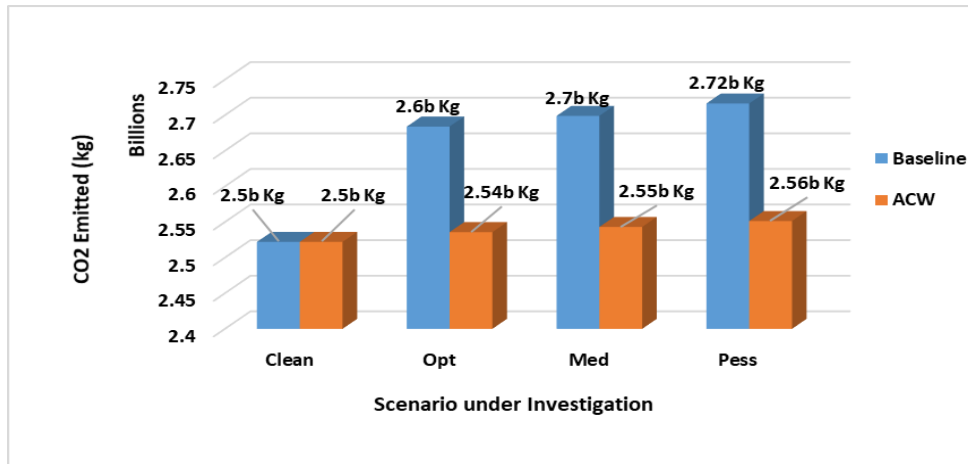


Figure 7-46 Total CO2 emission before and after compressor washing

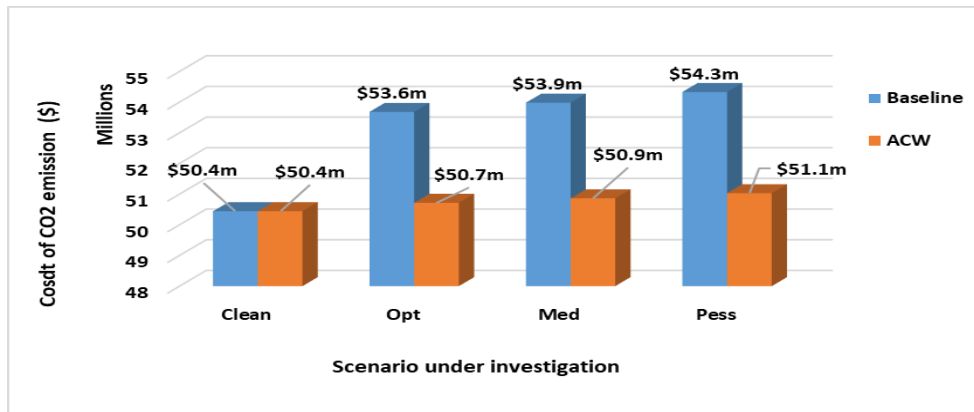


Figure 7-47 Total Cost of CO2 emission before and after compressor washing

Figure 7-48 represents the total NOx emission before, and after compressor washing. The result showed that the clean engine had 22.6 million kg of NOx generated for thirty years of engine operation applied to the condition before and after compressor washing. A further result showed that the opt, med, and pess had 2.6%, 4% and 5.53%, respectively reduction after compressor washing when compared to the baseline condition. The result is expected because, after compressor washing, the high TET and fuel consumption of the engine reduces while maintaining the same power for the gas compressor operation.

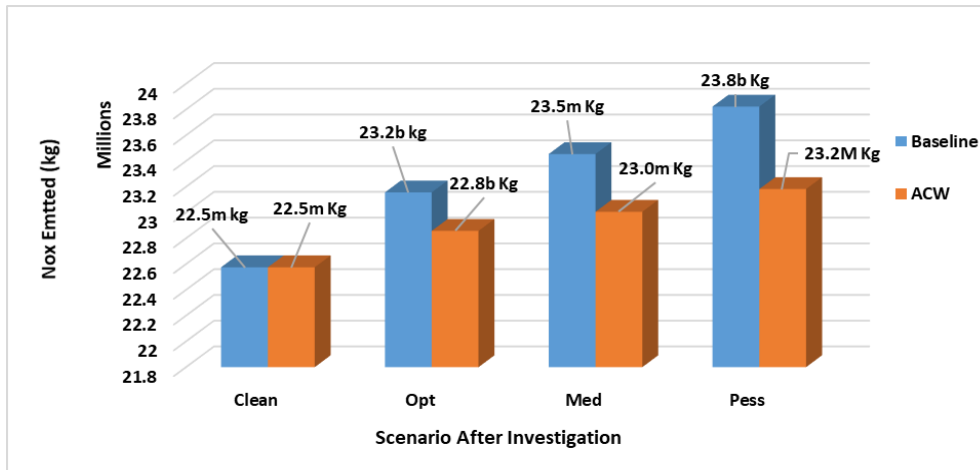


Figure 7-48 Total NOx emission before and after compressor washing

7.8.6 The Influence of Compressor Washing on the NPV

The net present value was the appraisal method to evaluate the economic viability of the pipeline project. As explained in chapter 8, the NPV was used in this section to investigate the economic viability of applying compressor washing on the pipeline investment. However, in this analysis, the divestment and leasing term was not included. The reason was that there was no engine divestment as the compressor station used 5 GTs throughout the project life irrespective of the engine operating condition. In concurrence with the method for NPV evaluation in chapter 8, Figure 7-49 shows the modified version of the method to evaluate the influence of compressor washing in NPV calculation. It is important to state here that the NPV analysis was done for compressor station 18 only. As represented on the methodological framework, the capital cost will comprise of the cost of gas compressor, gas turbine, pipeline, and the accessories. Meanwhile, the annual maintenance cost, which is an integral part of the annual cash flow comprises of the cost of fuel, engine maintenance, emission tax and the cost of compressor washing. The source of revenue remains gas sales.

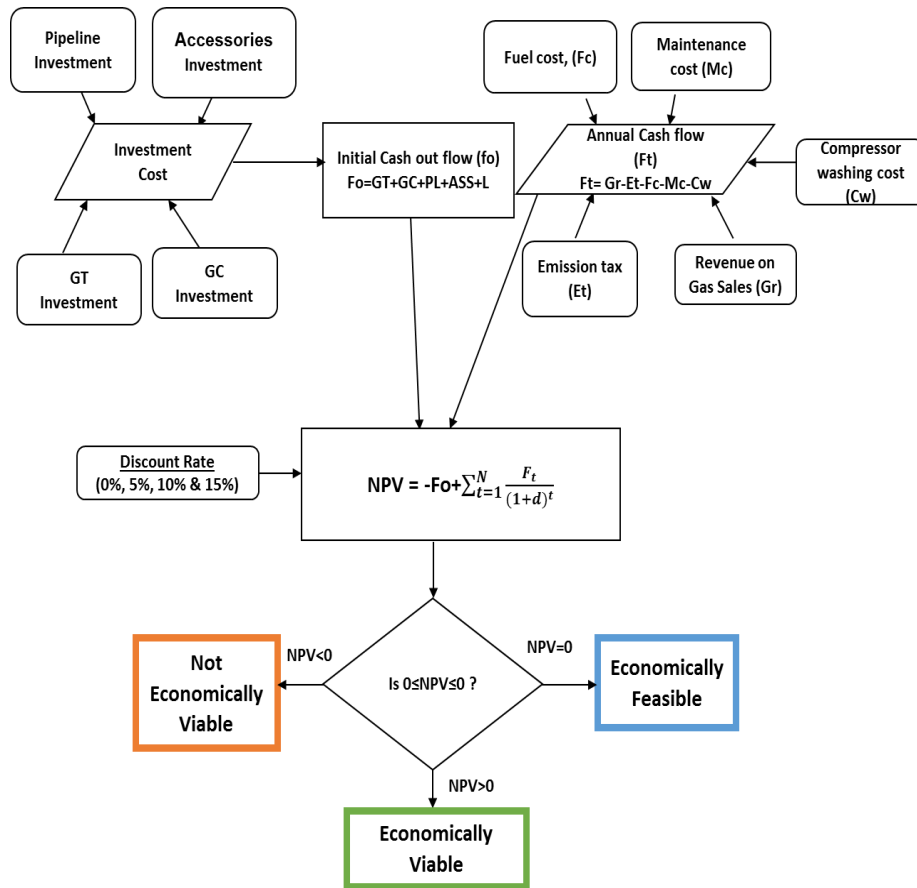


Figure 7-49 Method for NPV calculation in compressor washing

The same approach of calculating the GC and GT was also applied in this section. The total volume of gas (30bcm) to be delivered annually was divided by the total number of the compressor station to ascertain the revenue to be generated in this station. Meanwhile, Table 7-3 shows the components of the station under investigation. Also, the feature of the pipeline remains the same. 4 GCs was used with 5GTs.in this station. The reason was that, during the simulation process, using GTs to 4 GCs could not satisfy the condition of operation (the operational TET should not exceed 60⁰c above the engine DP). So using 5GT satisfactorily operate the GCs without any additional engine throughout the project life (30 years).

Table 7-3 Component of Compressor Station 18

Components	Values
GT engine (IGTA)	13.4
GT gas fuel price (\$/kg)	0.168
Cost of GT (\$/KW)	566.75
GC cost per kw(\$/kw)	3000
Diameter(Inch)	56"
thickness (inch)	0.812
Pipe Distance (km)	190

7.8.7 Result and Discussion

Figure 7-50 shows the capital cost of the station under investigation. Combining the capital cost, cost of washing, fuel cost and O&M cost will enable a successful evaluation of the NPV after compressor washing. Figure 7-51 and Figure 7-52 shows the NPV before and after compressor washing, respectively. The result generally shows that the NPV before compressor washing was lower than the NPV after compressor washing. The reason was that NPV reduces with increase in degradation, but the NPV increases after compressor washing, indicating the reduction in degradation after compressor washing. Also, taking the discount rate into account, the NPV reduces significantly as the discount rate increases. For instance at baseline condition, 0%, 5%, 10% and 15% DR for the clean condition was \$6.49, \$3.0, \$ 1.47 and \$ 0.81 billion. The same trend was also achieved in all the scenarios. Further result before and after compressor washing revealed the benefits after compressor washing. For instance, at 5% DR, 6%, 5.2% and 4.8% increase in the NPV for opt, med., and pess respectively was achieved when compared with the NPV before compressor washing at the same DR This implies that \$ 90, \$ 81 and \$72 million was recovered after compressor washing. In all the scenarios, there were benefits of implementing compressor washing for the pipeline business.,

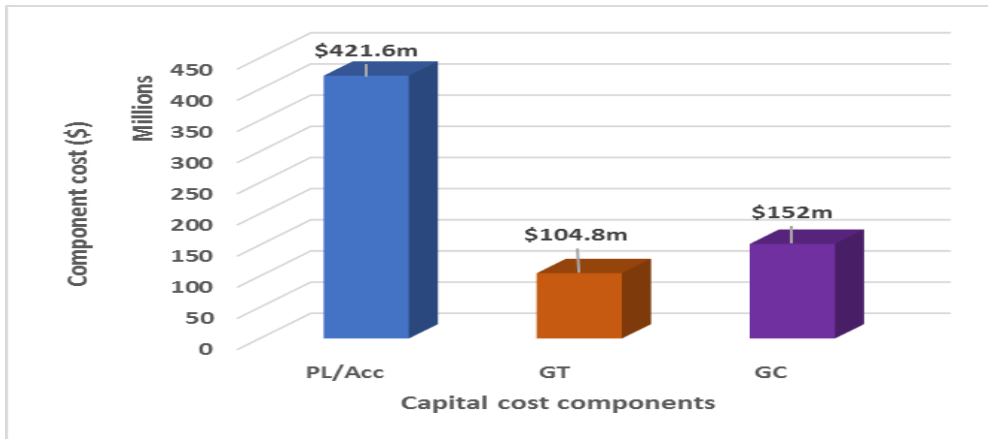


Figure 7-50 Total capital cost of the compressor station components

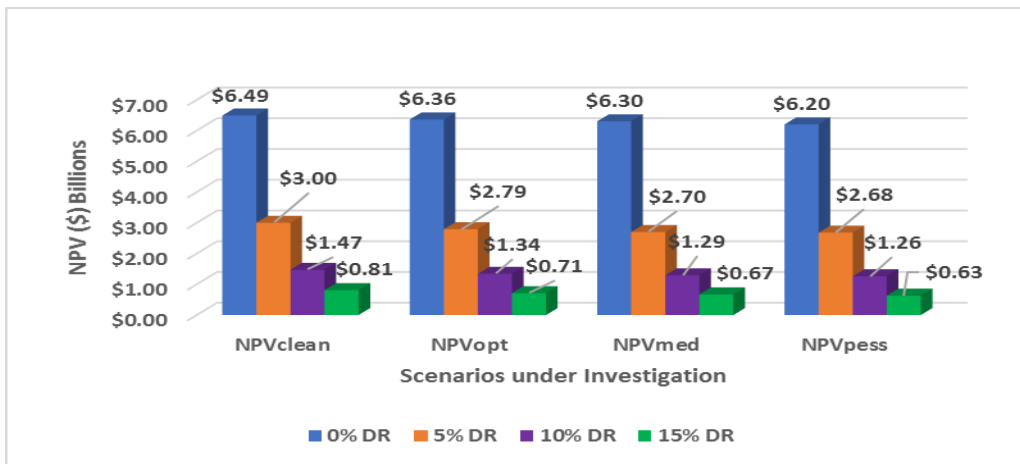


Figure 7-51 NPV before compressor washing

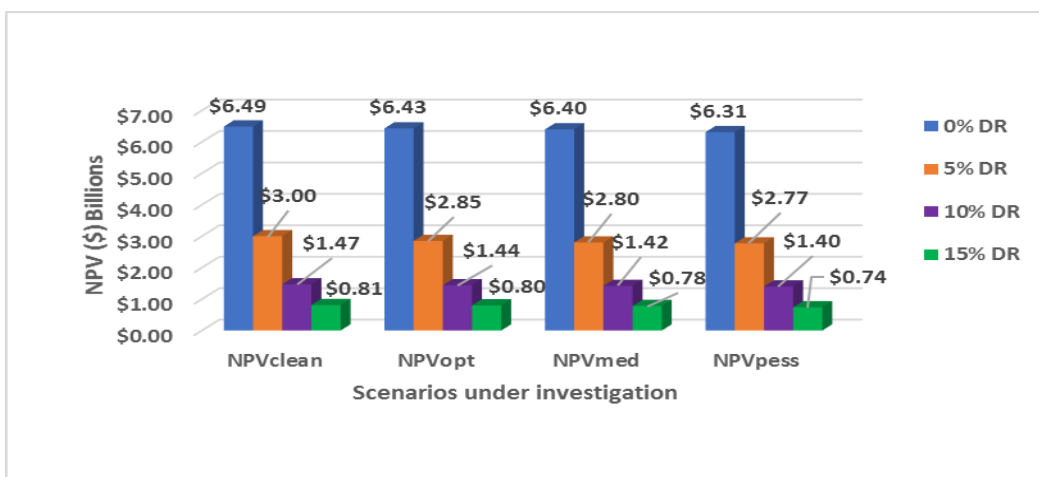


Figure 7-52 NPV after compressor washing

Table 7-4 Benefits of compressor washing on the NPV

Discount Rate	NPVopt (million)	NPVmed (million)	NPVpess (million)
0% DR	\$110	\$100	\$90
5% DR	\$90	\$80	\$70
10% DR	\$80	\$70	\$60
15% DR	\$70	\$60	\$50

8 CONCLUSIONS AND RECOMMENDATION FOR FUTURE WORKS

8.1 Conclusions

In summary, this research has given an account of the isothermal, non-isothermal performance, maintenance action and the application of TERA on natural gas pipeline transportation system.

The main contributions of this work include:

- A detailed economic analysis of the natural gas pipeline project that takes into account the annual engine degradation, pipeline/accessories and time-based ambient conditions using an NPV method that incorporates engine divestment and leasing for the investigations
- The predicted impacts of gas turbine rescheduled-overhaul on the engine performance in terms of TET reduction, fuel flow reduction, creep life improvement and the overall investment cost and benefit for a pipeline project at fixed GT power operation.
- Determination of the number of GT to be used in each compressor station at a controlled TET under the influence of degradation
- The investigated impacts of gas turbine compressor washing on NO_x, CO₂ emission reduction and the overall economic enhancement of the pipeline project.
- The predicted pipe exit pressure that could determine the volume of natural gas delivery at varying ambient temperature conditions
- the evaluated gas compressor power in a non-isothermal condition that takes into account the combined effects of pipeline elevation, mechanical losses, and the desired volumetric flow of the compressor

❖ Pipeline and Gas Compressor Module

The result obtained from the pipeline investigation showed that an increase in pipe exit pressure at a particular time of day resulted in a corresponding increase in the

amount of gas flow. The result also showed that for every 1% increase in pipe exit pressure resulted in a 1.8% increase in the volume of the gas flow in the pipeline. The result further revealed that 1550mm pipe diameter could conveniently deliver 30.2bcm/year within an exit pressure of 2400kpa and 4800kpa when the gas temperature is between 295.61K and 302.84K. However, if the pipeline operates at an environment below this temperature, the pipe exit pressure will be higher.

The obtained result from the gas compressor model indicated that the power of the centrifugal compressor increased with an increase in gas temperature and varied from one station to another due to the different time-based ambient temperature of the various CS. The investigation also revealed that for every 1% rise in the gas temperature resulted in a 3.4% rise in the power required by the gas compressor to move the gas. The further result revealed that a gas compressor with high inlet temperature and pressure ratio required a higher polytropic head and power to transport the gas from one location to another. The effects of varying the gas temperature on the polytropic head and gas compressor power on the map for a constant volumetric flow showed that the change in gas temperature led to a different operating point on the map, unlike other studies that assumed a constant gas temperature thereby obtaining a single operating point.

❖ **Performance module with a fixed shaft power at a controlled TET**

The simulation of the developed engine model shows a good correlation with actual engine data from the OEM with 99% accuracy[187], thus indicating the suitability of the engine model performance prediction. For a fixed power operation, the result shows that within an ambient temperature of 17.45°C to 19.95°C and with 2.65% degradation (reduction in flow capacity and isentropic efficiency of the turbomachinery) the TET and fuel flow increases by 6.04% and 6.3% respectively when compared with the clean condition. This highlights the need for performance augmentation to improve the life of the engine by reducing the TET and fuel flow. In such environments, the need for on-line compressor washing to reduce the fouling effects would be beneficial so that the TET and fuel flow can be reduced.

Furthermore, the development of the degraded model based on the different scenarios (optimistic, medium and pessimistic) for 30 years of operation amid time-based ambient temperature was achieved. The result depicted that a yearly increase

in engine degradation influenced the number of GT engines to be used in each compressor station (CS) based on the constraint on the engine operational TET. Hence, the TET profile map that matched with the power setting of the number of gas turbine engine used in each CS was produced. Based on the produced TET profile map, the result obtained showed that the clean, optimistic (opt), medium (med) and pessimistic (pess) scenarios used 90, 99, 106 and 120 mixed fleet of GTs, respectively for the baseline condition for the entire project life across the 18 CS. The method established in reducing the engine TET at fixed shaft power to arrive at the exact number of GT for each degraded scenarios will enhance engine life-extension strategies, which enables cost-effective management of the engines by engine life-cycle managers or natural gas pipeline investors for a safe and reliable pipeline business.

❖ **The developed emission model for the natural gas pipeline project**

An emission model to quantify gas turbine emission (NO_x and CO_2) generated and CO_2 tax using empirical and thermodynamic equation has been developed.

The result revealed that:

- for every 1% degradation (reduction in flow capacity and efficiency of the turbomachinery) at a constant power of engine operation, between an ambient temperature of 16.2°C and 29°C , CO_2 emission increased between 0.71% and 0.78% when compared with the clean engine operation. This is also applicable to the CO_2 tax
- the clean condition produced 73.4 billion kg of CO_2 for the entire 18 CS for 30 years of operation. However, the opt, med and pess scenarios produced 1.4%, 3% and 6.6%, respectively higher when compared with the clean condition. The result showed the impact of gas turbine degradation on CO_2 emission production at fixed power operation.
- the total cost of CO_2 throughout the project life for the clean condition was \$1.46 billion while the opt, med and pess scenarios were 2.7%, 4.8% %, and 8.2%, respectively higher when compared with the emission tax of the clean condition

- generally, NOx emission increased with an increase in engine degradation at a constant shaft power operation and constant number of engines. However, running the engines on part-load operation when the number of gas turbine increases, reduces the NOx emission as a result of running the engines on lower TET
- for every 1% engine degradation within an operational ambient temperature of 16.2°C and 29°C, the NOx emission increased between 1.66% and 1.8%

the emission generated at the various CS varied from one station to another due to different ambient conditions, engine power settings and the number of engines used. For instance, the result of NOx emitted in compressor station 4 for the winter season as represented in Figure I- 4Appendix, I showed that the NOx emission was higher at the clean condition of the engine. This happened because of the high power settings of the engines at the clean condition when compared with the lower power settings at the degraded condition, which used a higher number of GT engines operating at a lower TET. The result affirmed that NOx increased with an increase in the power settings.

❖ **The viability of the developed economic model for investing in the natural gas pipeline project**

An economic model and methodology to evaluate the economic implication of using a degraded gas turbine amid time-based ambient temperature for natural gas pipeline investment have been developed. The result obtained revealed that:

- The total CAPEX for the clean condition (Trans-Saharan gas pipeline project) is \$14.67 billion while the opt, med and pess scenarios are 1.8%, 3.21% and 6% higher when compared with the clean condition.
- a 15.2MW engine operating within a degradation value of 0.25% and 0.5% (reduction of flow capacity and isentropic efficiencies of the compressor, compressor turbine and power turbine) within an average temperature of 17.12°C and 28.92°C incurred a fuel cost of \$574.1 per hour.
- the clean engine operation has a total fuel cost of \$12.7billion for the project life while the optimistic, medium and pessimistic scenario has an increase of

3.1%, 4.7%, and 8.7 % respectively when compared with the clean condition, unlike other studies that stopped the analysis at the clean condition.

- the total O&M cost for the clean scenario throughout the project life was \$3.24 billion. However, running the engines at a lower TET for the degraded scenarios reduced the O&M costs by 8.3% and 3.4% for the optimistic and medium scenario, respectively while the pessimistic scenarios increased by 5.7% when compared with the cost at the clean
- the cost of GC, GT, pipe network including all the accessories, fuel, O&M cost and emission tax in the developed TERA model constituted the life cycle cost (LCC)
- the total LCC for the pipeline project at clean engine condition was \$32.01 billion, while the opt, med and pess scenarios were 1.59%, 2.94%, and 6.84%, respectively and higher than the clean condition.
- the NPV at clean engine condition with engine divestment at 10% discount rate (DR) recorded \$21.50 billion while the opt, med and pess scenarios reduced by 8.84%, 14.42% and 20.5 %, respectively when compared with the clean condition
- the NPV with leasing at the same 10% DR, reduced by 8.14%, 12.33% and 13.5% for the respective scenarios when compared with the clean condition

Conclusively, to accommodate all financial risk, 0%, 5%, 10% and 15% DR were used to evaluate the economic viability of investing in the pipeline project. The result obtained in all the DR and scenarios showed that the NPV for engine divestment was 0.2% to 20.3% lower than the NPV for leasing depending on the different scenarios and DR. This revealed that NPV leasing has a better result than that of engine divestment, and as the DR increases, the NPV reduces.

❖ **The impact of GT “re-scheduled overhaul” on the techno-economic of the pipeline network**

The model to investigate the impacts of GT re-scheduled overhaul (ROH) on the TERA assessment of the pipeline project has been successfully developed and presented. The obtained result revealed that:

- the opt, med and pess scenarios that used 99, 106 and 120 number of engines at baseline condition for the 18 CS reduced to 91, 104 and 115, respectively when ROH was implemented for the same number of CS and at the same operating conditions.
- the LCC of the pipeline investment for opt, med and pess, scenarios reduced by 2%, 2.3% and 3.8% when compared with the baseline case.
- NPV with divestment at 10% DR under ROH, gained an increase of 1.3%, 2.2% and 5.3% for opt., med., and pess scenario respectively, when compared with the baseline case.

NPV with leasing at 10% DR under ROH gained an increase of 0.2%, 3.7% and 5.6% higher for opt., med., and pess.scenario when compared with the NPV under engine divestment as stated above.

❖ **The impact of compressor washing on the gas turbine for the natural gas pipeline project**

The clean, degraded, and washed engine model operating with constant shaft power at a controlled TET has been developed and presented. The results showed that

- the application of on-line compressor washing for mechanical drive application at fixed power operation improved the performance of the engine with respect to the TET, fuel flow, mass flow and pressure ratio
- the LCC reduces thereby improving the NPV and the engine availability
- The CO₂ generated for the opt., med., and pess scenarios after compressor washing had a reduction of 5.8%, 6.1% and 6.5% respectively when compared with the baseline condition. The CO₂ tax showed the same trend.
- The NO_x generated for the opt, med, and pess scenarios after compressor washing had a reduction of 2.6%, 4% and 5.53% respectively when compared to the baseline condition. The result was expected because, after compressor washing, the high TET and fuel consumptions of the engine reduced while maintaining the same power for the gas compressor operation for a fixed

number of GT engines. There are no reported works on these contributions as at the time of this research.

- at 5% DR, the NPV for the three scenarios after compressor washing increase by 6%, 5.2% and 4.8%, respectively when compared with the baseline case.

Conclusively, the proposed approach and methodologies in this research will help operators on the real performance implication as well as the potential cost of natural gas pipeline investment, taking into account the different cost element, time-based ambient condition and engine degradation on the natural gas pipeline system. Importantly, the developed model and methodologies can be used as a useful decision-making guide for the Nigerian government to invest in Trans-Saharan gas pipeline project. It can also be applied or adapted to any natural gas pipeline transportation business anywhere in the world

8.2 RECOMMENDATION

The research conducted within the scope and period of this PhD work has not only contributed to the scientific body of knowledge but has identified further knowledge gaps in addressing the future works on the direction of this research. Therefore, the following recommendations are thus stated for future research.

1. The developed TERA model for the use of degraded gas turbine was based on the baseline pipeline design without any alteration on the original design. An optimisation model that will look into the use of degraded gas turbine on pipeline application with the objective of
 - Decreasing the capital cost
 - Decreasing fuel cost
 - Reducing maintenance cost
 - Reducing emission production
 - Reducing emission cost

The optimisation model should also have a view of:

- Varying/reducing the number of the compression station
- Using bigger GT engines. This is also applicable to the non-optimised condition that will run on part-load to deliver the required GC power with the overall view of minimising the project life cycle

cost to enhance better Net Present Value (NPV)

2. The impact of different gas temperature scenarios (isothermal, average of both isothermal and non-isothermal and the average of non-isothermal gas temperatures) and gas composition on the techno-economics assessment of the natural gas pipeline can be investigated. This analysis should cover all the compressor stations as each compressor station has a different time-based ambient temperature.
3. A well-detailed investigation on gas compressor and booster station's equipment failures should be analysed. Similarly, a risk model for natural gas pipeline components should be developed with engine availability and to integrate with the economic model
4. The economic investigation of the GT engines was based on the effect of the time-based ambient temperature and engine degradation. A combined effect of pipeline elevation, time-based ambient temperature and engine degradation should be investigated to comprehend the actual operational cost of GT engines on the pipeline investment.
5. This research did not look into the gas compressor degradation. A well-detailed analysis on the impacts of gas compressor degradation on the techno-economic of natural gas pipeline transportation network is hereby recommended for an extensive future study.
6. Lifting model that will take into account the Miner's law/ Larson miller parameter should be looked into to assess cumulative damage when operating at varying ambient condition during the time the day

REFERENCES

- [1] United Nations (2019) *World Population Prospect-Population Division*. Available at: https://population.un.org/wpp/Publications/Files/WPP2019_Highlights.pdf (Accessed: 31 July 2019).
- [2] *Business Comparison.com* (2019). Available at: https://www.google.com/search?q=interest+rate+for+business+loan+in+uk&rlz=1C1VFKB_enGB722GB722&oq=interest+rate+for+loan+in+the+uk&aqs=chrome.2.69i57j0l4.23461j0j7&sourceid=chrome&ie=UTF-8 (Accessed: 9 December 2019).
- [3] Sieminski, A. (2016) *International Energy: Publication of the centre for strategic and international studies*, 'Publication of the centre for strategic and international studies'. Available at: <https://www.csis.org/people/adam-sieminski> (Accessed: 11 February 2017).
- [4] Cabalu, H. (2010) 'Indicators of security of natural gas supply in Asia', *Energy Policy*, 38(1), pp. 218–225. doi: 10.1016/j.enpol.2009.09.008.
- [5] Riva, A., D'Angelosante, S. and Trebeschi, C. (2006) 'Natural gas and the environmental results of life cycle assessment', *Energy*, 31(1 SPEC. ISS.), pp. 138–148. doi: 10.1016/j.energy.2004.04.057.
- [6] (NNPC), N. N. P. C. (no date f) *Trans-Saharan Gas Pipeline Project*. Available at: www.nnpcgroup.com/PublicRelations/NNPCinthenews/tabid/92/article (Accessed: 21 March 2017).
- [7] Chaudry, M., Jenkins, N. and Strbac, G. (2008) 'Multi-time period combined gas and electricity network optimisation', *Electric Power System Research*, 78 (7), pp. 1265–1279. doi: 10.1016/j.epsr.2007.11.002.
- [8] Botros K.K (2013) 'Single vs. Dual Recycle System Requirements in the Design of High Pressure Ratio , Low Inertia Centrifugal Compressor Stations. GT2011-45002', *ASME Proceedings*, pp. 905-919; doi: 10.1115/GT2011-45002.
- [9] Suleiman S.M., L. Y. G.(2016) 'Steady State Performance Simulation of Natural Gas Pipeline Driven by Compressor Stations. GT2016-56301', *Proceedings of ASME Turbo Expo. Seoul, South Korea*, V003T06A00, pp. 1–11. doi: 10.1115/GT2016-56301.
- [10] Nasir, A. (2013) *Techno-economic study of gas turbine in pipeline applications*. PhD thesis. Cranfield University.
- [11] Smalley, A. J., Mauney, D. A., Ash, D. I., Clowney, S. L. and Pappas, G. P. (2000) 'Evaluation and application of data sources for assessing operating costs for mechanical drive gas turbines in pipeline service', *Journal of Engineering for Gas Turbines and Power*, 122(3), pp. 462–465. doi: 10.1115/1.1287034.
- [12] Kurz, R. and S. O. (2003) 'Modelinng Turbomachinery in Pipeline Simualtions', in *PSIG Annual Meeting*. Bern, Switzerland: Pipeline Simulation

Interest Group. Available at: <https://www.onepetro.org/conference-paper/PSIG-03A1>.

[13] Polyzakis, A. L., Koroneos, C. and Xydis, G. (2008) 'Optimum gas turbine cycle for combined cycle power plant', *Energy Conversion and Management*, 49(4), pp. 551–563. doi: 10.1016/j.enconman.2007.08.002.

[14] Hosseini, R., Beshkani, A. and Soltani, M. (2007) 'Performance improvement of gas turbines of Fars (Iran) combined cycle power plant by intake air cooling using a media evaporative cooler', *Energy Conversion and Management*, 48(4), pp. 1055–1064. doi: 10.1016/j.enconman.2006.10.015.

[15] Al-Ibrahim, A. M. and Varnham, A. (2010) 'A review of inlet air-cooling technologies for enhancing the performance of combustion turbines in Saudi Arabia', *Applied Thermal Engineering*, 30(14–15), pp. 1879–1888. doi: 10.1016/j.applthermaleng.2010.04.025.

[16] S. Lubomirsky , Matt, K. R. K. P. . M. (2010) 'Configuration impacts availability, fuel consumption and pipeline capacity', *Pipeline Gas Journal*, Vol. 237,.

[17] Nasir, A. (2013) *Techno-Economic Study of Gas Turbine in Pipeline Applications*. Cranfield University.

[18] Tukur, N. (2018) *Techno-Economics of Natural Gas Pipeline Compression System (2018)*. PhD Thesis. Cranfield University, Cranfield. United Kingdom.

[19] John M. Campbell (2011) *How to Estimate Compressor Efficiency*. Available at: <http://www.jmcampbell.com/tip-of-the-month/2015/07/how-to-estimate-compressor-efficiency/> (Accessed: 6 December 2019).

[20] Ohanian, S. and Kurz, R. (2002) 'Series of Parallel Arrangement in a Two-Unit Compressor Station', *Journal of Engineering for Gas Turbines and Power*, 124(4), p. 936. doi: 10.1115/1.1478074.

[21] Brenne, L., Koch, J. M. and Miller, H. (2001) 'PERFORMANCE EVALUATION OF A CENTRIFUGAL COMPRESSOR OPERATING UNDER WET GAS CONDITIONS by'.

[22] Shashi Menon, E. (2005) *Gas Pipeline Hydraulics*. CRC Press Taylor & Francis Group 6000 Broken Sound Parkway NW. doi: 10.1201/9781420038224.

[23] Musa, G. H. (2019) *Techno-Economic Analysis of Gas turbine Compressor Fouling and washing*. Cranfield University, Bedfordshire, United Kingdom.

[24] Leusden, C.P., Sorgenfry, C., and Dummel, L. (no date x) 'Performance benefits using semen advanced compressor cleaning system', *ASME J.Eng. Gas Turbine Power*, 126, pp. 763–769.

[25] Abdelrazik, A. and Cheney, P. (1991) 'Compressor cleaning effectiveness for marine gas turbines', *Proceedings of the ASME Turbo Expo*, 2. doi: 10.1115/91GT011.

- [26] NNPC, Sonatrach, Penspen,. (NNPC Towers, Central Business District, H. and Macaulay Way, P.M.B. 190, Garki, A. . (2006) *Trans-Saharan Gas 258 Pipeline Project: Final Feasibility Report (unpublished Feasibility Report)*, London.
- [27] EIA (2015) *World Gas Reserve by Countries*. Available at: <https://www.google.co.uk/search?q=2018+natural+gas+reserves&tbm=isch&source=iu&ictx=1&fir> (Accessed: 30 July 2016).
- [28] *The world's biggest natural gas reserves 2019* (no date ab). Available at: <https://www.hydrocarbons-technology.com/features/feature-the-worlds-biggest-natural-gas-reserves/> (Accessed: 14 February 2019).
- [29] Rajnauth, J., Ayeni, K. and Barrufet, M. (2008) 'Gas Transportation: present and future', *Society of Petroleum Engineers - SPE Gas Technology Symposium*, 2(June), pp. 528–544. doi: 10.2118/114935-MS.
- [30] Thomas, S. and Dawe, R. A. (2003) 'Review of ways to transport natural gas energy from countries which do not need the gas for domestic use', *Energy*, 28(14), pp. 1461–1477. doi: 10.1016/S0360-5442(03)00124-5.
- [31] Liu, H. (2003) *Pipeline Engineering*. Edited by P. J. M. et Al. United State of America: Lewis Publishers. Available at: www.crcpress.com.
- [32] Watanabe, T., Inoue, H., Horitsugi, M. and Oya, S. (2006) 'Gas to Wire (GTW) system for developing "small gas field" and exploiting "associated gas"', *International Oil and Gas Conference and Exhibition in China 2006 - Sustainable Growth for oil and Gas*, 1(2), pp. 310–315.
- [33] Wang, X. and Economides, M. (2009) 'Advanced Natural Gas Engineering', *Gulf Publishing Company Houston, Texas*, pp. 351–368. doi: 10.1016/B978-1-933762-38-8.50018-6.
- [34] Lun, L. and Xie, Y. (2008) 'Gas Turbine Cycle Recovering Pressure Energy of Natural Gas Transportation Pipelines by Vortex Tube', (July), pp. 28–30.
- [35] Montoya, S. J., Jovel, W. a., Hernandez, J. a. and Gonzalez, C. (2000) 'Genetic Algorithms Applied to the Optimum Design of Gas Transmission Networks', *Proceedings of SPE International Petroleum Conference and Exhibition in Mexico*. doi: 10.2118/59030-MS.
- [36] Subero, G., Sun, K., Deshpande, A., McLaughlin, J. and Economides, M. J. (2013) 'A Comparative Study of Sea-Going Natural Gas Transport', *SPE Annual Technical Conference and Exhibition*. doi: 10.2118/90243-MS.
- [37] Ouyang, L., Aziz, K. (1996) 'Steady-state gas flow in pipes', *Journal of Petroleum Science and Engineering*, Vol.14, no(3–4), p. pp.137-158.
- [38] Shashi Menon, E. (2005) *Gas Pipeline Hydraulics, System*. United State: CRC Press Taylor and Francis Group. doi: 10.1201/9781420038224.
- [39] Aziz, K. and Ouyang, L. B. (1995) 'Simplified equation predicts gas flow rate, pressure drop", *Oil and Gas journal*, 93, no 19, pp. 70–71.

- [40] Beggs, H. . (1984) *Gas production operations*, Oil & Gas Consultants International Inc. Tulsa. Available at: <https://fastcdn.pro/filegallery/golcode.com/Downloads/Gas Production Operations.pdf>.
- [41] Smith, R. V. (1990) *Practical natural gas engineering*, Pennwell Books, Tulsa, OK. 2nd Edition: Tulsa, Okla.: Pennwell Books,. Available at: <https://www.worldcat.org/title/practical-natural-gas-engineering/oclc/21296387>.
- [42] Menon, S. E. (2005) *Gas Pipeline Hydraulics, System*. United state: CRC: Press Taylor and Francis Group; doi: 10.1201/9781420038224.
- [43] Woldeyohannes, A. D., Amin, M. and Majid, A. (2014) 'Simulation model for natural gas transmission pipeline network system', *STIMULATION MODELLING PRACTICE AND THEORY*. Elsevier B.V., (January 2011). doi: 10.1016/j.simpat.2010.06.006.
- [44] U.S. Energy Information Administration (2007) 'Natural gas compressor stations on the interstate pipeline network: Developments since 1996', *Energy Information Administration, Office of Oil and Gas*, (November), pp. 1–12. Available at: http://www.eia.gov/pub/oil_gas/natural_gas/analysis_publications/ngcompressor/ngcompressor.pdf.
- [45] Tukur N (2012) *A Comparative Study of Liquefied Natural Gas and Pipelines as Means of Transporting Natural Gas in Nigeria*. Coventry University , United Kingdom.
- [46] Kurz, R. and Incorporated, S. T. (2012) 'Concepts in Gas Compressor Station Configuration', *International Petroleum Technology Conference*, Vol. IPTC1, pp. 1–10.
- [47] Beal, L. (2004) 'The Routine Overhaul of Natural Gas-Fired Combustion Turbines in Interstate Natural Gas Transmission Service', (December). Available at: file:///Z:/Overhaul paper.pdf.
- [48] Mokhatab, S., Santos, S.P. and Cleveland, T. (2007) 'Compressor Station Design Criteria', *Pipeline and Gas Journal*. Available at: <http://www.highbeam.com/doc/IGI-165730450.html>.
- [49] Santos, S. . (2000) 'Series Or Parallel-Tailor Made Design Or a General Rule For a Compressor Station Arrangement', *Pipeline Simulation Interest Group-PSIG Annual Meeting, October 28-20, 2000, Hyatt Regency Savannah Savannah, Georgia*.
- [50] Kurz R. Ohanian S, L. (2009) 'On Compressor Station Layout', *ASME Conference Proceeding: Turbo Expo*, Vol. 4: Tu, p. pp 1-10. doi: <https://doi.org/10.1115/GT2003-38019>.
- [51] Sebouh O., Kurz, R. (no date ay) 'Series of Parallel Arrangement in a Two-Unit Compressor Station', *ASME Turbo Expo, Louisiana, USA*, (Paper No: 2001-GT-0230, V002T03A008;), p. pp 8. doi: <https://doi.org/10.1115/2001-GT-0230>.

- [52] Kurz, R., Lubomirsky, M. and Brun, K. (2012) 'Gas compressor station economic optimization', *International Journal of Rotating Machinery*, 2012. doi: 10.1155/2012/715017.
- [53] Lubomirsky, M., Kurz, R., Klimov, P. and Mokhatab, S. (2010) 'Configuration impacts availability, fuel consumption and pipeline capacity', *Pipeline and Gas Journal*, 237(1), pp. 48–54.
- [54] Santos, S. . (1997) 'TRANSIENT ANALYSIS, A Must in Gas Pipeline Design"', *29th Annual Meeting Pipeline Simulation Interest Group (PSIG), October 15-17, 1997, Westward Look Resort Tuscon, Arizona*, Vol. 15.
- [55] dos Santos, S. P. (2004) 'Series Or Parallel Arrangement For a Compressor Station? - A Recurring Question That Needs a Convincing Answer', 30(1059). Available at: <http://www.onepetro.org/mslib/app/Preview.do?paperNumber=PSIG-0405&societyCode=PSIG>.
- [56] Bahadori, A. (2014) *Natural gas processing technology and engineering design*. Waltham, USA: Elsevier.
- [57] Bloch, H.P., Hoefner, J. J. (1996) 'Resiprocating Compressors, Opeartion and Maintenancen Gulf Professional Publishing', in. Available at: https://www.google.com/search?q=types+of+compressors+used+in+oil+and+gas+industry&rlz=1C1VFKB_enGB722GB722&oq=types+of+gas+compressors&aqs=chrome.4.69i57j0l5.37286j0j7&sourceid=chrome&ie=UTF-8.
- [58] Boyce, M. P. (2003) *Centrifugal compressors: A basic guide*. 1st editio. Oklahoma: Penn Well.
- [59] Bloch, H. P. (2006) *Compressors and modern process applications*,. First edit. Wiley Interscience, Hoboken, N.j.
- [60] Dessibourg D. (2011) 'Customer Training Seminar, Man Diesel & Turbo, Kuala Lumpur, Malaysia.'
- [61] Gallar, L., Volpe, V., Salussolia, M., Pachidis, V. and Jackson, A. (2012) 'Thermodynamic Gas Model Effect on Gas Turbine Performance Simulations', *Journal of Propulsion and Power*, 28(4), pp. 719–727. doi: 10.2514/1.B34359.
- [62] Mokhatab, S., Poe, W.A and speight, J. G. (2006) *Hanbook on Natural Gas transmission and Processing*. Gulf Professional Publishing, Texas, USA. Available at: <https://www.elsevier.com/books/handbook-of-natural-gas-transmission-and-processing/mokhatab/978-0-7506-7776-9>.
- [63] Kurz, R., Ohanian, S. and Brun, K. (2010) 'Compressors in High Pressure Pipeline Applications', *Proceedings of the ASME Turbo Expo*, 5(June 2010), pp. 631–640.
- [64] Pequot (2008) *Gas turbine World Handbook*, South, CT. Pequot Publishing Inc. Available at: <https://gasturbineworld.com/product-category/annual-handbook/>.

- [65] Brinckerhoff, P. and Zealand, N. (2008) 'COST ESTIMATES FOR THERMAL PEAKING PLANT " FINAL REPORT "', (June).
- [66] Glen Arnold (2008) *Corporate Finance Management*. 4th Editio. Financial Times Pitman Publishing Limited.
- [67] Rodilla, P., Cerisola, S. and Batlle, C. (2012) 'MODELING THE MAJOR OVERHAUL COST OF GAS-FIRED PLANTS IN THE UNIT COMMITMENT PROBLEM', (November), pp. 1–22.
- [68] Mafel, O. (2018) *Techno-Economic and Environmental Risk Assessment of Gasturbines for use with Flared Associated Gases*. Cranfield Univeristy.
- [69] Sekirnjack, E. (2007) 'Optimal Control-A Realistic Goal fo Gas Transmission', in. PSIG Annual Meeting hel in Calgary, Alberta.
- [70] Wu, S., Ríos-Mercado, R. Z., Boyd, E. A. and Scott, L. R. (2000) 'Model relaxations for the fuel cost minimization of steady-state gas pipeline networks', *Mathematical and Computer Modelling*, 31(2–3), pp. 197–220. doi: 10.1016/S0895-7177(99)00232-0.
- [71] Krishnaswami P.; Chapman K.S.; Abbaspour M. (2004) 'PSIG 0410 Compressor Station Optimization for Linepack Maintenance', *PSIG Annual Meeting held in Palm Springs, California*, pp. 1–12.
- [72] Santos S.P., Kurz R., L. M. (2005) 'Fuel Consumption Impact on Gas Pipeline Projects', in *Rio Pipeline Conference & Exposition*. Rio de Janeiro.
- [73] Kurz, R. and Ohanian, S. (2003) 'Modeling Turbomachinery in Pipeline Simulations", *Proceedings PSIG*, vol. 3.'
- [74] de Marco, F. C. G. and Elias, G. P. (2011) 'Fuel Consumption Model On Natural Gas Compression Stations Driven By Two-Shaft Gas Turbine', *PSIG Annual Meeting*. Available at: https://www.onepetro.org/conference-paper/PSIG-1107?sort=&start=0&q=PSIG+1107&from_year=&peer_reviewed=&published_between=&fromSearchResults=true&to_year=&rows=10#.
- [75] Kurz, R. and Brun, K. (2007) 'Gas turbine tutorial–Maintenance and operating practices effects on degradation and life', *Proceedings of 36th Turbomachinery*, pp. 173–185.
- [76] Hardy-Bishop (2012) *The Specification of Mechanical & Electrical Rotating Machines: Synchronous and Asynchronous Machines (unpublished Short Course)*. Cranfield, UK.
- [77] Walsh, P.P., and Fletcher, P. (2004) *Gas Turbine Performance*. 2nd editio. Oxford Blackwell Science, Oxford.
- [78] Kurz, R., Mokhatab, S. (2007) 'Consideration on compressor station layout', *Pipeline and Gas Journal*, 234(9), pp. 24–26.
- [79] Kurz, R. (2004) 'The Physics of Centrifugal Compressor Performance', in *Pipeline Simulation Interest Group*. California.: Palm Springs, California.

- [80] Horowitz, F. B. (2006) 'Compressor Control and Optimization', (1970), pp. 1763–1793.
- [81] H.I.H Saravanamuttoo,. G.F.C Rogers,. H. Cohen,. P, V, S. (2009) *Gas Turbine Theory*. Sixth Edit. Prentice Hall imprint. Available at: www.pearsoned.co.uk.
- [82] Soares, C. (2008) *Gas turbines*. Butterworth-Heinemann, Amsterdam; Boston.
- [83] Connors, H. (1970) 'Particle Separator for Engines Air Inlets'.
- [84] Clara, S. (2012) *Effect of Engine Degradation on Fuel Burn Optimum Civil Aircraft Trajectories*. Cranfield University.
- [85] Abdul Ghafir, M. (2011) *Performance Based Creep Life Estimation for Gas Turbines Application*. Cranfield University.
- [86] Naeem, M., Singh, R. and Probert, D. (2001) 'Impacts of aero-engine deteriorations on military aircraft mission's effectiveness', *Aeronautical Journal*, 105(1054), pp. 685–695.
- [87] Kurz, R. (2005) 'Gas Turbine Performnace', *Proceedings of the Thirty-Fourth Turbomachinery Symposium*, pp. 131–146.
- [88] Kurz, R. and Brun, K. (2001) 'Degradation in Gas Turbine Systems', *Journal of Engineering for Gas Turbines and Power*, 123(1), p. 70. doi: 10.1115/1.1340629.
- [89] Igie, U., Diez-Gonzalez, P., Giraud, A. and Minervino, O. (2016) 'Evaluating Gas Turbine Performance Using Machine-Generated Data: Quantifying Degradation and Impacts of Compressor Washing', *Journal of Engineering for Gas Turbines and Power*, 138(12), p. 122601. doi: 10.1115/1.4033748.
- [90] Meher-homji, C. B. and Bromley, A. (2001) 'Gas Turbine Performance Deterioration', *Symposium A Quarterly Journal In Modern Foreign Literatures*.
- [91] Diakunchak, I. S. (1992) 'Performance Deterioration in Industrial Gas Turbines', *Journal of Engineering for Gas Turbines and Power*, 114(2), p. 161. doi: 10.1115/1.2906565.
- [92] Bammert, K. and Stobbe, H. (1970) 'Result of experiments for determining the influence of blade profile changes and manufacturing tolerances on efficiency, the enthalpy drop, and the mass flow of multi-stage axial turbines', in *ASME Winter Annual Meeting*. New York, N.Y: ASME, p. 12. doi: 10.1115/70-WA/GT-4.
- [93] Bammert, K., and Woelk, G. U. (1980) 'The Influence of the Blading Surface Roughness on the Aerodynamic Behaviour and Characterisrttics of an Axial Compressor', *ASME Journal of Engineering for Powewer*, Vol. 102,.
- [94] DeGreef, J. L., Maes, P., and Johnson, K. W. (1978) 'Operating experience on Residual Fuel Oil With a W251 Combustion Turbine', *ASME paper 78-GT-*

- [95] Haub, G.L., and Hauhe, W. E. (1990) 'Field Evalution of on-line Compressor Cleaning in Heavy Duty Industrial Gas Turbine', *ASME Paper 90-GT-107*.
- [96] Upton, A. W. J. (1974) 'Axial flow Compressor and Turbine Blade for some causes, Effects, and cleaning Methods', *Presented at the first Symposium on Gas Turbine Operation and maintenance, National Research council, Canada*.
- [97] Scheper, G.W., Mayoral, A.J., and Hipp, E. J. (1978) 'maintaining gas turbine compressors for high efficiency', *Power Eng.*, 82(8), p. pp.54-57.
- [98] Meher-homji, C. B. and Bromley, A. (2004) 'Gas Turbine Axial Compressor Fouling and Washing', *Proceedings of the 33rd Turbomachinery Symposium*, p. pp 163-1992.
- [99] William Cousins, by T., O, W. F., Eugene Brown Milt W Davis, C. F., Wing Ng Joseph A Schetz, J. F. and William Cousins Committee Chairman, by T. (1997) 'The Dynamics of Stall and Surge Behavior in Axial-Centrifugal Compressors'. doi: etd-11298-35210.
- [100] Meher-Homji, C. B., Chaker, M. and Bromley, A. F. (2009) 'The Fouling of Axial Flow Compressors: Causes, Effects, Susceptibility, and Sensitivity', *Volume 4: Cycle Innovations; Industrial and Cogeneration; Manufacturing Materials and Metallurgy; Marine*, pp. 571–590. doi: 10.1115/GT2009-59239.
- [101] Schilke P.W, Foster AD, Pepe J.H, B. A. . (1991) 'Advanced materials propel progress in land-based gas turbines.', *Adv Mater Process*, 141 (2):22.
- [102] Eliaz, N., Shemesh, G. and Latanision, R. M. (2002) 'Hot corrosion in gas turbine components', *Engineering Failure Analysis*, 9(1), pp. 31–43. doi: 10.1016/S1350-6307(00)00035-2.
- [103] N, Y. V and Al-hadhrani, L. M. (2009) 'Degradation of turbine blades and vanes by overheating in a power station', 16, pp. 273–280. doi: 10.1016/j.engfailanal.2008.05.001.
- [104] Meherwan, P. B. (2007) *Gas turbine Engineering Handbook*. 3rd Editio. Edited by T. B. Consultancy. Gulf Professional Publishing, Texas, USA.
- [105] Ben Hariz, H. (2010) *The Optimisation of the Usage of Gas Turbine Generation Sets for Oil and Gas Production Using Genetic Algorithms*. Cranfield University, Cranfield, United Kingdom.
- [106] Aker, G. F. and Saravanamuttoo, H. I. H. (1989) 'Predicting Gas Turbine Performance Degradation Due to Compressor Fouling Using Computer Simulation Techniques', *Journal of Engineering for Gas Turbines and Power*, 111(2), p. 343. doi: 10.1115/1.3240259.
- [107] Urban, L. A. (1975) 'Parameter selection for multiple fault diagnostics of gas turbine engines', *Journal of Engineering for Power*, 74-GT(62), pp. 225–230.
- [108] Saravanamuttoo, H. I. H. and Maclsaac, B. D. (1983) 'Thermodynamic

Models for Pipeline Gas Turbine Diagnostics', *Journal of Engineering for Power*, 105(4), pp. 875–884. doi: 10.1115/1.3227496.

[109] Lukachko, S. P. and Waitz, I. a (1996) 'Effects of Engine aging on aircraft NOx Emissions', *Focus*, (x), pp. 1–15. doi: 10.1115/97-GT-386.

[110] Garcia-Hernandez, A. and Brun, K. (2011) 'Energy Usage in Natural Gas Pipeline Applications', *Volume 4: Cycle Innovations; Fans and Blowers; Industrial and Cogeneration; Manufacturing Materials and Metallurgy; Marine; Oil and Gas Applications*, 134(February 2012), pp. 1049–1057. doi: 10.1115/GT2011-46582.

[111] Mamat, Z. A. (no date dg) "'Gas Turbine Compressor Water Washing Practice & Effective for TBN's Combined Cycle Plants'" Generation Asset Management, Tenaga Nasional Berhad.' Tenaga Nasional Berhad.

[112] Thames, J. M., Stegmaier, J. W. and Ford, J. J. (1989) 'On-line compressor washing practices and benefits', *American Society of Mechanical Engineers (Paper)*, pp. 2–7. Available at: <http://www.scopus.com/inward/record.url?eid=2-s2.0-0024858958&partnerID=tZOtx3y1>.

[113]
http://www.tubetech.com/wpcontent/gallery/turbinebladecleaning/2_tube_tech_t_echnician_sing_fastice_to_clean_turbine_blades.jpg (no date di).

[114] Faddegon, C. J. (1999) 'Effective compressor cleaning as a result of scientific testing and field experience', (*Private communication*).

[115] Kulle, V. (no date dk) 'Gas Turbine Blade Cleaning at Alberta Gas Trunk Line (AGTL)', *Gas Turbine operation and Maintainance Symposiumb 1974, National Research Council (eds.),, Toronto, Canada*, p. pp 265-271.

[116] Brittain., D. (1983) 'Cleaning Gas Turbine Compressor', *Aircr., Eng.*, 55(1), pp. pp15-17.

[117] Godon, R. (2010) 'The Evolution of Gas turbine Compressor Cleaning IDGTE Meeting'. Bedford U.K: Publication 576.

[118] Boyce, M. . & G. (2007) 'A study of on-line and off-line Turbine Washing to Optmise the operation of a Gas turbine.', *Journal of Engineering for Gas Turbines and Power*, 129 (1), p. pp 114-122.

[119] Boyce, M. P., Bowman, J. C., Focke, A. B. and Meher-Homji, C. B. (1985) 'Energy Efficient Operation of Gas Turbine Compressor Sets', *American Society of Mechanical Engineers*, 85-GT-70, pp. 1–7.

[120] Fielder, J. (2003) 'evaluation of zero compressor wash routine in RN service', *ASME Paper No. GT2003-38887*.

[121] Testman, F.L., Jr. (1999) 'Engine Wash Recovery System', *U.S Patent No. 5,899,217*.

[122] Anon. (1999) 'LM2500 Military Marine Gas Turbine Instalation Design Manual', *GE Marine Egines, Cincinnati*.

- [123] Asplund, P. (2017) 'Gas Turbine Cleaning Upgrade (Compressor Wash)', *ASME Paper 97-AA-135*.
- [124] Mezheritsky, A. D. and Sudarev, A. V. (1990) 'The Mechanism of Fouling and the Cleaning Technique in Application to Flow Parts of the Power Generation Plant Compressors', *The American Society of Mechanical Engineers*, p. 13. doi: 10.1115/90-GT-103.
- [125] Margolis, H. (1991) 'U . S . Navy On-Line Compressor Washing of Marine Gas Turbine Engines', pp. 6–11. doi: 10.1115/91GT309.
- [126] VGB (1991) 'Reinigungsverfahren für Verdichter -und Turbinenbeschaufelungen an Gasturbinen im offenen Prozess,' VGB Merkblatt 106, 2nd Edition, VGB-Kraftwerkstechnik GmbH, Essen, Germany, pp.10-11'.
- [127] Lambert, P., Gordon, R., and Burnet, M. (2003) 'Developments in on-line gasturbine compressor cleaning', *Institution of Diesel and Gas Turbinme Engineers, 2nd Gas Turbine Conference 2003, Miton kynnes, UK*, p. PP. 136-142.
- [128] Ltd, R.-M. P. R. (2016) *R-MC Power Recovery Ltd*.
- [129] Stalder, J.-P. (2001) 'Gas Turbine Compressor Washing State of the Art: Field Experiences', *Journal of Engineering for Gas Turbines and Power*, 123(2), p. 363. doi: 10.1115/1.1361108.
- [130] Ramsden, K. (2017) 'Industrial Gas Turbine Performance : Compressor Fouling and On-Line Washing', 136(October 2014), pp. 1–13. doi: 10.1115/1.4027747.
- [131] Aretakis, N., Roumeliotis, I., Doumouras, G. and Mathioudakis, K. (2012) 'Compressor washing economic analysis and optimization for power generation', *Applied Energy*, 95, pp. 77–86. doi: 10.1016/j.apenergy.2012.02.016.
- [132] Kacprzyński G.J., Gumina M.Roemer, M.J., Caguiat, D.E., Galie, T.R, McGroarty, J. . (2001) 'a prognostic modeling approach for predicting recurring maintenance for shipboard propulsion systems', *ASME Paper 2001-GT-0218*.
- [133] <http://sitelife.aviationweek.com/ver1.0/Content/images/store/3/0/b32d002-de43-4ae0-805c-cb9889caffb9.Large.jpg> (no date ec).
- [134] NSPS proposal (1977) *Turbine Replacement due to Routine Maintenance: NSPS Proposal Support Document, EPA-450/2-77-017a*.
- [135] Wulf, R. H. (1980) *Engine Diagnostics Program: CF6-50 Engine Performance Deterioration* NASA CR-159867, conducted by the General Electric Company, Aircraft Engine Business Group.
- [136] EPA (2004) *INGAA Comments on EPA Reconsideration of NSR Equipment Replacement Provisions. EPA Docket Number OAR-2002-0068-2565*,.
- [137] Nasir, A., Ogaji, S., Ahmed, W. and Atty, A. (2012) 'ICONE20POWER2012-54273', (July). doi: 10.1115/ICONE20-POWER2012-54273.

- [138] Lefebvre, H. and Ballal, R. (2010) *Gas turbine combustion, alternative fuels and emissions*. 3rd edn. Boca Raton, USA: CRC Press, Taylor and Francis Group.
- [139] Abdul-Wahab, S.A., Zurigat, Y.H. and Bortmany, J. . (2005) 'Gas turbine emissions and environmental impact of efficiency boosting techniques', *International Journal of Environment*, Vol.23 no., p. pp.273-288.
- [140] United States Energy Information Administration (EIA). (2017) *Emissions of greenhouse gases in the United States 2017*. Available at: <https://www.epa.gov/ghgemissions/sources-greenhouse-gas-emissions>.
- [141] Department for Business, E. and I. S. (2018) *Greenhouse Gas Emissions, Provisional Figures and 2018 UK Greenhouse Gas Emissions*,.
- [142] Rokke, P. E., Hustad, J. E., Rokke, N. A. and Svendsgaard, O. B. (2003) 'Technology update on gas turbine dual fuel, dry low emission combustion systems', *ASME Turbo Expo, June 16 - 19, Atlanta, Georgia, USA.*, Vol. 2,.
- [143] Brainard, J. (2012) *Combustion Reaction-Physical Science*. Available at: <https://www.ck12.org/c/physical-science/combustion-reaction/lesson/Combustion-Reactions-MS-PS/> (Accessed: 24 May 2019).
- [144] Ogaji, S., Pilidis, P. and Hales, R. (2007) 'TERA - a tool for aero-engine modelling and management', in *Second World Congress on Engineering Asset Management and the Fourth International Conference on Condition Monitoring*. Harrogate, UK, pp. 11–12.
- [145] Baioni C.A (2009) *Risk Methodology for Gas Turbine Selection in LNG application*.
- [146] Kyprianidis, K. G., Colmenares Quintero, R. F., Pascovici, D. S., Ogaji, S. O. T., Pilidis, P. and Kalfas, A. I. (2008) 'EVA - a tool for environmental assessment of novel propulsion cycles', in ASME (ed.) *ASME Turbo Expo 2008: Power for Land, Sea and Air*. Berlin, Germany: ASME, pp. 1–10. doi: 10.1115/GT2008-50602.
- [147] Colmenares Quintero, R., Kyprianidis, K., Gomez, J., Ogaji, S. and Pilidis, P. (2008) "Future aero-engines" optimisation for minimal fuel burn"", in *ASME TURBO EXPO 2008*. Berlin, Germany.
- [148] Gayraud, S. (1996) *Technical and environmental assessment for industrial gas turbine selection*. MS.c thesis. Cranfield University.
- [149] Gayraud, S. (1998) *Design of a decision support system for combined cycle schemes*. M. Phil. thesis. Cranfield University.
- [150] Gayraud, S. and Singh, R. (1997) 'A Techno-economic computational tool for power generation project assessments and life cycle risk management', in *ASME 1997 Turbo Asia Conference*, pp. 1–8. doi: 10.1115/97-AA-130.
- [151] Maccapani, M., Khan, R. S. R., Burgmann, P. J., Di Lorenzo, G., Ogaji, S. O. T., Pilidis, P. and Bennett, I. (2014) 'A TERA Based Comparison of Heavy

Duty Engines and Their Artificial Design Variants for Liquefied Natural Gas Service', *Journal of Engineering for Gas Turbines and Power-Transactions of the Asme*, 136(2). doi: 10.1115/1.4025474.

[152] Nkoi, B. (2014) *Techno-economic studies of environmentally friendly brayton cycles in the petrochemical industry*. PhD thesis. Cranfield University.

[153] El-Suleiman, A. (2014) *Gas Turbine Application to CO2 Pipeline: Techno-Economic and Environmental Risk Analysis (PhD Thesis)*. Cranfield University.

[154] Klarenbach, S. W. (2014) 'Economic evaluation of dialysis therapies Nat. Rev. Nephrol.' doi: 10.1038/nrneph.2014.145.

[155] Brealey; Myers; Allen (2006) *Corporate Finance*. 8th edn. McGraw-Hill.

[156] Mohamed, W. (2013) *Techno-economic, environmental and risk analysis (TERA) for power generation - market growth*. PhD thesis. Cranfield University.

[157] Ogiriki, A. E. (2015) *Effects of environmental factors on gas turbine engine creep life and performance*. PhD thesis. Cranfield University.

[158] Khatib, H. (2003) *Economic evaluation of projects in the electricity supply industry*. 3rd edn. Edited by Institution of Engineering and Technology.

[159] Carsberg, B. . (1974) *Analysis for Investemnt Decision , Accountancy Age Books*.

[160] *Accounting Tools Incorporation* (2017). Available at: <https://www.accountingtools.com/articles/what-is-salvage-value.html> (Accessed: 22 January 2018).

[161] NNPC (2006) *Sonatrach, Penspen,. (NNPC Towers, Central Business District H.Macaulay Way, P.M.B. 190, Gark A. Trans-Saharan Gas 258 Pipeline Project: Final Feasibilty Report (Unpublished Feasibilty Report), London*.

[162] Pereira, S., Lubomirsky, M. and Turbines, S. (2006) 'Gas Composition Effect on Centrifugal Compressor Performance', (1), pp. 1–12.

[163] Ludtke, K. (2004) *Process Centrifugal Compressor: basics, function, operation, design, application*, Springer, Berlin.

[164] BLOCH, H. . (2006) *Compressor and Modern Process application*,. First edit. Wiley Interscience, Hoboken, N.j.

[165] Wang, Y., Lin, F., Nie, C. and Engeda, A. (2013) 'Design and Performance Evaluation of a Very Low Flow Coefficient Centrifugal Design and Performance Evaluation of a Very Low Flow Coefficient Centrifugal Compressor', (September). doi: 10.1155/2013/293486.

[166] Lapina, R. P. (1982) *Estimating Centrifugal Comperssor Performance, Process Compressor Technology, Volume 1*. First Edit. Gulf Publishing Company, Houston, Texas.

[167] Al-Busaidi, W. and Pilidis, P. (2016) 'Investigation of Efficiency Deterioration

Causes in Process Centrifugal Compressor Operation', *Journal of Failure Analysis and Prevention*, 16(1), pp. 19–36. doi: 10.1007/s11668-016-0069-2.

[168] Berdanier, R. A., Smith, N. R., Fabian, J. C. and Key, N. L. (2014) 'Humidity Effects on Experimental Compressor Performance—Corrected Conditions for Real Gases', *Journal of Turbomachinery*, 137(3), p. 031011. doi: 10.1115/1.4028356.

[169] Nikolaidis T (2016) 'TURBOMATCH SCHEME, Modelling and Simulation code for Gas turbine Engine, Centre for Power and Propulsion, Cranfield University, United Kingdom'.

[170] A. Gad-Briggs, A. Haslam, P. L. (2013) 'Effect of change in role of an aircraft on engine life', *The Aeronautical Journal*, 117(1196), pp. 1053–1070.

[171] Kurz, R., Brun, K. and Legrand, D. D. (1998) 'Field Performance Testing of Gas Turbine Driven Compressor Sets', PSIG Annual Meeting'.

[172] klaus Brun, J. M. (2018) 'API Specification Review for Gas Turbine Driven TURBOCOMPRESSORS', *Southwest Research Institute, San Antonio, Texa*. Available at: <https://www.scribd.com/document/376674109/API-specification-review-for-gas-turbine-driven-turbocompressors-pdf>.

[173] (AP1), A. P. I. (2011) *Gas Turbine for Petroleum, Chemical, and Gas Industry Service*. Available at: https://global.ihs.com/doc_detail.cfm?document_name=API STD 616&item_s_key=00010657#abstract-section (Accessed: 20 February 2017).

[174] Walsh, P.P., and Fletcher, P. (2004) 'Gas Turbine Performance" 2nd edition, Blackwell Science LTD'.

[175] Pilidis, P., Palmer, J. R. (2013) *Gas Turbine Theory and Performance*. Cranfield: Cranfield University.

[176] Kurz, R. and Brun, K. (2009) 'Degradation of gas turbine performance in natural gas service', *Journal of Natural Gas Science and Engineering*, 1(3), pp. 95–102. doi: 10.1016/j.jngse.2009.03.007.

[177] Meherwan; P.Boyce (2006) *Gas turbine Engineering Hand book*. Third Edit. Gulf Professional Publishing, 30 Corporate Drive, Suit 400, Burlington, MA 01803, USA.

[178] Kurz, R. and Brun, K. (2008) 'Degradation Effects on Industrial Gas Turbines'.

[179] Onari, M. M. (2014) '43 rd Turbomachinery & 30 th Pump Users Symposia (Pump & Turbo 2014) September 23-25 , 2014 | Houston , TX | pumpturbo . tamu . edu', (1998).

[180] Røkke, N. A., Hustad, J. E. and Berg, S. (1993) 'Pollutant Emissions From Gas Fired Turbine Engines in Offshore Practice: Measurements and Scaling', *ASME 1993 International Gas Turbine and Aeroengine Congress and Exposition*, p. V03AT15A021-V03AT15A021. doi: 10.1115/93-GT-170.

- [181] Bejan, A.; Tsatsaronis G; Moran, M. J. (1996) 'Thermal design and optimization.', *Wiley Interscience, John Wiley Sons Inc., New York*, Vol.21, No, pp. 433–434.
- [182] API (2016) *API Standard Specification for Line Pipe*. Available at: <https://www.api5lx.com/api5lx-grades/>.
- [183] Meherwan P.Boyce (2007) 'Gas Turbine Engineering Handbook', in. Gulf Professional Publishing, 30 Corporate Drive, Suit 400, Burlington, MA 01803, USA, p. pg 8.
- [184] Tian-gang Zhang, X. H. (2011) 'Nox emission control in gas turbine', *Applied Mechanics and Materials*, Vol. 66-68, pp. pp319-321. doi: doi 10,4028 www.scitific.net/AMM 66.68.319.
- [185] Jung, B. H. J. and Becker, E. R. (1987) 'Emission Control for Gas Turbines', (4), pp. 162–170.
- [186] Toyoaki Komori, Shigehiro Shiozaki, Nobuyuki Yamagami, Yohsuke Kitauchi, W. A. (2007) 'C O 2 E m i s s i o n R e d u c t i o n Method Through Various Gas Turbine Fuel Applications', *Mitsubishi Heavy Industries Ltd. Technical Review*, 44(1), pp. 1–5.
- [187] GE Engines (2016) *Gas turbine for Land and Marine application*. Available at: <https://pdf.nauticexpo.com/pdf/ge-marine/lm-1600/31041-11949.html> (Accessed: 20 November 2016).

APPENDICES

Appendix A

Table A-1 Gas Properties and Composition

<i>S/NO</i>	<i>Gas component</i>	<i>Value</i>	<i>Unit</i>
1	Molecular weight of Gas mixture (MW_m)	17.544	g/g-mol
2	Critical Tempt of gas mixture (T_{c_m})	199.182	K
3	Mole/volume fraction of gas mixture	100	%
4	Critical Pressure of gas mixture (P_{c_m})	46.147	bara
5	Specific Heat Ratio (K)	1.296	-
7	Reduced Pressure (P_r)	1.470	
8	Reduced Temperature (T_r)	1.447	
11	Specific heat (Cp_m)	36.436	
13	Normal Density	0.783	Kg/m ³

Table A-2 Individual Gas mixture

<i>Compound</i>	<i>Molecular weight (MW_i)</i>	<i>Volume fraction (%)</i>	<i>Formulae</i>
<i>Methane</i>	16.043	92.180	CH ₄
<i>Ethane</i>	30.07	4.930	C ₂ H ₆
<i>Propane</i>	44.097	0.560	C ₃ H ₈
<i>ISO-Butane</i>	58.124	0.310	C ₄ H ₁₀
<i>N-Butane</i>	58.124	0.110	C ₄ H ₁₀
<i>Nitrogen</i>	28.013	0.91	N ₂
<i>N-Pentane</i>	72.151	0.04	C ₅ H ₁₂
<i>n-Hexane</i>	86.178	0.05	C ₆ H ₁₄
<i>n-Octane</i>	114.232	0.01	C ₈ H ₁₈
<i>n-Heptane</i>	100.205	0.03	C ₇ H ₁₆

<i>n</i> -Nonane	128.26	0	
<i>n</i> -Decane	142.29	0	
<i>N</i> -Undecane	156.31	0	
Benzane	78.11	0	
Toluene	92.14	0	
<i>O</i> -xylene	106.17	0	
<i>ISO</i> -Pentane	72.15	0.11	C ₅ H ₁₂
Hydrogen Sulphide	34.08	0	
Carbon Dioxide	44.01	0.76	CO ₂
Water	18.02	0	

Appendix B Compressor map from the OEM

The OEM gas compressor data was supplied for the analysis of this research. The OEM gas compressor data are used to scale the proposed TSGP gas compressors. The new scaled TSGP gas compressors have a relatively accurate representation of the expected gas compressor performance curves.

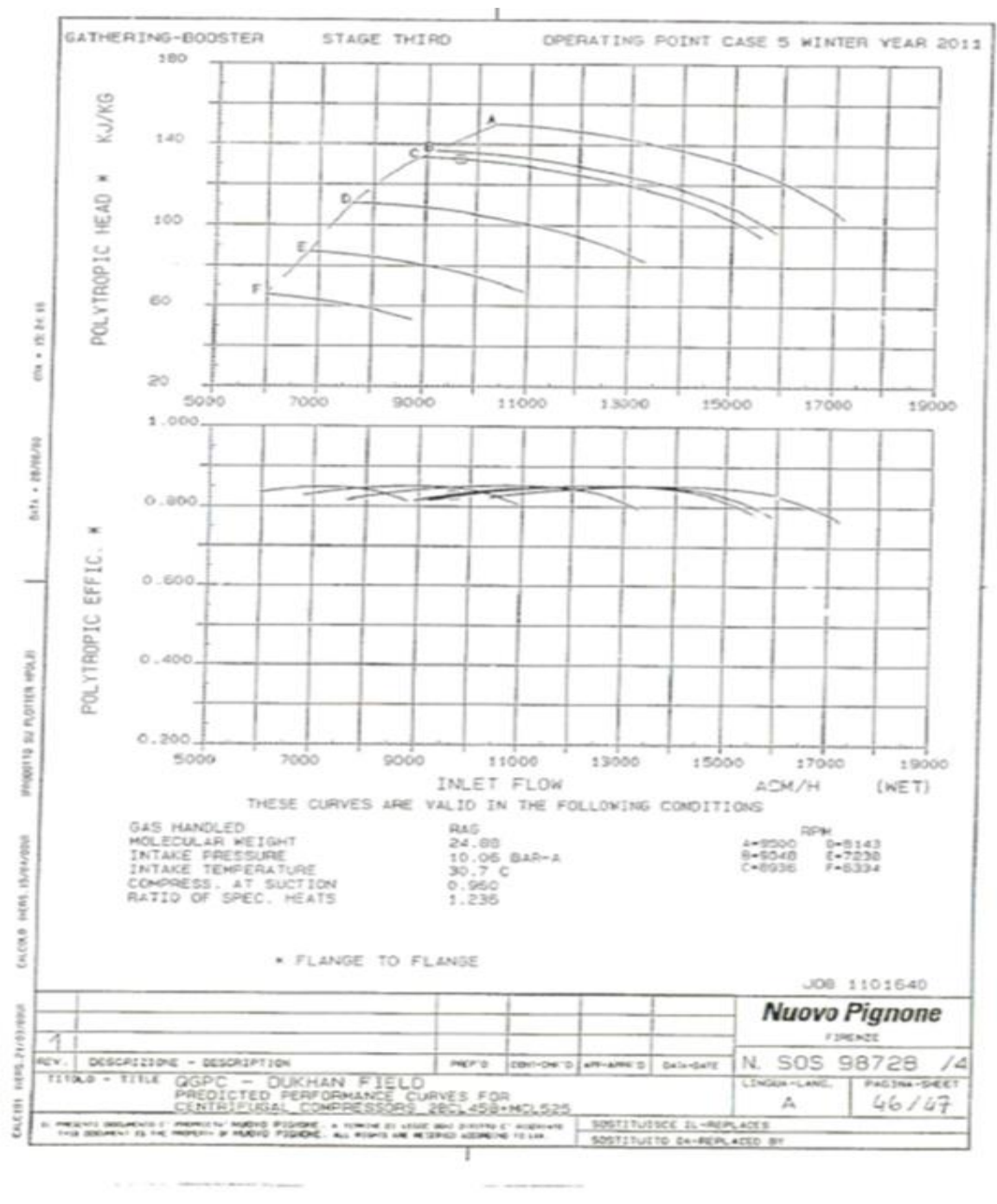


Figure B- 1 Polytropic head and polytropic efficiency from OEM

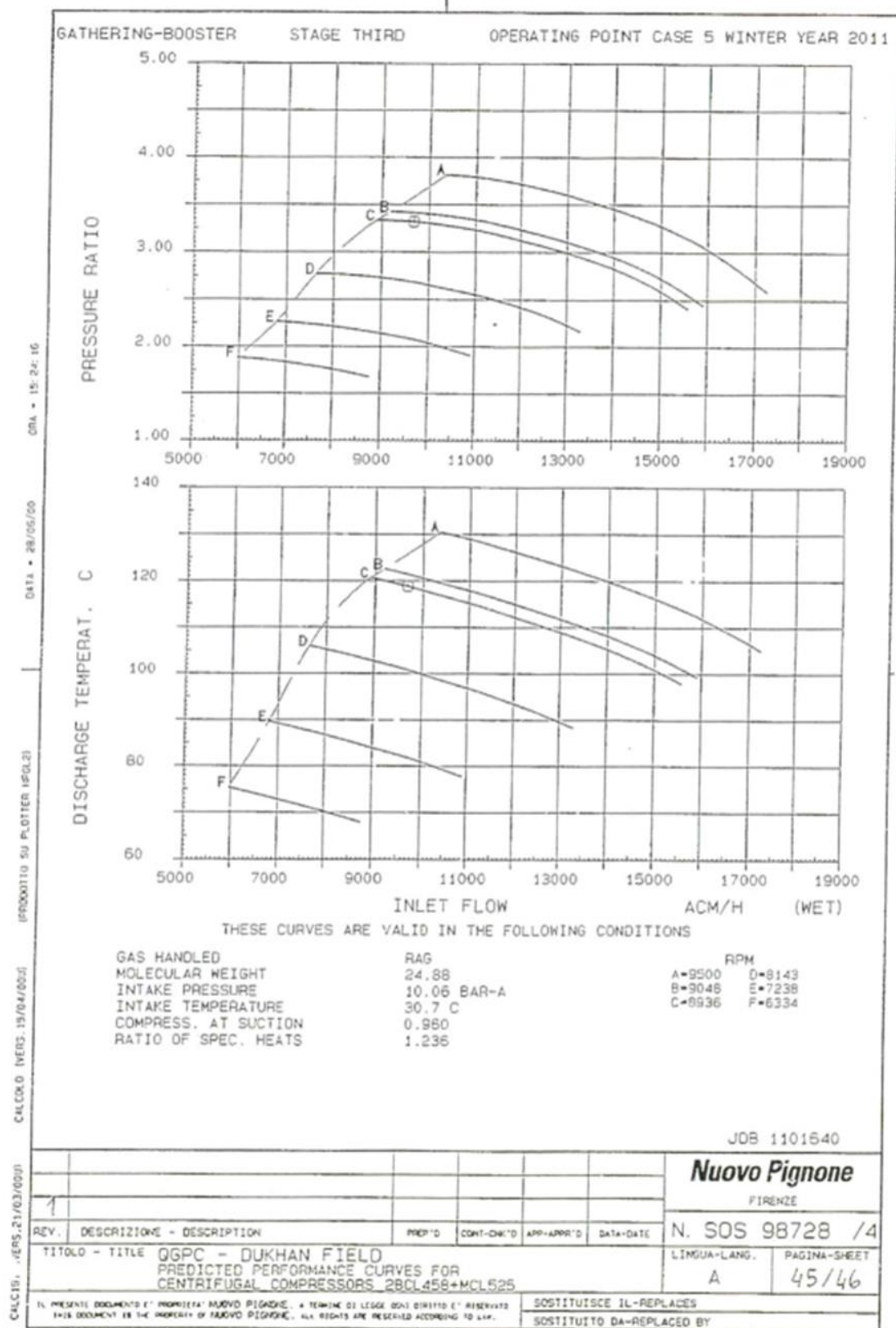


Figure B- 2 Pressure ratio and discharge temperature from OEM

Appendix C : TURBOMATCH design point input file

Below are some of the input files for the FORTRAN programme instruction for the TURBOMATCH which simulate the engine thermodynamic performance at DP and OD conditions. The results are written to a separate TMR file.

Appendix C1 IGTA Simulation code

IGTA industrial Gas Turbine Simulation Code

Inspired by SGT-400

Author: Aziaka, Duabari Silas

////

OD SI GM VA FP

-1

-1

INTAKE S1,2 D1-6 R200

COMPRES2,3 D7-18 R205 V7 V8

PREMAS S3,10,4 D19-22

BURNER S4,5 D23-30 R215 W5,6

MIXEES S5,10,6

TURBIN S6,7 D31-45 V32

TURBIN S7,8 D46-60 V47

NOZCON S8,9,1 D61-62 R220

PLOTBD D50,57

PERFOR S1,0,0 D46,64-66,220,200,215,0,0,0,0

CODEND

DATA ITEMS ////

1 0.0 ! INTAKE: Altitude [m]

2 0.0 ! Deviation from ISA temperature [K]

3 0.0 ! Mach number

4 0.995 ! Pressure recovery, according to USAF

5 0.0 ! Deviation from ISA pressure [atm]

6 60.0 ! Relative humidity [%]

7 -1.0	! COMPRESSOR I: $Z = (R - R[\text{choke}]) / (R[\text{surge}] - R[\text{choke}])$
8 0.876	! Relative rotational speed PCN
9 16.8	! DP Pressure ratio
10 0.87	! DP ETA
11 0.0	! Error selection
12 3.0	! Compressor map number
13 1.0	! Shaft number
14 1.0	! PR degradation scaling factor
15 1.0	! NDMF degradation scaling factor
16 1.0	! ETA degradation scaling factor
17 0.0	! Effective component volume [m ³] (only for transient)
18 0.0	! Stator angle (VSV) relative to DP
19 0.01	! PREMAS: LAMDA W Cooling bypass (Wout1/Win)
20 0.0	! DELTA W
21 1.0	! LAMBDA P
22 0.0	! DELTA P
23 0.05	! COMBUSTOR: Pressure loss (=DP/P inlet total)
24 0.999	! Combustion efficiency
25 -1.0	! Fuel flow (if -1.0, TET must be defined)
26 0.0	! (>0) Water flow [kg s ⁻¹ or lb s ⁻¹] or (<0) WAR
27 288	! Temperature of water stream [K]
28 0.0	! Phase of water (0=liquid, 1=vapour)
29 1.0	! ETA degradation scaling factor
30 -1.0	! Effective component volume [m ³] (only for transient)

31 0.0	! COMPRESSOR TURBINE: Auxiliary or power output [W]
32 -1.0	! Relative to max enthalpy drop to temperature ratio:ZT
33 1.0	! Relative non-dim speed CN
34 0.95	! DP ETA
35 -1.0	! Relative non-dim PCN (= -1 for compressor turbine)
36 1.0	! Shaft Number (for power turbine, the value 0 is used)
37 4.0	! Turbine map number
38 -1.0	! Power law index "n" (POWER = PCN^n)
39 1.0	! TF degradation scaling factor
40 1.0	! DH degradation scaling factor
41 1.0	! ETA degradation scaling factor
42 0.0	! Rotor rotational speed [RPS] (only for transient)
43 0.0	! Rotor moment of inertia [kg.m^2] (only for transient)

44 0.0 ! Effective component volume [m^3] (only for transient)
 45 0.0 ! NGV angle, relative to D.P.

 46 13400000.0 ! FPT TURBINE: Auxiliary or power output [W]
 47 0.8 ! Relative to max enthalpy drop to temperature ratio:ZT
 48 1.0 ! Relative non-dim speed CN
 49 0.95 ! DP ETA
 50 1.0 ! Relative non-dim PCN (= -1 for compressor turbine)
 51 0.0 ! Shaft Number (for power turbine, the value 0 is used)
 52 4.0 ! Turbine map number
 53 -1.0 ! Power law index "n" (POWER = PCN^n)
 54 1.0 ! TF degradation scaling factor
 55 1.0 ! DH degradation scaling factor
 56 1.0 ! ETA degradation scaling factor
 57 200 ! Rotor rotational speed [RPS] (only for transient)
 58 20.0 ! Rotor moment of inertia [kg.m^2] (only for transient)
 59 -1.0 ! Effective component volume [m^3] (only for transient)
 60 0.0 ! NGV angle, relative to D.P.

 61 -1.0 ! CONVERGENT NOZZLE: = "-1" exit area is fixed
 62 1.0 ! Scaling factor

 63 1.0 ! ENGINE RESULTS: Power output - Power or Power turbine
 64 0.0 ! Propeller efficiency (= -1 for turbojet/turbofan)
 65 0.0 ! Scaling index ("1" = scaling needed, "0" = no scaling)
 66 0.0 ! Required DP shaft power (no scaling is needed)

 -1
 1 2 38.64 ! item 2 at station 1 = Mass flow(kg/s)
 5 6 1440.0 ! item 6 at station 5 = Total temperature (K)
 -1 ! End of DP data
 -3

Appendix C2 IGTB Simulation Code

IGTB industrial Gas Turbine Simulation Code

Inspired by SGT-500

Author: Aziaka, Duabari Silas

////

OD SI GM VA FP

-1

-1|

INTAKE S1,2 D1-6 R200

COMPRES S2,3 D7-18 R205 V7 V8

COMPRES S3,4 D19-30 R205 V19 V20

PREMAS S4,25,5 D31-34

PREMAS S5,26,6 D35-38

BURNER S6,7 D39-46 R215 W7,6

MIXEES S7,26,8

TURBIN S8,9 D47-61 V48

MIXEES S9,25,10

TURBIN S10,11 D62-76 V63

TURBIN S11,12 D77-91 V78

NOZCON S12,13,1 D92,93 R220

PERFOR S1,0,0 D77,94-96,220,200,215,0,0,0,0,0

CODEND

DATA ITEMS ////

1 0.0 ! INTAKE: Altitude [m]

2 0.0 ! Deviation from ISA temperature [K]

3 0.0 ! Mach number

4 0.9954 ! Pressure recovery, according to USAF

5 0.0 ! Deviation from ISA pressure [atm]

6 60.0 ! Relative humidity [%]

7 -1.0 ! COMPRESSOR - FAN: $Z = (R - R[\text{choke}]) / (R[\text{surge}] - R[\text{choke}])$ (if =-1. the default value 0.85 is invoked)

8 -1.0 ! Relative rotational speed PCN

9 2.167 ! DP Pressure ratio

10 0.80	! isentropic efficiency
11 0.0	! Error selection
12 5.0	! Compressor Map Number
13 1.0	! Shaft number
14 1.0	! Scaling factor of Pressure Ratio – Degradation factor
15 1.0	! Scaling factor of Non-D Mass Flow – Degradation factor
16 1.0	! Scaling factor of ETAc is (Compressor isentropic efficiency) – Degradation factor
17 1.5	! Effective component volume [m ³]
18 0.0	! Stator angle (VSV) relative to DP
19 -1.0	! COMPRESSOR HP: $Z = (R-R[\text{choke}]) / (R[\text{surge}] - R[\text{choke}])$ (if =-1. the default value 0.85 is invoked)
20 -1.0	! Relative rotational speed PCN
21 6.0	! DP Pressure ratio
22 0.80	! isentropic efficiency
23 1.0	! Error selection
24 5.0	! Compressor Map Number
25 2.0	! Shaft number
26 1.0	! Scaling factor of Pressure Ratio – Degradation factor
27 1.0	! Scaling factor of Non-D Mass Flow – Degradation factor
28 1.0	! Scaling factor of ETAc is (Compressor isentropic efficiency) – Degradation factor
29 -1.0	! Effective component volume [m ³]
30 0.0	! Stator angle (VSV) relative to DP
31 0.05	! BYPASS: (Wout/Win) BPR = 6.4 CRUISE!
32 0.0	! DELTA W
33 1.0	! LAMBDA P
34 0.0	! DELTA P
35 0.05	! PREMAS: LAMDA W Cooling bypass (Wout/Win)
36 0.0	! DELTA W
37 1.0	! LAMBDA P
38 0.0	! DELTA P

39 0.05	! COMBUSTOR: Pressure loss (=Total pressure loss/Inlet total pressure)
40 0.999	! Combustion efficiency
41 -1.0	! Fuel flow (If -1. is given the TET must be determined in the station vector)
42 0.0	! (>0) Water flow [kg s ⁻¹ or lb s ⁻¹] or (<0) Water to air ratio
43 288.0	! Temperature of water stream [K]
44 0.0	! Phase of water (0=liquid, 1=vapour)
45 1.0	! Scaling factor of ETAb (combustion efficiency) – Degradation factor
46 -1.0	! Effective component volume [m ³]
47 0.0	! COMPRESSOR TURBINE: Auxiliary or power output [W]
48 -1	! Relative non-dimensional massflow W/Wmax (if = -1, value 0.8 is invoked)
49 -1	! Relative non-dimensional speed CN (if = -1, value 0.6 is invoked)
50 0.88	! Design isentropic efficiency
51 -1.0	! Relative non-dimensional speed PCN (= -1 for compressor turbine)
52 2.0	! Shaft Number (for power turbine, the value "0." is used)
53 3.0	! Turbine map number
54 -1.0	! Power law index "n" (POWER = PCN ⁿ) If = -1, power is assumed to be a constant
55 1.0	! Scaling factor of TF (non-D inlet mass flow) – Degradation factor
56 1.0	! Scaling factor of DH (enthalpy change) – Degradation factor
57 1.0	! Scaling factor of ETAc is (Turbine isentropic efficiency) – Degradation factor
58 200	! Rotor rotational speed [RPS]
59 15.0	! Rotor moment of inertia [kg.m ²]
60 -1.0	! Effective component volume [m ³]
61 0.0	! NGV angle, relative to D.P.
62 0.0	! COMPRESSOR TURBINE: Auxiliary or power output [W]
63 -1	! Relative non-dimensional massflow W/Wmax (if = -1, value 0.8 is invoked)
64 -1	! Relative non-dimensional speed CN (if = -1, value 0.6 is invoked)
65 0.88	! Design isentropic efficiency
66 -1.0	! Relative non-dimensional speed PCN (= -1 for compressor turbine)
67 1.0	! Shaft Number (for power turbine, the value "0." is used)
68 3.0	! Turbine map number
69 -1.0	! Power law index "n" (POWER = PCN ⁿ) If = -1, power is assumed to be a constant
70 1.0	! Scaling factor of TF (non-D inlet mass flow) – Degradation factor

71 1.0 ! Scaling factor of DH (enthalpy change) – Degradation factor
 72 1.0 ! Scaling factor of ETAc is (Turbine isentropic efficiency) – Degradation factor
 73 200 ! Rotor rotational speed [RPS]
 74 15.0 ! Rotor moment of inertia [kg.m²]
 75 -1.0 ! Effective component volume [m³]
 76 0.0 ! NGV angle, relative to D.P.

 77 19300000.0 ! POWER TURBINE: Auxiliary or power output [W]
 78 -1 ! Relative non-dimensional massflow W/Wmax (if = -1, value 0.8 is invoked)
 79 -1 ! Relative non-dimensional speed CN (if = -1, value 0.6 is invoked)
 80 0.88 ! Design isentropic efficiency
 81 1.0 ! Relative non-dimensional speed PCN (= -1 for compressor turbine)
 82 0.0 ! Shaft Number (for power turbine, the value "0." is used)
 83 3.0 ! Turbine map number
 84 -1.0 ! Power law index "n" (POWER = PCNⁿ) If = -1, power is assumed to be a constant
 85 1.0 ! Scaling factor of TF (non-D inlet mass flow) – Degradation factor
 86 1.0 ! Scaling factor of DH (enthalpy change) – Degradation factor
 87 1.0 ! Scaling factor of ETAc is (Turbine isentropic efficiency) – Degradation factor
 88 200 ! Rotor rotational speed [RPS]
 89 20.0 ! Rotor moment of inertia [kg.m²]
 90 -1.0 ! Effective component volume [m³]
 91 0.0 ! NGV angle, relative to D.P.

 92 -1.0 ! CONVERGENT NOZZLE: = "-1" exit area is fixed
 93 1.0 ! Scaling factor

 94 1.0 ! ENGINE RESULTS: Propeller efficiency (= -1 for turbojet/turbofan)
 95 0.0 ! Scaling index ("1" = scaling needed, "0" = no scaling)
 96 0.0 ! Required DP net thrust(Turbojet,turbofan) or shaft power (Turboprop.turboshaft)
 ! = 0 if Scaling index = 0
 -1
 1 2 96.75 ! item 2 at station 1 = Mass flow(kg/s)
 7 6 1300.0 ! item 6 at station 7 = Total temperature (K)
 -1 ! End of DP data
 -3

Appendix D OD results for IGTB

The result for the clean engine of the IGTB was shown alongside with the degraded engine. The engine was subjected to a range of temperature “between” 10⁰c to 45⁰c. Also, for the degraded model, a 3% reduction in isentropic efficiency and flow capacity was implanted in both compressor and turbine together. TETs was the handle during the clean and degraded engine simulation. The TETs taken was between “1200⁰k to 1450⁰k” with an interval of 50⁰k.

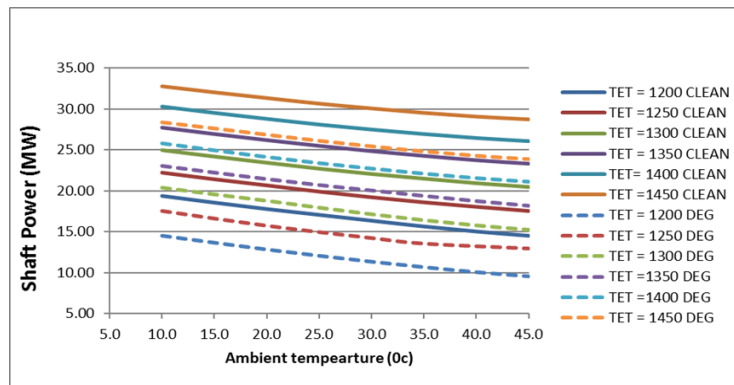


Figure D- 1 Shaft power vs Ambient temperature

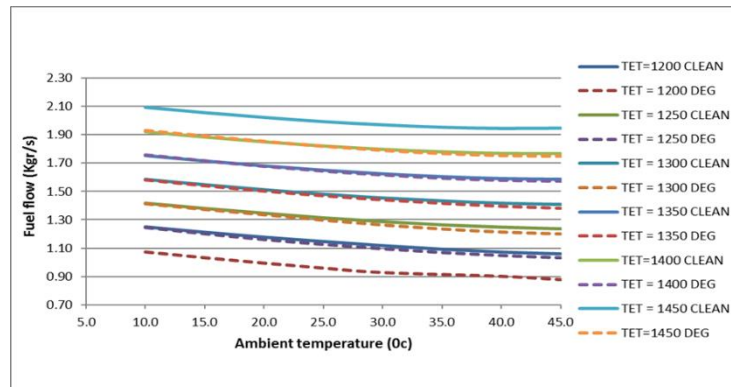


Figure D- 2 Fuel flow vs Ambient temperature

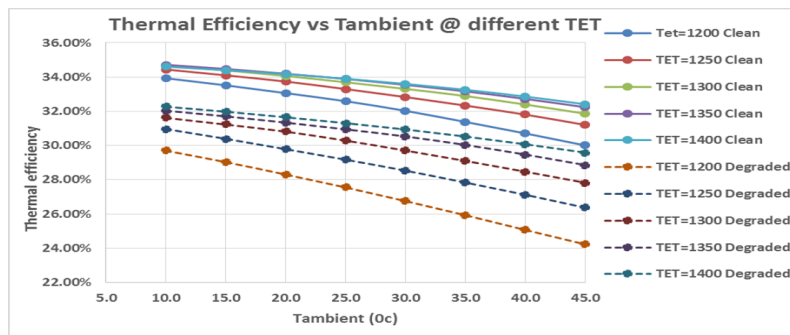


Figure D- 3 Thermal efficiency vs Ambient temperature

Appendix D2: Effects of Altitude and Degradation on the Engine

Effects of Altitude on the IGTB engine model was investigated. The altitude value above the ISA SLS condition under consideration was from 0 m to 2000 m. Also, 4% reduction in flow capacity and isentropic efficiency was implanted in the engine model. Under this simulation, only the compressor was degraded

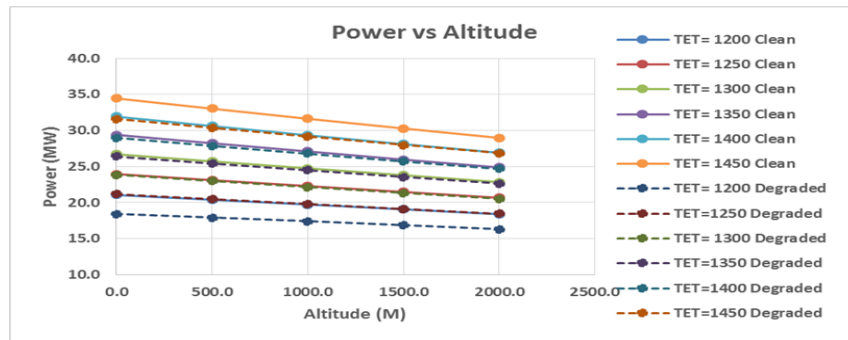


Figure D2- 1 Power vs Altitude

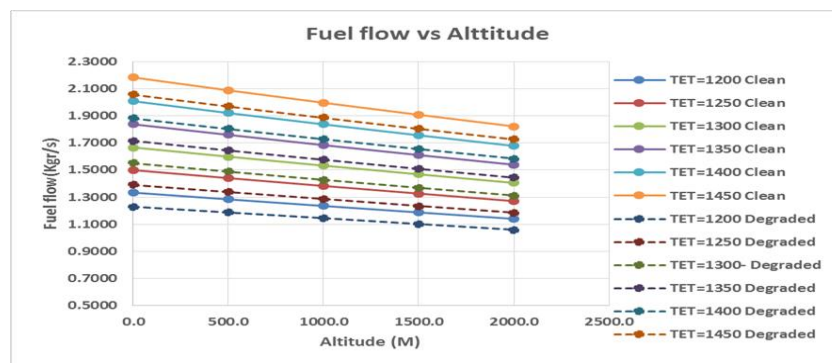


Figure D2- 2 Fuel flow vs Altitude

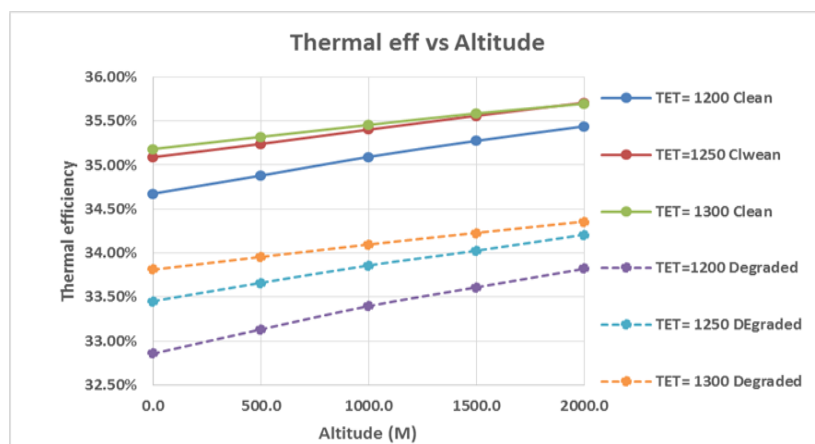


Figure D2- 3 Thermal efficiency vs Altitude

Appendix E : Input file for the Degraded Scenario

Appendix E1: Pessimistic scenario with clean condition

IGTD Degraded code for Pessimistic scenario in the Dry Season for 30 years.

Part-Load Simulation

Author: Aziaka Duabari Silas

////

OD SI GM VA FP

-1

-1

INTAKE S1,2 D1-6 R200

COMPRES S2,3 D7-18 R205 V7 V8

COMPRES S3,4 D19-30 R205 V19 V20

PREMAS S4,25,5 D31-34

PREMAS S5,26,6 D35-38

BURNER S6,7 D39-46 R215 W7,6

MIXEES S7,26,8

TURBIN S8,9 D47-61 V48

MIXEES S9,25,10

TURBIN S10,11 D62-76 V63

TURBIN S11,12 D77-91 V78

NOZCON S12,13,1 D92,93 R220

PERFOR S1,0,0 D77,94-96,220,200,215,0,0,0,0,0

CODEND

DATA ITEMS ////

1 0.0 ! INTAKE: Altitude [m]

2 0.0 ! Deviation from ISA temperature [K]

3 0.0 ! Mach number

4 0.9954 ! Pressure recovery, according to USAF

5 0.0 ! Deviation from ISA pressure [atm]

6 60.0 ! Relative humidity [%]

7 -1.0 ! COMPRESSOR - FAN: $Z = (R-R[\text{choke}]) / (R[\text{surge}] - R[\text{choke}])$ (if $= -1$, the default value 0.85 is invoked)

8 -1.0 ! Relative rotational speed PCN

9 2.538 ! DP Pressure ratio

10 0.87	! isentropic efficiency
11 0.0	! Error selection
12 5.0	! Compressor Map Number
13 1.0	! Shaft number
14 1.0	! Scaling factor of Pressure Ratio – Degradation factor
15 1.0	! Scaling factor of Non-D Mass Flow – Degradation factor
16 1.0	! Scaling factor of ETAc is (Compressor isentropic efficiency) – Degradation factor
17 1.5	! Effective component volume [m^3]
18 0.0	! Stator angle (VSV) relative to DP
19 -1.0 invoked)	! COMPRESSOR HP: $Z = (R-R[\text{choke}]) / (R[\text{surge}] - R[\text{choke}])$ (if ==-1. the default value 0.85 is invoked)
20 -1.0	! Relative rotational speed PCN
21 8.786	! DP Pressure ratio
22 0.87	! isentropic efficiency
23 1.0	! Error selection
24 5.0	! Compressor Map Number
25 2.0	! Shaft number
26 1.0	! Scaling factor of Pressure Ratio – Degradation factor
27 1.0	! Scaling factor of Non-D Mass Flow – Degradation factor
28 1.0	! Scaling factor of ETAc is (Compressor isentropic efficiency) – Degradation factor
29 -1.0	! Effective component volume [m^3]
30 0.0	! Stator angle (VSV) relative to DP
31 0.085	! BYPASS: (Wout/Win) BPR = 6.4 CRUISE!
32 0.0	! DELTA W
33 1.0	! LAMBDA P
34 0.0	! DELTA P
35 0.0725	! PREMAS: LAMDA W Cooling bypass (Wout/Win)
36 0.0	! DELTA W
37 1.0	! LAMBDA P
38 0.0	! DELTA P

39 0.05 ! COMBUSTOR: Pressure loss (=Total pressure loss/Inlet total pressure)
 40 0.999 ! Combustion efficiency
 41 -1.0 ! Fuel flow (If -1. is given the TET must be determined in the station vector)
 42 0.0 ! (>0) Water flow [kg s⁻¹ or lb s⁻¹] or (<0) Water to air ratio
 43 288.0 ! Temperature of water stream [K]
 44 0.0 ! Phase of water (0=liquid, 1=vapour)
 45 1.0 ! Scaling factor of ETAb (combustion efficiency) – Degradation factor
 46 -1.0 ! Effective component volume [m³]

 47 0.0 ! COMPRESSOR TURBINE: Auxiliary or power output [W]
 48 -1 ! Relative non-dimensional massflow W/Wmax (if = -1, value 0.8 is invoked)
 49 -1 ! Relative non-dimensional speed CN (if = -1, value 0.6 is invoked)
 50 0.88 ! Design isentropic efficiency
 51 -1.0 ! Relative non-dimensional speed PCN (= -1 for compressor turbine)
 52 2.0 ! Shaft Number (for power turbine, the value "0." is used)
 53 3.0 ! Turbine map number
 54 -1.0 ! Power law index "n" (POWER = PCNⁿ) If = -1, power is assumed to be a constant
 55 1.0 ! Scaling factor of TF (non-D inlet mass flow) – Degradation factor
 56 1.0 ! Scaling factor of DH (enthalpy change) – Degradation factor
 57 1.0 ! Scaling factor of ETAc is (Turbine isentropic efficiency) – Degradation factor
 58 200 ! Rotor rotational speed [RPS]
 59 15.0 ! Rotor moment of inertia [kg.m²]
 60 -1.0 ! Effective component volume [m³]
 61 0.0 ! NGV angle, relative to D.P.

 62 0.0 ! COMPRESSOR TURBINE: Auxiliary or power output [W]
 63 0.8 ! Relative non-dimensional massflow W/Wmax (if = -1, value 0.8 is invoked)
 64 0.6 ! Relative non-dimensional speed CN (if = -1, value 0.6 is invoked)
 65 0.89 ! Design isentropic efficiency
 66 -1.0 ! Relative non-dimensional speed PCN (= -1 for compressor turbine)
 67 1.0 ! Shaft Number (for power turbine, the value "0." is used)
 68 3.0 ! Turbine map number
 69 -1.0 ! Power law index "n" (POWER = PCNⁿ) If = -1, power is assumed to be a constant
 70 1.0 ! Scaling factor of TF (non-D inlet mass flow) – Degradation factor

71 1.0 ! Scaling factor of DH (enthalpy change) – Degradation factor
 72 1.0 ! Scaling factor of ETAc is (Turbine isentropic efficiency) – Degradation factor
 73 200 ! Rotor rotational speed [RPS]
 74 15.0 ! Rotor moment of inertia [kg.m^2]
 75 -1.0 ! Effective component volume [m^3]
 76 0.0 ! NGV angle, relative to D.P.

 77 15200000 ! POWER TURBINE: Auxiliary or power output [W]
 78 -1 ! Relative non-dimensional massflow W/Wmax (if = -1, value 0.8 is invoked)
 79 -1 ! Relative non-dimensional speed CN (if = -1, value 0.6 is invoked)
 80 0.8895 ! Design isentropic efficiency
 81 1.0 ! Relative non-dimensional speed PCN (= -1 for compressor turbine)
 82 0.0 ! Shaft Number (for power turbine, the value "0." is used)
 83 3.0 ! Turbine map umber
 84 -1.0 ! Power law index "n" (POWER = PCN^n) If = -1, power is assumed to be a constant
 85 1.0 ! Scaling factor of TF (non-D inlet mass flow) – Degradation factor
 86 1.0 ! Scaling factor of DH (enthalpy change) – Degradation factor
 87 1.0 ! Scaling factor of ETAc is (Turbine isentropic efficiency) – Degradation factor
 88 200 ! Rotor rotational speed [RPS]
 89 20.0 ! Rotor moment of inertia [kg.m^2]
 90 -1.0 ! Effective component volume [m^3]
 91 0.0 ! NGV angle, relative to D.P.

 92 -1.0 ! CONVERGENT NOZZLE: = "-1" exit area is fixed
 93 1.0 ! Scaling factor
 |
 94 1.0 ! ENGINE RESULTS: Propeller efficiency (= -1 for turbojet/turbofan)
 95 0.0 ! Scaling index ("1" = scalling needed, "0" = no scaling)
 96 0.0 ! Required DP net thrust(Turbojet,turbofan) or shaft power (Turboprop.turboshaft)
 ! = 0 if Scaling index = 0
 -1
 1 2 46.0 ! item 2 at station 1 = Mass flow(kg/s)
 7 6 1550.0 ! item 6 at station 7 = Total temperature (K)
 -1 ! End of DP data

2 20.95	
77 7203000.0	
-1	
-1	
2 19.95	
77 7203000.0	
-1	
-1	
2 19.95	
77 7203000.0	
-1	
-1	
2 24.2	
77 7203000.0	
-1	
-1	
2 28.2	
77 7203000.0	
-1	
-1	
2 29.95	
77 7203000.0	
-1	
-1	
2 26.7	
77 7203000.0	
-1	
-1	
2 22.7	
77 7203000.0	
-1	
-1	

Clean engine operation

2 20.95

77 7203000.0

15 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW

16 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN EFFICIENCY

27 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW

28 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN EFFICIENCY

55 0.975 !TURBINE#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW

57 0.975 !TURBINE#1 DETERIORATION OF 2.5% IN EFFICIENCY

70 0.975 !TURBINE#2 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW

72 0.975 !TURBINE#2 DETERIORATION OF 2.5% IN EFFICIENCY

85 0.975 !TURBINE#3 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW

87 0.975 !TURBINE#3 DETERIORATION OF 2.5% IN EFFICIENCY

-1

-1

2 19.95

77 7203000.0

77 7203000.0|

15 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW

16 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN EFFICIENCY

27 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW

28 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN EFFICIENCY

55 0.975 !TURBINE#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW

57 0.975 !TURBINE#1 DETERIORATION OF 2.5% IN EFFICIENCY

70 0.975 !TURBINE#2 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW

72 0.975 !TURBINE#2 DETERIORATION OF 2.5% IN EFFICIENCY

85 0.975 !TURBINE#3 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW

87 0.975 !TURBINE#3 DETERIORATION OF 2.5% IN EFFICIENCY

-1

-1

2 19.95

77 7203000.0

77 7203000.0

15 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW

16 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN EFFICIENCY

Degraded Condition for 30 years

27 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
 28 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN EFFICIENCY
 55 0.975 !TURBIN#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
 57 0.975 !TURBIN#1 DETERIORATION OF 2.5% IN EFFICIENCY
 70 0.975 !TURBIN#2 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
 72 0.975 !TURBIN#2 DETERIORATION OF 2.5% IN EFFICIENCY
 85 0.975 !TURBIN#3 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
 87 0.975 !TURBIN#3 DETERIORATION OF 2.5% IN EFFICIENCY
 -1
 -1
 2 24.2
 77 7203000.0
 77 7203000.0
 15 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
 16 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN EFFICIENCY
 27 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
 28 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN EFFICIENCY
 55 0.975 !TURBIN#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
 57 0.975 !TURBIN#1 DETERIORATION OF 2.5% IN EFFICIENCY
 70 0.975 !TURBIN#2 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
 72 0.975 !TURBIN#2 DETERIORATION OF 2.5% IN EFFICIENCY
 85 0.975 !TURBIN#3 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
 87 0.975 !TURBIN#3 DETERIORATION OF 2.5% IN EFFICIENCY
 -1
 -1
 2 28.2
 77 7203000.0
 77 7203000.0
 15 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
 16 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN EFFICIENCY
 27 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
 28 0.975 !COMPRES#1 DETERIORATION OF 2.5% IN EFFICIENCY
 55 0.975 !TURBIN#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
 57 0.975 !TURBIN#1 DETERIORATION OF 2.5% IN EFFICIENCY

70	0.975	!TURBIN#2 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
72	0.975	!TURBIN#2 DETERIORATION OF 2.5% IN EFFICIENCY
85	0.975	!TURBIN#3 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
87	0.975	!TURBIN#3 DETERIORATION OF 2.5% IN EFFICIENCY
-1		
-1		
2 29.95		
77 7203000.0		
77 7203000.0		
15	0.975	!COMPRES#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
16	0.975	!COMPRES#1 DETERIORATION OF 2.5% IN EFFICIENCY
27	0.975	!COMPRES#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
28	0.975	!COMPRES#1 DETERIORATION OF 2.5% IN EFFICIENCY
55	0.975	!TURBIN#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
57	0.975	!TURBIN#1 DETERIORATION OF 2.5% IN EFFICIENCY
70	0.975	!TURBIN#2 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
72	0.975	!TURBIN#2 DETERIORATION OF 2.5% IN EFFICIENCY
85	0.975	!TURBIN#3 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
87	0.975	!TURBIN#3 DETERIORATION OF 2.5% IN EFFICIENCY
-1		
-1		
2 26.7		
77 7203000.0		
77 7203000.0		
15	0.975	!COMPRES#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
16	0.975	!COMPRES#1 DETERIORATION OF 2.5% IN EFFICIENCY
27	0.975	!COMPRES#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
28	0.975	!COMPRES#1 DETERIORATION OF 2.5% IN EFFICIENCY
55	0.975	!TURBIN#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
57	0.975	!TURBIN#1 DETERIORATION OF 2.5% IN EFFICIENCY
70	0.975	!TURBIN#2 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
72	0.975	!TURBIN#2 DETERIORATION OF 2.5% IN EFFICIENCY
85	0.975	!TURBIN#3 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
87	0.975	!TURBIN#3 DETERIORATION OF 2.5% IN EFFICIENCY

-1
-1
2 22.7
77 7203000.0
77 7203000.0
15 0.975 !COMPRESS#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
16 0.975 !COMPRESS#1 DETERIORATION OF 2.5% IN EFFICIENCY
27 0.975 !COMPRESS#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
28 0.975 !COMPRESS#1 DETERIORATION OF 2.5% IN EFFICIENCY
55 0.975 !TURBINE#1 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
57 0.975 !TURBINE#1 DETERIORATION OF 2.5% IN EFFICIENCY
70 0.975 !TURBINE#2 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
72 0.975 !TURBINE#2 DETERIORATION OF 2.5% IN EFFICIENCY
85 0.975 !TURBINE#3 DETERIORATION OF 2.5% IN NON-DIMENSIONAL MASS FLOW
87 0.975 !TURBINE#3 DETERIORATION OF 2.5% IN EFFICIENCY
-1
-1
2 20.95
77 7203000.0
15 0.9675 !COMPRESS#1 DETERIORATION OF 3.25% IN NON-DIMENSIONAL MASS FLOW
16 0.9675 !COMPRESS#1 DETERIORATION OF 3.25% IN EFFICIENCY
27 0.9675 !COMPRESS#1 DETERIORATION OF 3.25% IN NON-DIMENSIONAL MASS FLOW
28 0.9675 !COMPRESS#1 DETERIORATION OF 3.25% IN EFFICIENCY
55 0.9675 !TURBINE#1 DETERIORATION OF 3.25% IN NON-DIMENSIONAL MASS FLOW
57 0.9675 !TURBINE#1 DETERIORATION OF 3.25% IN EFFICIENCY
70 0.9675 !TURBINE#2 DETERIORATION OF 3.25% IN NON-DIMENSIONAL MASS FLOW
72 0.9675 !TURBINE#2 DETERIORATION OF 3.25% IN EFFICIENCY
85 0.9675 !TURBINE#3 DETERIORATION OF 3.25% IN NON-DIMENSIONAL MASS FLOW
87 0.9675 !TURBINE#3 DETERIORATION OF 3.25% IN EFFICIENCY
-1
-1

↓
Till 30 years

Appendix E2: Implanted % degradation for all seasons and scenarios for the baseline condition

Year	WINTER			DRY			Hot		
	Opti	med	Pess	Opti	med	pess	Opti	med	pess
	(%)			(%)			(%)		
1	0.25	1.25	2.25	0.5	1.5	2.5	0.75	1.75	2.75
2	1	2	3	1.25	2.25	3.25	1.5	2.5	3.5
3	1.75	2.75	3.75	2	3	4	2.25	3.25	4.25
4	0.035	0.08	0.15	0.05	0.1	0.175	0.053	0.12	0.193
5	1.1	2.1	3.14	1.35	2.35	3.39	1.6	2.6	3.64
6	1.85	2.85	3.89	2.1	3.1	4.14	2.35	3.35	4.39
7	0.045	0.11	0.18	0.065	0.13	0.205	0.073	0.15	0.223
8	1.2	2.2	3.28	1.45	2.45	3.53	1.7	2.7	3.78
9	1.95	2.95	4.03	2.2	3.2	4.28	2.45	3.45	4.53
10	0.055	0.14	0.21	0.08	0.16	0.235	0.093	0.18	0.253
11	1.3	2.3	3.42	1.55	2.55	3.67	1.8	2.8	3.92
12	2.05	3.05	4.17	2.3	3.3	4.42	2.55	3.55	4.67
13	0.065	0.17	0.24	0.095	0.19	0.265	0.113	0.21	0.283
14	1.4	2.4	3.56	1.65	2.65	3.81	1.9	2.9	4.06
15	2.15	3.15	4.31	2.4	3.4	4.56	2.65	3.65	4.81
16	0.075	0.2	0.27	0.11	0.22	0.295	0.133	0.24	0.313
17	1.5	2.5	3.7	1.75	2.75	3.95	2	3	4.2
18	2.25	3.25	4.45	2.5	3.5	4.7	2.75	3.75	4.95
19	0.085	0.23	0.3	0.125	0.25	0.325	0.153	0.27	0.343
20	1.6	2.6	3.84	1.85	2.85	4.09	2.1	3.1	4.34
21	2.35	3.35	4.59	2.6	3.6	4.84	2.85	3.85	5.09
22	0.095	0.26	0.33	0.14	0.28	0.355	0.173	0.3	0.373
23	1.7	2.7	3.98	1.95	2.95	4.23	2.2	3.2	4.48
24	2.45	3.45	4.73	2.7	3.7	4.98	2.95	3.95	5.23
25	0.105	0.29	0.36	0.155	0.31	0.385	0.193	0.33	0.403
26	1.8	2.8	4.12	2.05	3.05	4.37	2.3	3.3	4.62
27	2.55	3.55	4.87	2.8	3.8	5.12	3.05	4.05	5.37
28	0.115	0.32	0.39	0.17	0.34	0.415	0.213	0.36	0.433
29	1.9	2.9	4.26	2.15	3.15	4.51	2.4	3.4	4.76
30	2.65	3.65	5.01	2.9	3.9	5.26	3.15	4.15	5.51

Appendix F : Selected compressor based on GC calculated power

Table F- 1 Selected compressor with calculated values

Compressor Stations Number	Highest GC power calculated (kW)	Gas Compressor selected in the market	Modelled Gas Turbines
1	9323	Dresser-Rand Datum C (16964 HP)	IGTA1 (13.4 MW)
2	12606	Dresser-Rand Datum C (16964 HP)	IGTD1 (15.2 MW)
3	14802	Dresser-Rand Datum C (20120 HP)	IGTC1 (17MW)
4	14411	Dresser-Rand Datum C (20120 HP)	IGTC2 (17MW)
5	14461	Dresser-Rand Datum C (20120 HP)	IGTC3 (17MW)
6	14467	Dresser-Rand Datum C (20120 HP)	IGTC4 (17MW)
7	13966	Dresser-Rand Datum C (20120 HP)	IGTC5 (17MW)
8	15968	Dresser-Rand Datum C (20120 HP)	IGTB1 (19.3MW)
9	14897	GE HE-S (22000 H11P)	IGTB2 (19.3MW)
10	15046	GE HE-S (22000 HP)	IGTB3 (19.3MW)
11	13311	Dresser-Rand Datum C (20120 HP)	IGTC6 (17MW)
12	13412	Dresser-Rand Datum C (20120 HP)	IGTC7 (17MW)
13	14763	Dresser-Rand Datum C (20120 HP)	IGTC8 (17MW)
14	11252	Dresser-Rand Datum C (16964 HP)	IGTD2 (15.2 MW)
15	13739	Dresser-Rand Datum C (20120 HP)	IGTC9 (17MW)
16	15118	GE HE-S (22000 HP)	IGTB4 (19.3MW)
17	14975	GE HE-S (22000 HP)	IGTB5 (19.3MW)
18	10292	Dresser-Rand Datum C (16964 HP)	IGTA2 (13.4 MW)

Appendix G Cost components of the O&M cost

Appendix G showed the different cost component that makes up the total O&M cost for the dry and hot season.

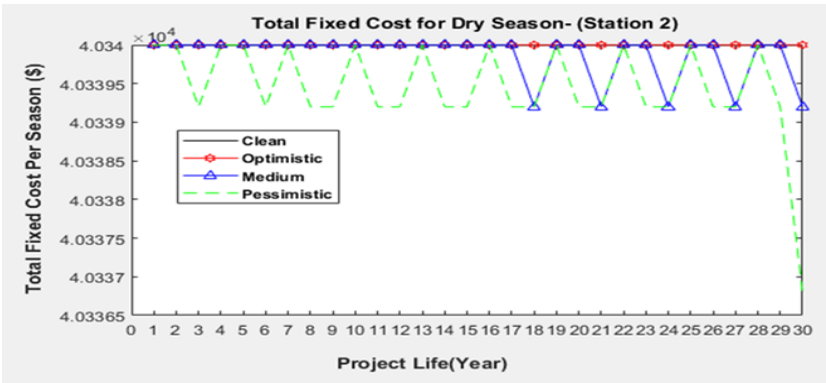


Figure G- 1 Fixed costs for Dry season –CS 2

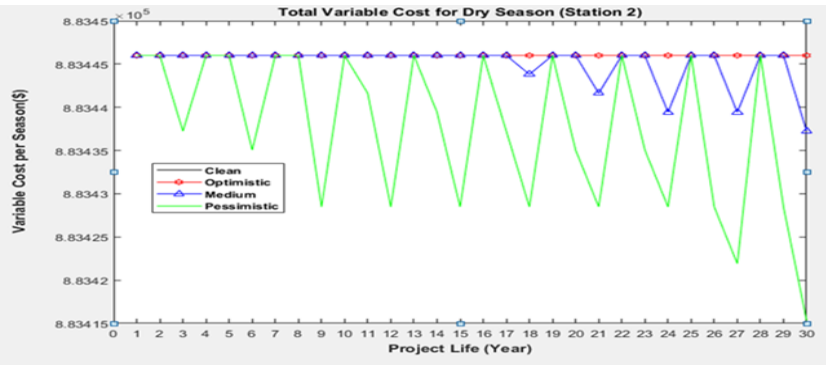


Figure G- 2 Variable costs for Dry season –CS 2

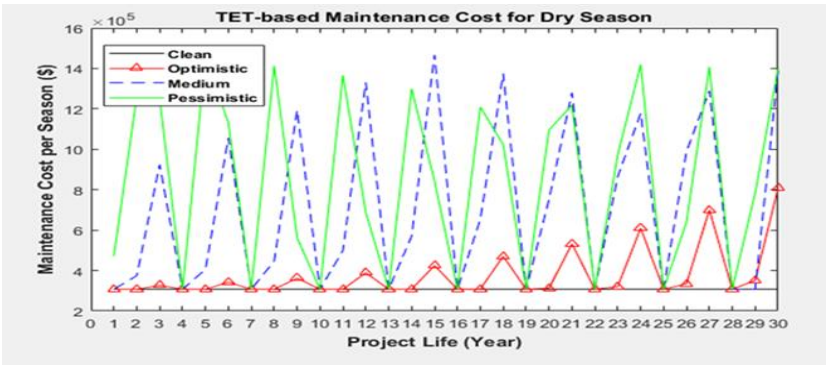


Figure G- 3 TET-based cost for dry season –CS 2

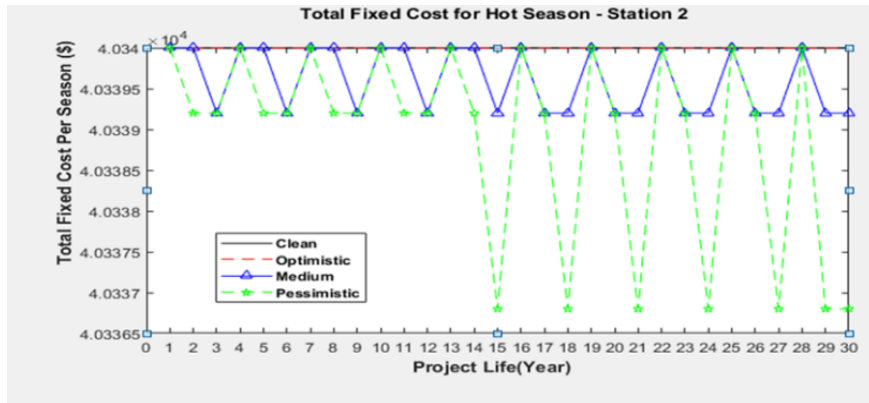


Figure G- 4 Fixed cost for dry season –CS 2

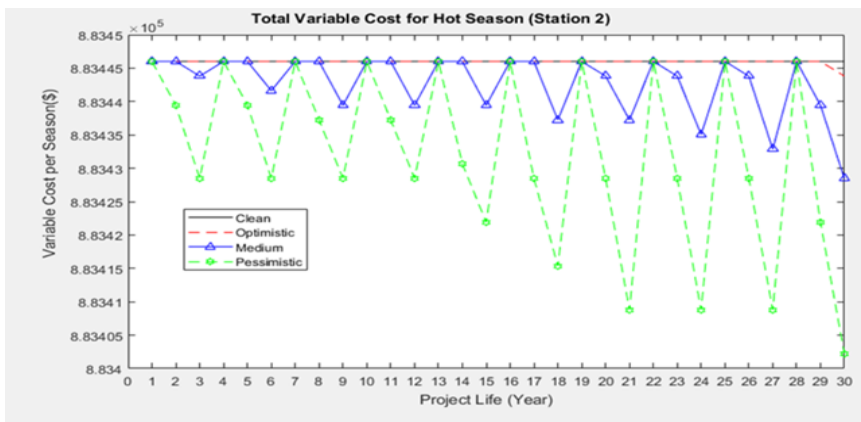


Figure G- 5 Variable costs for hot season –CS 2

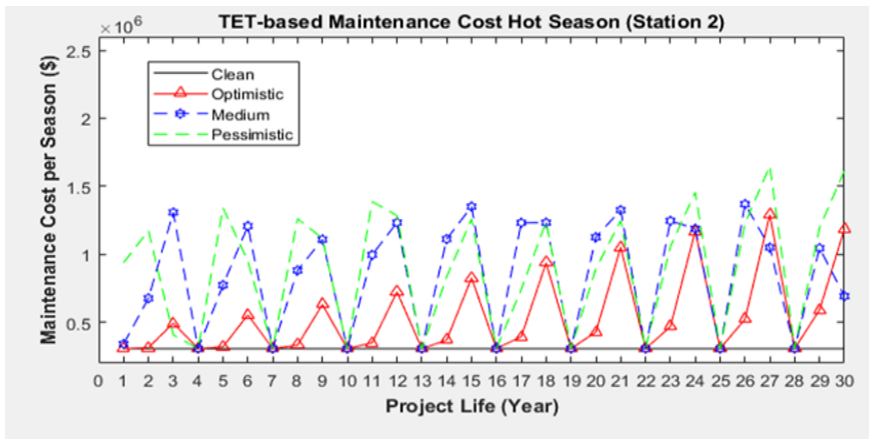


Figure G- 6 TET-based cost for hot season –CS 2

Appendix H The implemented percentage value for the re-schedule overhaul

This is the percentage values that were used to reduce the flow capacity and the component efficiency of the GT engines during the modelling process.

Table F 1 Implemented values of degradation for ROH

	WINTER			DRY			Hot		
	Opti	Med (%)	pess	Opti	Med (%)	pess	Opti	Med (%)	pess
1	0.25	1.25	2.25	0.5	1.5	2.5	0.75	1.75	2.75
2	1	2	3	1.25	2.25	3.25	1.5	2.5	3.5
3	0.025	0.06	0.11	0.03	0.08	0.15	0.035	0.1	0.17
4	1.1	2.1	3.1	1.35	2.35	3.35	1.6	2.6	3.6
5	0.035	0.08	0.135	0.04	0.1	0.175	0.045	0.12	0.195
6	1.2	2.2	3.2	1.45	2.45	3.45	1.7	2.7	3.7
7	0.045	0.1	0.16	0.05	0.12	0.2	0.055	0.14	0.22
8	1.3	2.3	3.3	1.55	2.55	3.55	1.8	2.8	3.8
9	0.055	0.12	0.185	0.06	0.14	0.225	0.065	0.16	0.245
10	1.4	2.4	3.4	1.65	2.65	3.65	1.9	2.9	3.9
11	0.065	0.14	0.21	0.07	0.16	0.25	0.075	0.18	0.27
12	1.5	2.5	3.5	1.75	2.75	3.75	2	3	4
13	0.075	0.16	0.235	0.08	0.18	0.275	0.085	0.2	0.295
14	1.6	2.6	3.6	1.85	2.85	3.85	2.1	3.1	4.1
15	0.085	0.18	0.26	0.09	0.2	0.3	0.095	0.22	0.32
16	1.7	2.7	3.7	1.95	2.95	3.95	2.2	3.2	4.2
17	0.095	0.2	0.285	0.1	0.22	0.325	0.105	0.24	0.345
18	1.8	2.8	3.8	2.05	3.05	4.05	2.3	3.3	4.3
19	0.105	0.22	0.31	0.11	0.24	0.35	0.115	0.26	0.37
20	1.9	2.9	3.9	2.15	3.15	4.15	2.4	3.4	4.4
21	0.115	0.24	0.335	0.12	0.26	0.375	0.125	0.28	0.395
22	2	3	4	2.25	3.25	4.25	2.5	3.5	4.5
23	0.125	0.26	0.36	0.13	0.28	0.4	0.135	0.3	0.42
24	2.1	3.1	4.1	2.35	3.35	4.35	2.6	3.6	4.6
25	0.135	0.28	0.385	0.14	0.3	0.425	0.145	0.32	0.445
26	2.2	3.2	4.2	2.45	3.45	4.45	2.7	3.7	4.7
27	0.145	0.3	0.41	0.15	0.32	0.45	0.155	0.34	0.47
28	2.3	3.3	4.3	2.55	3.55	4.55	2.8	3.8	4.8
29	0.155	0.32	0.435	0.16	0.34	0.475	0.165	0.36	0.495
30	2.4	3.4	4.4	2.65	3.65	4.65	2.9	3.9	4.9

Appendix I Emissions and CO₂ tax for CS 4

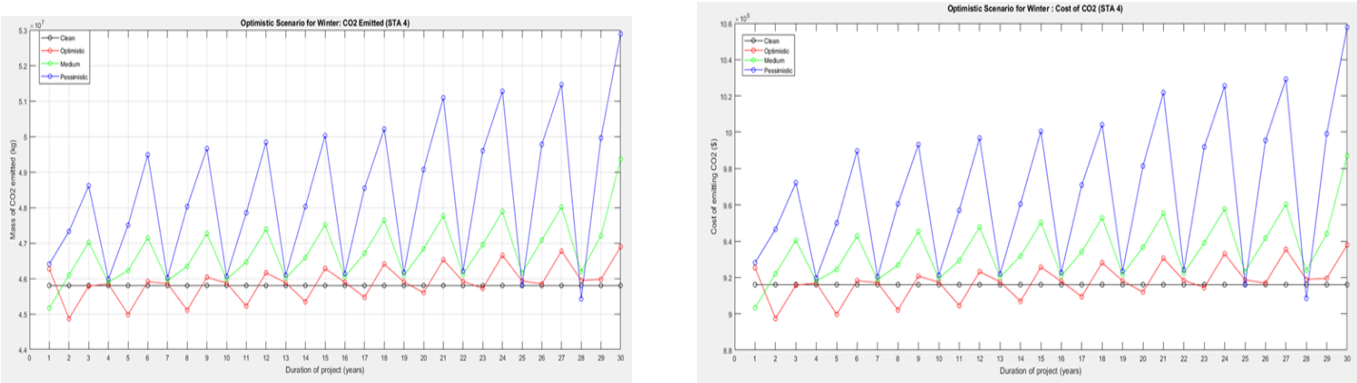


Figure I- 1 Annual CO₂ emission for CS 4 Figure I- 2 Annual tax emission for CS 4

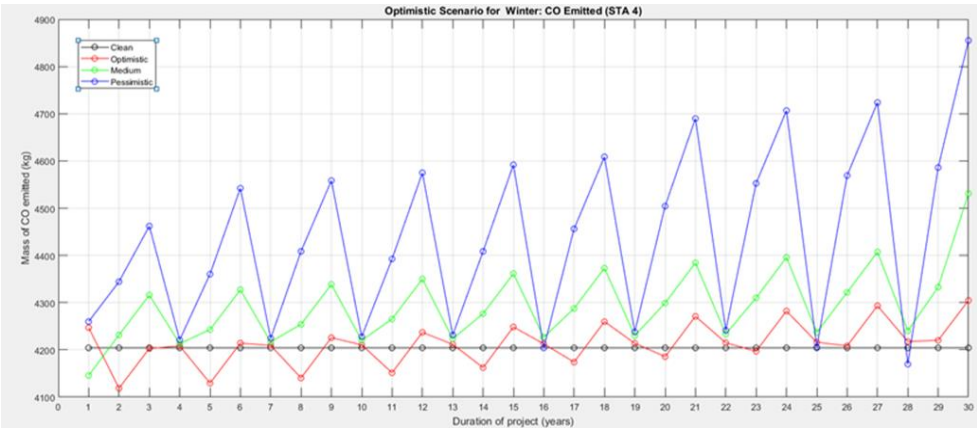


Figure I- 3 Annual mass of CO emission for CS 4

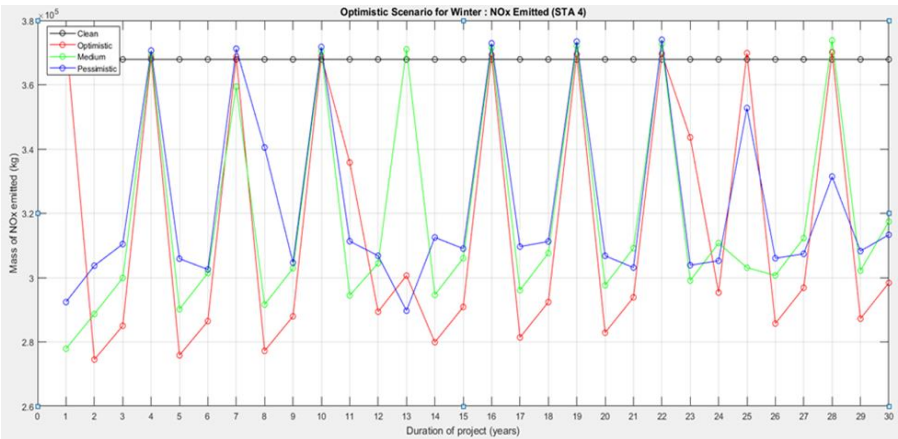


Figure I- 4 Annual NOx emission for CS 4