CRANFIELD UNIVERSITY

Abdussalam El-Suleiman

Gas Turbine Application to CO_2 Pipeline: A Techno-Economic and Environmental Risk Analysis

School of Aerospace, Transport and Manufacturing
Centre for Propulsion Engineering

PhD
Academic Year: 2013 - 2014

Supervisors: Prof. P. Pilidis / Dr. G. Di Lorenzo
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ABSTRACT

Gas Turbines (GTs) are used extensively in pipelines to compress gas at suitable points. The primary objective of this study is to look at CO_2 return pipelines and the close coupling of the compression system with advanced prime mover cycles.

Adopting a techno-economic and environmental risk analysis (TERA) frame work, this study conducts the modelling and evaluation of CO_2 compression power requirements for gas turbine driven equipment (pump and compressor). The author developed and validated subroutines to implement variable stators in an in-house GT simulation code known as Variflow in order to enhance the off-design performance simulation of the code. This modification was achieved by altering the existing compressor maps and main program algorithm of the code. Economic model based on the net present value (NPV) method, CO_2 compressibility factor model based on the Peng-Robinson equation of state and pipeline hydraulic analysis model based on fundamental gas flow equation were also developed to facilitate the TERA of selected GT mechanical drives in two case scenarios. These case scenarios were specifically built around Turbomatch simulated GT design and off-design performance which ensure that the CO_2 is introduced into the pipeline at the supercritical pressure as well as sustain the CO_2 pressure above a minimum designated pressure during transmission along an adapted real life pipeline profile.

The required compression duty for the maximum and minimum CO_2 throughput as well as the operation site ambient condition, guided the selection of two GTs of 33.9 MW and 9.4 MW capacities. At the site ambient condition, the off design simulations of these GTs give an output of 25.9 MW and 7.6 MW respectively. Given the assumed economic parameters over a plant life of 25 years, the NPV for deploying the 33.9 MW GT is about £13.9M while that of the 9.4 MW GT is about £1.2M. The corresponding payback periods (PBPs) were 3 and 7 years respectively. Thus, a good return on investment is achieved within reasonable risk. The sensitivity analysis results show a NPV of about £19.1M - £24.3M and about £3.1M - £4.9M for the 33.9 MW and 9.4 MW GTs respectively over a 25 - 50% fuel cost reduction. Their PBPs were 3 - 2 years and 5 - 4 years respectively. In addition, as the CO_2 throughput drops, the risk becomes higher with less return on investment. In fact, when the CO_2 throughput drops to a certain level, the investment becomes highly unattractive and unable to payback itself within the assumed 25 years plant life. The hydraulic analysis results for three different pipe sizes of 24, 14 and 12¾ inch diameters show an increase in pressure drop with increase in CO_2 throughput and a decrease in pressure drop with increase in pipe size for a given throughput. Owing to the effect of elevation difference, the 511 km long pipeline profile gives rise to an equivalent length of

511.52 km. Similarly, given the pipeline inlet pressure of 15 MPa and other assumed pipeline data, the 3.70 MTPY (0.27 mmscfd) maximum average CO_2 throughput considered in the 12¾ inch diameter pipeline results in a delivery pressure of about 15.06 MPa. Under this condition, points of pressure spikes above the pipeline maximum operating allowable pressure (15.3 MPa) were obtained along the profile. Lowering the pipeline operating pressure to 10.5 MPa gives a delivery pressure of about 10.45 MPa within safe pressure limits. At this 10.5 MPa, over a flat pipeline profile of same length, the delivery pressure is about 10.4 MPa. Thus, given the operating conditions for the dense phase CO_2 pipeline transmission and the limit of this study, it is very unlikely that a booster station will be required. So also, compressing the CO_2 to 15 MPa may no longer be necessary; which eliminates the need of combining a compressor and pump for the initial pressure boost in order to save power. This is because, irrespective of the saving in energy, the increase in capital cost associated with obtaining a pump and suitable driver far outweighs the extra expense incurred in acquiring a rated GT mechanical drive to meet the compression duty.

Keywords:

Gas Turbine; CO_2; Pipeline; Throughput; Power Generation, TERA; NPV

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Ozuveva, Taru, and *Isozi* respectively by their granny, I appreciate you all for your love and endurance.

Thank you all and remain blessed.

DEDICATION

This research work is dedicated to the less privileged in the society

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NOMENCLATURE

ΔH	change in head
ΔP	pressure drop
ΔP_{ic}	pressure drop in intercooler
ΔT	change in temperature
ΔZ	change in elevation
°C	degrees Celsius
°F	degrees Fahrenheit
Ar	argon
API	American petroleum institute
ARR	accounting rate of return
ASU	air separation unit
AZEP	advanced zero emission power plant
bar	unit of pressure
BEP	best efficiency point
bhp	brake horse power
CCGT	combined cycle gas turbine
CCGT-PP	combined cycle gas turbine power plant
CCS	carbon capture and sequestration
CDM	clean development mechanism
CHP	combine heat and power
COAL-PP	coal power generation plant
CH_4	methane
CO	carbon monoxide
CO_2	carbon dioxide
C_p	specific heat at constant pressure
C_v	specific heat at constant volume
D	pipe diameter
DEV	deviation
DP	design point
DLE	dry low emission

EFF	efficiency
EGT	exhaust gas temperature
EHM	engine health monitoring
EIA	energy information administration
EOR	enhanced oil recovery
EPRI	electric power research institute
ft	feet
G	acceleration due to gravity
GE	general electric
GHG	greenhouse gas
GT	gas turbine
GT-PP	gas turbine power generation plant
g/kg	gram per kilogram
HP	horse power
HRSG	heat recovery steam generator
HF	hydrogen fluoride
HCl	hydrochloric acid
H_{actual}	actual head
H_{ad}	adiabatic head
H_d	head at discharge
$H_{elevation}$	head loss due to elevation
H_f	head loss due to friction
H_{poly}	polytropic head
H_s	head at suction
H_{total}	total head loss
H_2	hydrogen
H_2O	water
H_2S	hydrogen sulphide
IRR	internal rate of return
ISA	international standard atmospheres
ISO	international organisation for standardisation
K	kelvin

Km	kilometre
$kgCO_2$/kwh	kilogram carbon dioxide per kilowatt hour
Kg/m^3	kilogram per cubic metre
kg/sec; kg/s	kilogram per second
kg/min	kilogram per minute
KJ/kg	kilo joules per kilogram
KJ/kwhr	kilo joules per kilowatt hour
KPa	kilo Pascal
KW	kilo watt
KWh	kilowatt- hour
L	length of pipe
LAER	lowest available emission rate
LHV	lower heating value
M	million
MJ/kg	mega joules per kilogram
Max	maximum
m	mass flow rate, metre
mm	millimetre
mmscfd	million standard cubic feet per day
m^3/kg	cubic metre per kilogram
m^3/sec	cubic metre per second
m^3/min	cubic metre per minute
m^3/hr	cubic metre per hour
MPa	mega Pascal
MEA	mono-ethanolamine
MTPY	million tonnes per year
MTBF	mean time between failure
MTTR	mean time to repair
MW	mega watt
MWh	megawatt hour
mw	molecular weight
N	number of stages; compressor speed

NPSH	net positive suction head
N_2	nitrogen
n	polytropic factor
Nm	newton metre
Nm/kg	newton metre / per kilogram
Nm/sec	newton metre / per second
NPV	net present value
NO_x	oxides of nitrogen
O_2	oxygen
O & M	operation and maintenance
OEM	original equipment manufacturer
ONG-PP	oil and natural gas dual fired power plant
OPEX	operating expenditure
P	power; pressure
POLY	polytropic
PR	pressure ratio
psia	pounds per square inch atmosphere
psig	pounds per square inch gauge
P_{ave}	average pressure
PBP	payback period
P_d	discharge pressure
P_{poly}	polytropic power
PR-EOS	Peng –Robinson equation of state
P_s	suction pressure
P_{shaft}	shaft power
Q	volume flow rate, actual flow
R	specific gas constant
R & D	research and development
Re	Reynolds number
r	compression ratio
r_i	stage compression ratio
R_o	universal gas constant

SCADA	supervisory control and data acquisition
SCMD	standard cubic metre per day
SFADW	corrected non dimensional mass flow scale factor
SFEFF	efficiency scale factor
SFPR	pressure ratio scale factor
SOAP	spectrometric oil analysis program
SO_x	oxides of sulphur
SO_2	sulphur dioxide
T	temperature
TERA	techno-economic and environmental risk analysis
TET	turbine entry temperature
TR	temperature ratio
ton	tonne
$T_{(in-pump)}$	pump inlet temperature
$T_{(out-comp)}$	discharge temperature of compressor
T_d	discharge temperature
T_s	suction temperature
UHC	unburnt hydrocarbon
UK	United Kingdom
US	United State
USA	United States of America
V	specific volume; velocity
VIGV	variable inlet guide vane
VS	variable stators
VSV	variable stator vanes
W	mass flow
£ 2013M	year 2013 pounds sterling value in million
W_{pump}	pump power
W_{ref}	refrigeration power
Z	compressibility factor
ZEPP	zero emission oxy-fuel power plant
Z_{ave}	average compressibility factor

Z_s	suction compressibility factor
%	percentage
£	British pound sterling
£/kW	pounds per kilowatt
£/kg	pounds per kilogram
£/kW-year	pounds per kilowatt year
£/kWh	pounds per kilowatt hour
£/MWh	pounds per megawatt hour
£/ton CO_2	pounds per tonne of carbon dioxide
γ, ϵ	ratio of specific heats
Π	pi
$\boldsymbol{\mu}$	dynamic viscosity
Υ	specific gravity
$\boldsymbol{\delta}$	corrected pressure
$\boldsymbol{\theta}$	corrected temperature
η_{mech}	mechanical efficiency
η_{poly}	polytropic efficiency
η_{pump}	pump efficiency
η_{DP}	efficiency at design point
$\eta_{DP\ Map}$	efficiency at design point of map
ρ	density
″	inch

1 General Introduction

1.1 Research Background

Gas turbines are extensively used in pipelines to compress gas at suitable points. The objective of this study is to look at carbon dioxide (CO_2) return pipelines and the close coupling of the compression system with advanced prime mover cycles. The investigation involves a comparative assessment of traditional and novel prime mover options including the design and off-design performance of the engine as well as the economic analysis of the system. The originality of the work lies in the technical and economic optimisation of gas turbines based on current and novel cycles for a novel pipeline application in a wide range of operating conditions.

1.2 Environmental Concern and the CCS Technology

There are environmental concerns that the release of greenhouse gases (GHG) into the atmosphere is attributed to the global warming manifestation in the earth's climatic condition. The main anthropogenic GHG is carbon dioxide (CO_2); and CO_2 is produced through combined action of widespread fossil fuel combustion, deforestation and range of industrial processes. However, power generation plants burning fossil-fuel has been identified as the major source of anthropogenic CO_2 [1-3].

In the light of the above, there have been policy shift to generate power by environmentally friendly means. Sequel, much research and development in the use of so-called renewable energy had led to huge investment in wind turbines, bio-fuels, solar energy and even Nuclear power generation plants. Despite the environmental concerns, it is almost certain that in many years to come, there will be no substitute for fossil-fuel fired power generation plants due to obvious realities surrounding these renewables when cost and plant availability comes to play. In fact, the current attention to non- conventional sources of fossil fuel like oil sand and oil shale attest to this; so also is the current desire to use coal / coal-derived fuels for power generation in the so-called "clean coal" technologies.

For sustainability, it becomes imperative that power generation from fossil- fuel fired power plants implement strategies to mitigate the release of CO_2 into the atmosphere. It is therefore envisioned, that in the near future new power generation plants will be carbon capture ready while existing ones are retrofitted with carbon capture systems.

This vision currently pursued under the carbon capture and sequestration (CCS) technology [4; 5], essentially involves capturing the CO_2 at source, then transporting and storing it in depleted reservoirs, saline aquifers, or its use for enhanced oil recovery (EOR) in depleted oil and gas fields [1]. CCS is expected to provide 20% reduction of CO_2 in the low carbon power generation initiatives in order to meet the target of stabilising CO_2 concentration at 450 parts per million by volume in the atmosphere [6]. Figure 1-1 below shows the various means of capturing CO_2 emission from power generation plants as well as process plants.

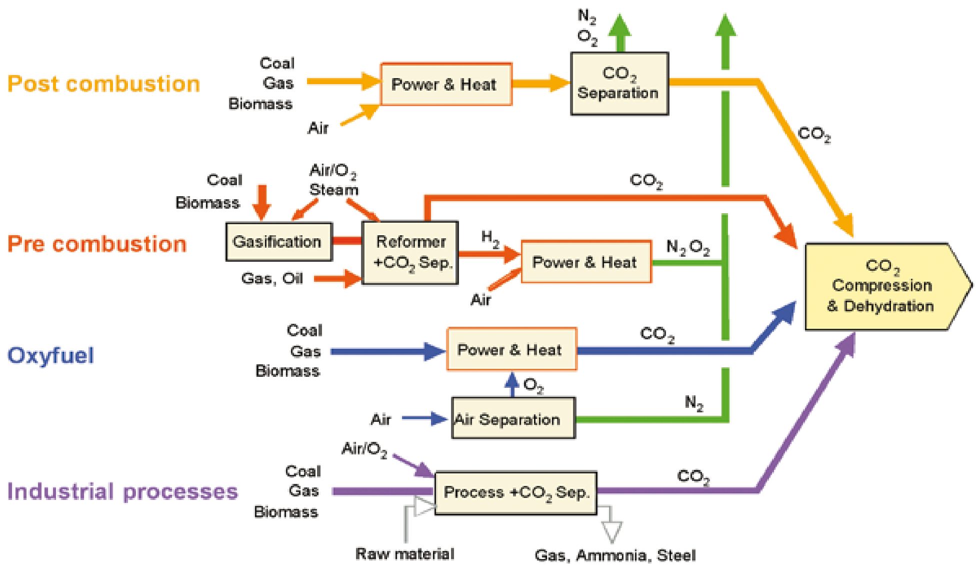

Figure 1-1 CO_2 Capture Systems

Source: [1]

Large quantities of CO_2 will be produced; and unfortunately, most sources of CO_2 emission are usually not located near the point of storage. Therefore, transportation over long distance is envisaged. The use of vessels, tanks or barges as a means of transportation in this situation is deemed uneconomical.

For long distance and large volumes, pipeline has been identified as a primary means of transportation and current practice [7-10]. The requirement to compress the CO_2 to pipeline operating pressure as well as sustain the pressure of the CO_2 along the pipeline into the sink (point of storage) becomes an area of concern.

The compression duty per se will demand a viable prime mover such as the gas turbine (GT) that has proven very successful in pipeline application. This novel application of GT will necessarily demand techno-economic analysis. Such analyses coupled with risk assessment will provide basis for a well informed decision prior to investment which this study seek to address.

1.3 The TERA Philosophy

TERA is an acronym for Techno-Economic and Environmental Risk Analysis, a concept developed in Cranfield University, UK for holistic assessment of power plants with the sole aim of making an educative judgement by comparing and contrasting competing schemes before embarking on investment [11; 12]. It was originally developed for the aviation propulsion system; however, over the years it has been extended to industrial GT applications in power generation, marine transport and the oil and gas sector [13-16]. It has also been employed in asset management by evaluating case scenarios in order to assess the trade-offs involved. Hence, TERA has been described by Raja etal [13] as “a multidisciplinary tool for modelling of GT and engine asset management”.

TERA enable reasonable quantification of risk and it provides a formal and consistent platform to compare and rank novel and existing power plants.

This becomes particularly important in areas where there are no prior experiences to hold unto for decision-making. Thus, scarce resources can be utilised in the most efficient and profitable manner.

The core of the TERA is the performance module which is a detailed and rigorous thermodynamic representation of the components parameters and power plant by simulating the design, off-design and degraded performance of the power plant. The economic, environmental and risk modules are built around the performance module and integrated with an optimiser to compare power plants optimised for a particular duty such as fuel consumption and operating cost.

1.4 Aim and Objectives

The aim of this research is to investigate the application of GT as prime mover for transmission of CO_2 in pipeline over long distance. This aim was achieved through the following set objectives:

- Simulation of design and off-design performance of GT mechanical drive for pipeline application
- Estimating CO_2 pump and compressor power requirement
- Modification of an in–house GT simulation code by developing subroutines to implement variable stators in the GT compressor
- Development of economic and pipeline hydraulic models and the adapting of existing emission model to perform techno - economic and environmental risk analysis

1.5 Contribution

This research has made the following contribution to knowledge:

i. Development of computer algorithm to implement variable stators in an existing in-house GT simulation code (Variflow).
ii. Development of a flexible economic simulation code based on NPV method for GT application assessment.
iii. Development of a robust pipeline hydraulic model using fundamental flow equation to analyse the CO_2 pressure distribution along a real life pipeline profile.
iv. Using the TERA approach to provide an economic evaluation of employing GT mechanical drive for CO_2 compression.

1.6 Thesis Structure

This thesis consists of seven (7) chapters beginning with a general introduction and a conclusion. Each chapter begins with a background information or brief introduction. It is structured in such a way that every chapter build upon the previous one. Apart from the first and last chapter, concluding remarks were presented at the end of the chapters. Chapters 4 – 6 contain the contributions by the author.

Chapter 1 presents a general introduction to the study by highlighting the background of the study; discussion of environmental concern about CO_2 and technology of its capture and storage away from the atmosphere; the philosophy of techno-economic and environmental risk analysis (TERA); the aim and objective of the study and finally the perceived contribution to knowledge.

Chapter 2 presents the review of necessary literature upon which the study revolves around and the identification of the gap in knowledge which this study attempts to fill. The main areas of review are on GT mechanical drive performance; performance characteristics of GT driven equipment i.e. compressor and pumps; pipeline transmission of CO_2 and the technicalities involved; brief insight into natural gas pipeline transport and a comparison between natural gas and CO_2 pipeline transport. Other areas reviewed in this chapter are GT cost and economic performance evaluation methods.

Chapter 3 presents the methodology of the study which is based on the TERA frame work established in Cranfield University. The real-life pipeline profile adapted for this study is highlighted as well as an overview of the Turbomatch scheme.

Chapter 4 presents an EXCEL based computer model to evaluate the CO_2 compression power requirement using multi-stage centrifugal compressors; results analysis and the effect of intercooler pressure drop on the compression power; simulation of CO_2 compressibility factor based on the Peng-Robinson equation of state and its validation.

Chapter 5 presents the modification of an in-house GT simulation code known as “Variflow” which has the capability of using other working fluid. The implementation of variable stators for performance enhancement of single-shaft

GT is discussed; the associated subroutine development in FORTRAN; compressor map modification and the validation of the code.

Chapter 6 presents the techno-economic and environmental risk analysis of deploying GT in a CO_2 pipeline. It begins with the description of the case scenarios, then the design and off-design performance analyses of selected GT mechanical drives; economic and pipeline hydraulic analysis modelling in FOTRAN; pipeline flow simulation; economic and risk performance simulation through case scenario studies; and a brief summary of the case scenario findings.

Chapter 7 is the final chapter. It presents conclusions drawn from findings and results obtained in the entire research work. Recommendations for further work is highlighted.

2 Literature Review

2.1 Introduction

The review of relevant documents is presented here. An overview of gas turbine (GT), its performance and application to pipeline is conducted. The review of GT driven equipment (compressors and pump) and technical requirements for CO_2 compression for pipeline transport is carried out. This chapter equally highlights the gap in knowledge which this study seeks to fill.

2.2 The Gas Turbine

GTs also known as air-breathing engines work on the basic thermodynamic principle of the Brayton Cycle. The GT (Figure 2-1 below) is made up of three main interconnected components namely: compressor, combustion chamber or combustor and turbine [17]. Atmospheric air is basically compressed in the compressor; delivered to the combustor in a regulated manner where it mixes with fuel and ignited; and the combustion product is expanded in the turbine to extract power. The components are connected via shaft(s) which in modest configuration is referred to as a simple cycle.

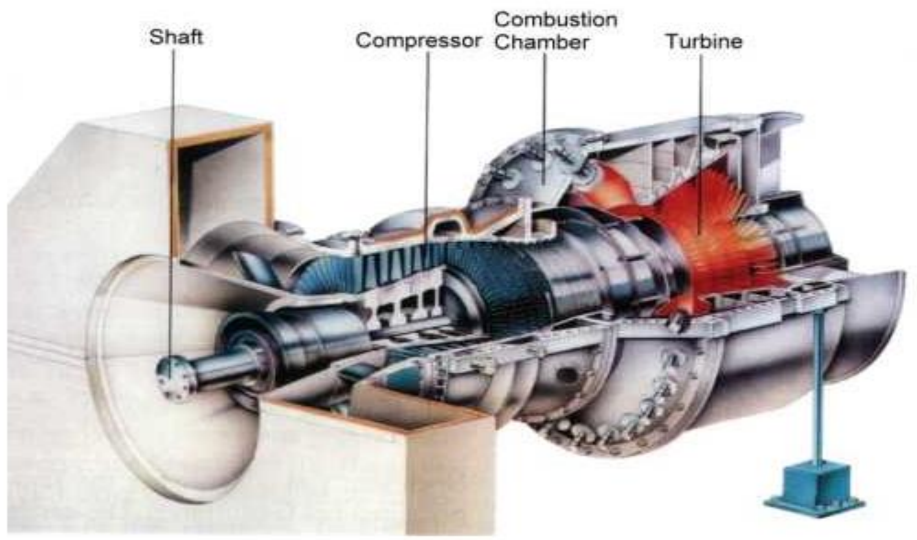

Figure 2-1 Pictorial View of a Cut-Away GT Engine (Courtesy GE)

GT simple cycle could either be single shaft or multi-shaft depending on the operation requirement. When an operation requires flexibility like in marine or pipeline application where there is changing power requirement, multi-shaft arrangement becomes necessary otherwise a single shaft is ideal. Part-load performance can also be enhanced in GT cycles by the use of bleed valves and variable inlet guide vanes (VIGV).

Major manufacturers of GTs otherwise called the OEMs (Original Equipment Manufacturers) among others include Rolls Royce, General Electric, Siemens, Pratt and Whitney, Westinghouse and Alstom.

2.2.1 Open and Closed Cycle Gas Turbine

Most GT cycles in current use are the open cycle type in the sense that energy addition takes place by adding fuel directly to the working fluid in a combustion chamber. Upon expansion in the turbine, the combustion product is released to the surrounding. Another kind of GT cycle is the closed cycle, where the working fluid is repeatedly circulated in the system instead of discharging it into the atmosphere. Energy is transferred to the working fluid via heat exchanger which implies both the working fluid and the turbine component have no direct contact with the source of heat (i.e. the combustion products). The closed cycle GT is rarely used but still being developed with lots of advantages been advanced for it [17]. When used, it is of the regenerative type usually incorporating intercoolers for effectiveness of the regenerator. The intercooler reduces the temperature of the working fluid at entrance to the compressor, thereby reducing compressor outlet temperature for improved temperature gradient across the regenerator. Owing to the heat exchange surface involved, the turbine entry temperature in closed cycle GT is always limited to about 1100K [18].

2.2.2 Advanced Cycle Gas Turbine

Early GTs were characterised by low turbine inlet temperatures, low power output and poor thermal efficiency. These were tied to the level of technology available and mechanical integrity issues. However, with advancement in material technology, GT inlet temperature has risen over the years from 1200 K in the 1950’s to over 1800 K today [19; 20].

The power output of a GT is affected by component inefficiencies, compressor pressure ratio and turbine inlet temperature. In an effort to improve the performance of GT, the simple cycle GTs have been modified by introduction of other components and are known as advanced GT cycles [17; 21; 22]. GT cycles have been incorporated with systems like recuperator, intercooler, reheat and steam injection or combination thereof to improve the cycle efficiency and its overall performance.

Other modifications include inlet air cooling; and the addition of a steam turbine at the hot end of a GT cycle to form a combined cycle configuration. In this configuration, the GT is referred to as the topping cycle while the steam turbine is referred to as the bottoming cycle. The bottoming cycle essentially uses the topping cycle's exhaust, typically at a temperature of 500°C-600°C to convert water to steam in a heat recovery steam generator (HRSG) which is used to drive the turbine for power generation [17]. The combined cycle GT (CCGT) has brought tremendous improvement to both power output and cycle efficiency. The CCGT is widely replacing older power generating plants because of the added advantage of comparative environmental friendliness in its efficient utilisation of the exhaust gas temperature from the topping cycle. It equally gives a significant reduction in specific fuel consumption and improvement in the overall cycle efficiency [23; 24].

Simple cycle GTs with thermal efficiencies in the order of 40% and in combined cycle configuration in order of 60% are currently achieved [25-27]. Similarly, the latest fleets of GT offer features such as advanced combustion systems, multi-fuel capabilities and reduced maintenance [28]..

2.2.3 Novel Gas Turbine Cycles for Oxy – Fuel Power Generation

A modified GT cycle which substantially eliminates emission of NOx or other pollutants, and which efficiently collects CO_2 for beneficial use or elimination is explored here. **Oxy-fuel combustion** involves the use of nearly pure oxygen (O_2) for combustion instead of air. The near pure oxygen (95 – 99% purity) is produced in a low temperature (cryogenic) air separation unit (ASU); or membranes in the near future [29]. It essentially removes nitrogen from the combustion process, there by producing flue gas that is mainly CO_2 and H_2O. The flue gas is cooled down to condense the water vapour and subsequently dried and purified for pipeline delivery to storage. Oxy-fuel combustion capture gives nearly 100% efficient CO_2 capture with the net flue gas containing 80-98% CO_2 by volume depending on the fuel used and the oxy-fuel combustion process [1].

The modification of the GT cycle for adaptation to this process has led to novel GT cycles that are currently being investigated [1; 18; 18; 26; 29-33]. The Matiant and Graz cycles configurations are known forms of these novel cycles. The former employs features like intercooling between compressors, reheat and recuperation while the latter incorporates a heat recovery steam generator to add a bottoming steam cycle to the GT cycle. Other modification exists like replacing the GT combustor with a reactor system integrating the ASU, the recuperating unit and the combustion unit. They are known by several names such as Semi-closed Cycle GT [33]; zero emission oxy-fuel power plant (ZEPP) [29] and advanced zero emission power plant (AZEP)[1; 31].

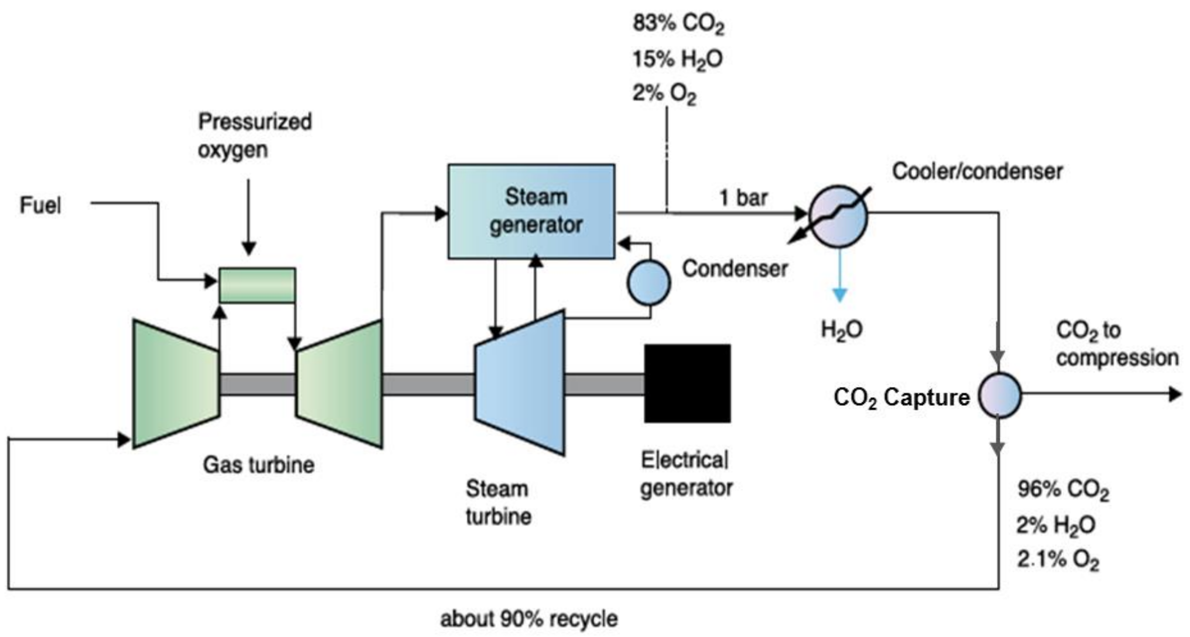

Figure 2-2 Principle of Oxy-Fuel Gas Turbine Combined Cycle

Source: [8]

The working fluid is basically CO_2 with some degree of impurity depending on the purity of the fuel and O_2 used for combustion [29; 31]. The Ulizar's semi-closed cycle GT for instance contains argon (Ar) as an impurity [18].

A typical configuration of the oxy-fuel GT in a combined cycle mode is shown in Figure 2-2 above. The working principle involves the burning of the coal gas fuel with high pressure O_2 and recycled flue gas in the combustion chamber. The burning of fuel in pure O_2 results in a very high temperature, thus the flue gas or combustion product (CO_2 and H_2O) is usually recycled to control this temperature. Upon expansion in the GT to extract power, the flue gas at a temperature of about 600°C is used in a steam generator to convert water into steam and further power extracted by expansion in a steam turbine. The exhaust flue gas passes through the condenser and cooler to remove H_2O and cool down the flue gas which is majorly CO_2 and traces of O_2, H_2O and any other fuel bound impurity. Part of the CO_2 is recycled while a major percentage is captured, purified and compressed to required pressure for sequestration.

The performance of these cycles is said to yield efficiencies in the range of 40-47% depending on the technology and over 50% for the combined cycle configuration. However, these efficiencies are highly penalised by parasitic loads brought about by the requirement to separate O_2 from air and the capture and compression of CO_2 to desired pressure [29; 31].

2.3 Overview of Gas Turbine Application

The initial development of GT was for electric power application but was unsuccessful due to far better performance of existing steam turbines and diesel engine. However, its application in military jet engine at the end of World War II became its first major contribution [17; 34]. GTs can be broadly divided into two types based on application, namely, aero GT engines and industrial GT engines. Aero engines are used for aircraft propulsion; they are quite compact; have high power to weight ratio; operate at high turbine inlet temperature and high pressure ratio ~ 35. They are designed to be very efficient and reliable and depending on either civil or military application, component life may or may not be a requirement.

The industrial GT engines are further grouped into two: the aero-derivative GT and the heavy duty GT [35]. They are generally designed based on the requirement for power, availability, reliability and maintainability (extended time between overhauls and ease of maintenance [36]. The aero-derivative GTs were designed based on aero- application requirements; hence, they retain similar features mentioned above. They are quick to load; majorly of the civil aircraft application type thereby having medium component life and good fuel economy; have thermal efficiency up to 42% [17; 37].

The heavy duty GTs have relative low power to weight ratio since they were not initially designed to fly and quite bulky. They are characterised by slower loading depending on size; have moderate pressure ratio of 10–18; and have thermal efficiency of about 30-38%. They are generally designed for long life of components. In the early 1960s, there exist significant technological gap between the aero-derivative and the heavy duty GTs but in the last 30yrs, this gap has been reduced considerably. Nowadays, they share same technological development there by closing in the gap [17; 38]

Current applications of GT are as follows:

i. Electrical power generation where they are used for base load, peaking load and emergency load applications.
ii. Combine heat and power (CHP) or cogeneration
iii. Marine propulsion primarily used in combination with either a diesel engine or steam turbine engine.
iv. Pipeline transmission.

The sizes of GTs are quantified based on application. When used for power generation, they are measured in Kilowatt (KW) or Megawatt (MW) and when used as mechanical drives, they are measured in Horsepower (HP).

2.4 The Gas Turbine Prime Mover

Essentially, prime movers are used to drive a pump or compressor in a pipeline network. The GT is one out of four primary types of prime movers used in pipeline application. Others are steam turbine, reciprocating engine, and electric motor drive [39-43].

The prime mover key requirements for mechanical drive in order of importance are [44]:

i. Low weight, for easy transportation to remote location
ii. Good base load thermal efficiency due to near 100% utilisation
iii. High availability and reliability
iv. Reasonable part power torque to allow for varying compressor/pump duty.

GTs are widely used as drivers on gas transmission system especially as they are generally most appropriate for driving centrifugal compressors now commonly used in pipeline gas compression [40]. They match the operating range of the compressors which is usually between 50 – 105% of the compressor rated speed or even lower [39; 45]. GTs for pipeline application requires flexibility; hence they are normally designed with a mechanically independent turbine otherwise known as a free power turbine. This result in a twin shaft configuration and sometime referred to as split-shaft mechanical drive GT. These GTs are in two parts, namely the gas generator and the free power turbine (see Figure 2-3 below).

The combination of the compressor module, combustor module, and turbine module is known as the gas generator. The free power turbine is aerodynamically coupled to the gas generator while the driven equipment is mechanically (directly or indirectly via gear-box) coupled to the free power turbine [34; 39]. This configuration allows the gas generator to run at different speed with the power turbine. Hence the power turbine can be designed to run at typical centrifugal compressor and pump operating speeds [37].

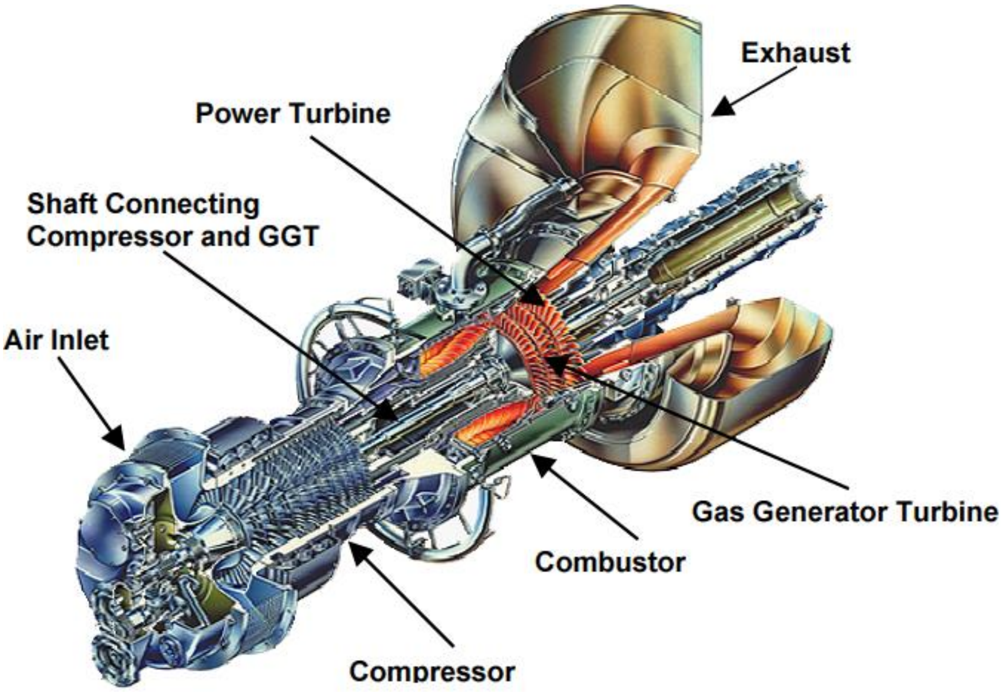

Figure 2-3 Twin Shaft Gas Turbine

Source: [46]

In the aero-derivatives, the nozzle of either a turbojet or turbofan engine is replaced by a free power turbine. The modular arrangement, high power to weight ratio and compact size of these GTs are features that permit easy transportation and maintenance. This becomes very important noting the fact that most compressor stations are in remote areas.

The aero-derivatives are more fuel efficient compared to the heavy frame industrial GTs because of their stringent design features. Compared to steam turbines they are even better. Similar comparison between an all motor drive and GT drive favours the latter from the perspective of losses due to electrical generation and transmission.

GTs for pipeline application are manufactured in the power brackets of 6 - 10 MW, 15 MW and 25 - 30 MW [44]. In these power brackets, the high power to weight ratio of GTs gives them an edge over the high speed diesel engines and steam turbines. GTs with low pressure ratio and turbine inlet temperature leading to thermal efficiencies ~33% are custom designed for the lower power bracket. Industrial engines built by Solar, Siemens and Nuovo pignone as well as aero-derivatives like Allison 501 are common examples. Simple cycle, free power turbine aero-derivatives are usually used for the mid and high power brackets. Typical efficiencies of these turbines are between 35-39% with turbine inlet temperatures ~1450 – 1550 K across a pressure ratio of 20:1 to 25:1. The Rolls-Royce industrial RB211 and GE LM2500 are common GTs used in this power divide [17].

The power output of GTs could adversely be affected by changing environmental conditions especially ambient temperature. GTs are also disadvantaged by the fact that they burn expensive and clean fuel. Typical consumption of GTs operating on natural gas pipeline is about 7-10% of the pipeline throughput for compression purposes [17; 44]. However, given the tremendous improvement in GT cycle efficiency, widespread availability of gaseous fuel and current price level, the GT will remain a favoured prime mover in this application.

2.5 Gas Turbine Availability and Reliability

Availability is the amount of time equipment operates relative to the amount of time it is required to operate. In order words, it is the ratio between the hours per year that the equipment actually operates to the hours per year the equipment is supposed to operate [47; 48]. Thus availability takes into account the total equipment down time which could be due to a planned or unplanned maintenance. Reliability on the other hand is the ability of the equipment to perform the desired function without any forced outage or unplanned maintenance. Reliability depends on the GT design, manufacturing process, operating environment and quality control.

Availability is a function of both mean time between failure (MTBF) defined by reliability and down time (or mean time to repair - MTTR) defined by schedule maintenance required for efficient operation and unplanned maintenance. Therefore, availability (A) may be expressed mathematically as [49]:

$$\boldsymbol{A} = \frac{\boldsymbol{MTBF}}{\boldsymbol{MTBF+MTTR}} \qquad \textbf{[Eq. 2-1]}$$

When forced outages are experienced, the associated down time is hinged on the length of time taken to complete the repairs or maintenance action. In GT engine, forced outages often emanate from engine support systems such as control and fuel systems. The systems' down time are efficiently managed through design of redundancy and holding of appropriate spares. Similarly, sophistication in instrumentation and the use of microprocessor based controllers today has further enhanced the availability of these engine support systems.

The forced outage caused by the major components that make up the GT engine e.g. the compressor and turbine leads to a very long down time. This is partly due to the fact that they are rarely held in spares because of cost consideration and the continuous improvement in their design. However, GTs have high reliability and their availability has further being enhanced through modular design. Another enhancement of availability is through the implementation of engine health monitoring (EHM). EHM employs methods of vibration monitoring, oil analysis (e.g. SOAP or ferrography), visual inspection and performance monitoring to detect time-dependent failures in GT with a view of correcting such before causing a forced outage [50].

Availability affects the economic performance of GT in that revenue is lost because of the inability of the GT to provide the full duty required in a given application. The loss of income in this particular application will be the reduction of pipeline throughput or the cost of using alternative power supply within the period of carrying out repairs or maintenance.

2.6 Pipeline Transmission of CO_2

The economics for transporting large volumes of CO_2 over long distance in the gaseous phase is quite unfavourable [10; 51; 52]. This is because such pure gas phase transportation will be restricted to operate below 30 - 50 bar at which the densities and capacities will be too small. However, the pure gas phase will fit a system where the CO_2 is initially transported on gathering lines from low CO_2 producers for connection to larger trunk pipelines [10].

Low pressure liquid phase is not desirable either except for transportation in vessels, tanks and barges. The liquid phase for pipeline transportation comes with problems of frost formation in cold climate and the need for pipeline material suitable for cryogenic application. Furthermore, unlike natural gas, the critical point of CO_2 is near the potential compressor (and pipeline) operating point such that slight changes in ambient temperature or soil temperature for a buried pipeline can result in two-phase flow [51].

Abrupt pressure drops occasioned by two phase flows and the detrimental effect of impurities must be avoided during CO_2 transmission in pipeline. Hence, CO_2 is said to be most efficiently transported in a dense-phase state especially for onshore pipelines [7; 51; 53; 54]. The dense phase transport gives an advantage for high delivery pressure at injection to sink [1].

In general, the CO_2 is transported at temperature and pressure ranges of 12°C - 44°C and 85 – 150 bar respectively [7; 39]. The upper temperature limit is fixed by the compressor-station discharge temperature while the lower limit is fixed by winter ground temperature for buried pipelines. The lower pressure limit is dictated by the phase behaviour of CO_2 especially with impurities present and the need to maintain supercritical conditions while the upper pressure limit is mostly due to economic concerns [7].

Highly recommended within this pipeline operating conditions is the “API 5L X-70” line pipe specification in the US Code of Federal Regulations - CFR (the CFR regulates the design, construction and operation of CO_2 pipeline transport in the USA) [8; 55; 55]. Similarly, the ASME-ANSI class 900# flanges are quite suitable since they can tolerate operating pressure of about 153 bar (15.3 MPa) at a temperature of about 311.15 K (38°C) [39; 51]. Further insight into CO_2 pipeline design, construction and operation are documented in references [39] and [56].

2.6.1 Physical Property of Pure CO_2

Since the captured CO_2 will be transported in pipeline in a dense phase condition, an insight into the phase behaviour of pure CO_2 (Figure 2-4 below) is worthwhile for operational exigencies. CO_2 in its pure state is colourless, odourless and non-inflammable at ambient temperature and pressure [7].

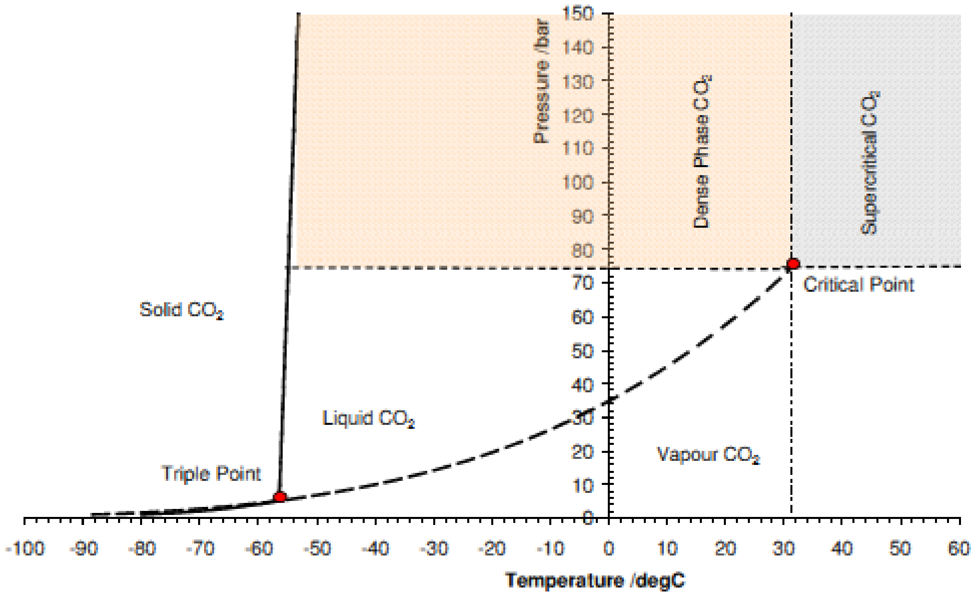

Figure 2-4 Phase Diagram for Pure CO_2

Source: [57]

Two distinct points can be observed namely the triple point (5 bar, - 56°C) and the critical point (74 bar, 31°C). At the triple point, CO_2 can exist as solid, liquid or gas. Above the critical point, the pure CO_2 is in a supercritical phase where it acts as neither gas nor liquid [54]. At pressures above the critical pressure but below the critical temperature, a fluid is said to be in a dense phase else it is non-dense phase. Note in Figure 2-4 above, that the dense phase could be achieved without passing the liquid area. The dense phase is a very peculiar state in that on one hand it is not a liquid because it has a viscosity similar to that of a gas; while on the other hand its density is closer to that of a liquid and has similar flow conditions as that of a liquid [10; 53]. This presents the opportunity to pump the CO_2 with either a pump or compressor during transmission.

2.6.2 Impurities in the Captured CO_2

The captured CO_2 due to technical and economic reasons contain impurities which have great impact on the physical properties of the CO_2 being transported [58; 59]. The consequence of this, manifest itself on the design, operation, and cost of CO_2 transport. The type of fuel used and the capture process employed dictates the kind and level of impurities involved. Li and Yan [53] in an exhaustive study categorised these impurities into two parts: impurities from air and fuels, such as oxygen (O_2), nitrogen (N_2), methane (CH_4), argon (Ar) and hydrogen sulphide (H_2S); and impurities from combustion products, such as oxides of sulphur (SO_x), water (H_2O) and oxides of nitrogen (NO_x). Other air pollutants include particulates, hydrochloric acid (HCl), hydrogen fluoride (HF), mercury, other metals and other trace organic and inorganic contaminants. H_2 is also present in pre-combustion processes as an impurity [1; 59].

The effect of impurities like H_2, O_2 and N_2 in the CO_2 is to operate at higher pressures to avoid two – phase flow in the system, hence, an increase in power requirement. In like manner, inert gases and CH_4 must be reduced to a low concentration to avoid 2-phase flow [1; 59] . Although the presence of H_2O relatively decreases the power requirement [60]; in order to prevent water condensation and corrosion in pipelines and to allow the use of conventional carbon-steel materials, the CO_2 must be dried and free of H_2S [1; 59; 61]. A field study estimated corrosion rate of 0.00025 – 0.0025 mm per year without the presence of H_2O [54]; therefore, there is risk of increased corrosion rate if the CO_2 is not dehydrated.

2.6.3 CO_2 Flow Capacity

The CO_2 flow capacities from source of emission are usually quantified in million tonnes per year which can be reduced to appropriate units during flow analysis in pipeline. Several factors determine the amount of CO_2 flow namely [10; 39; 62] :

- CO_2 density which depends on the phase condition
- Internal pipe diameter
- Distance between booster stations
- Hydraulic pressure gradient

- Temperature of the transported CO_2 (both initial temperature after compression and the ambient temperature)
- CO_2 fluid viscosity
- Internal pipe wall roughness and
- The pipeline profile or terrain

2.6.4 Temperature Range of Buried Pipeline

Pipelines transporting fluids are usually buried for environmental, security and safety reasons. The stability of underground temperature compared to surface temperatures is also a major consideration. At ambient temperature of $36.5°C$, the ground surface temperature at noon was observed to be about $65°C$ while at one meter depth underground, the temperature was found to be $30°C$ [63].

In temperate regions the soil temperature is said to vary from below zero in winter to $6 - 8°C$ in summer while in the tropics the soil temperature may reach up to $20°C$ [10; 39; 62]. Therefore, a buried pipeline for dense phase CO_2 transmission is envisaged to be advantageous despite the associated increase in pipeline investment cost.

2.6.5 CO_2 Compression

Prior to transportation in pipeline, the captured CO_2 after due cleaning (scrubbing) is compressed to pipeline operating pressure with the aid of mechanical driven equipment – compressors or pumps as the case may be. Further compression may be necessary; hence compressor or pump stations are located along the pipeline profile. The main objective here is to ensure the CO_2 is introduced into the pipeline at the right pressure and ensure the sustenance of the CO_2 pressure above a minimum designated pressure during transmission along the pipeline profile.

The amount of impurities present in the CO_2 impact on the compression work requirement. The presence of Ar, O_2 and N_2 is said to increase the compression work while impurities like SO_2 and H_2O could decrease this work [60]. The temperature and pressure ranges that give the most economical dense phase flow in pipeline are 15 - 30°C and 100 - 150 bar respectively [53]. In line with most studies, the CO_2 is compressed from an initial pressure of 1bar to pipeline intake pressure of about 150bar using a combination of centrifugal compressor and pump. Effective compression of the CO_2 is achieved with intercoolers (intercooler temperature always above condensing temperature of the CO_2 stream) while at inlet to the pipeline after-coolers are employed [10; 39; 54; 56; 63-65].

2.7 Natural Gas Pipeline and Compression Stations

For many years, natural gas has been transported in pipeline for both domestic and international consumption. Being the most convenient means, natural gas transport in pipeline is still very popular and remains a significant mechanism for gas delivery to markets [66]. The pipeline could be underground, over-ground or subsea and the pipeline is designed in such a way as to quickly and efficiently transport natural gas from its origin to area of need. The size of existing gas pipelines range from 6″ to 56″ in diameter; e.g. the West African Gas Pipeline main line is 20″ while the Trinidad Cross Island Pipeline is 56″. Depending on the purpose, natural gas pipelines are classified in three categories as:

i. Gathering pipelines- which are short (couple of hundred meters) and small diameter pipelines used for conveying natural gas from nearby wells to a treatment plant or processing facility.
ii. Transportation Pipelines- these category of pipelines are usually long distance pipes with large diameters conveying gas between cities, countries and even across continents. The long distance associated with this pipeline makes the network to incorporate compressor station along the gas line.
iii. Distribution Pipelines- these are essentially series of interconnected small diameter pipelines used for delivering the gas to the consumer. These are usually found in terminals where the gas products are distributed to tanks and storage facilities for onward distribution to homes and industrial users through feeder lines.

2.7.1 Elements of a Pipeline System

The main elements of a pipeline system are shown in Figure 2-5 below. It comprises pieces of equipment arranged in such a way as to facilitate the safe movement of gas from one point to another. The functions of these elements are highlighted below:

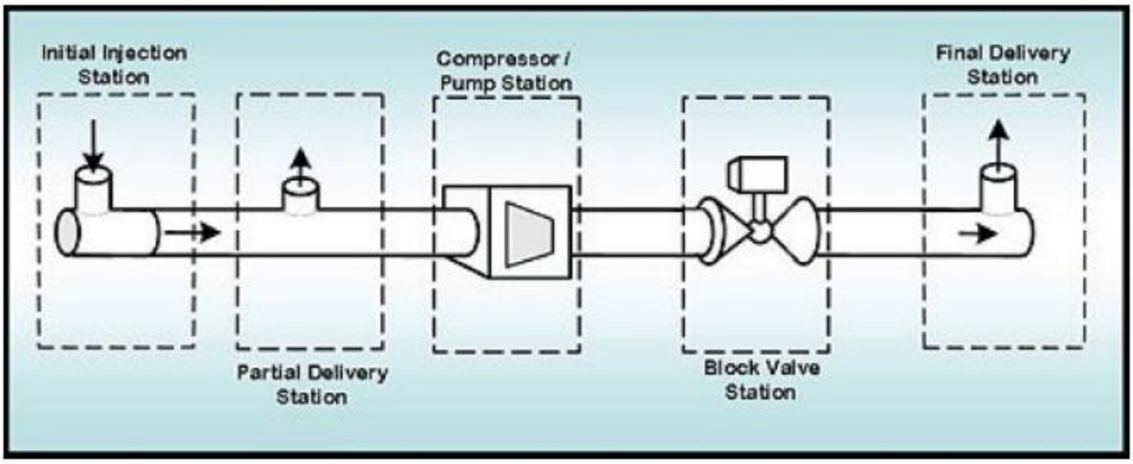

Figure 2-5 Pipeline Transportation Schematic

(Adapted from [66]**)**

i. **Initial Injection Station**

   This is the starting point of a pipeline transport system. Here the initial pressure boost to pipeline operating pressure is achieved before introducing the gas to the pipeline. This station houses a storage facility and compressors/pumps.

ii. **Partial Delivery Station**

   This is also called an intermediate station whose function is to allow part delivery of the gas or product being transported along the line. This is usually the case where the pipeline is designed to feed different locations along the line.

iii. **Compressor/Pump Station**

This is otherwise known as booster station. Depending on the kind of fluid being transported and the pressure loss along the pipeline; pump or compressors are located at strategic points to boost the line pressure. This station maintains the line pressure to ensure the delivery point pressure is achieved.

iv. **Block Valve Stations**

These serve as gateway; in that they are used to divert flow along a certain pipe segment during maintenance work, or isolate a leak or rupture. The nature of product being transported, the pipeline profile and conditions along the line determines the location of these stations. They are usually located near critical locations such as road and river crossing or in urban areas. They are said to be located every 20 to 30miles (48km) along gas pipeline.

v. **Final Delivery Station**

It is also called delivery terminal. This is the point where the gas is distributed to the end users via distribution network or delivered to tanks in case of liquid.

Other elements include **regulator stations** located downhill along the pipeline profile for pressure relief and **metering stations** for flow measurement along the pipeline. **Pigging** is another element of pipeline operation where sophisticated robotic devices are routinely sent down the pipeline to carry out inspection for corrosion and detection of defects to ensure safe pipeline operation. The functioning of all these elements are linked together via a central control station facilitated by the supervisory control and data acquisition **(SCADA)** systems. Pipelines are nowadays monitored and operated using sophisticated SCADA systems. SCADA systems regulate pressure and flow by monitoring and controlling the compressor/pump stations as well as the positions of valves. SCADA systems work in real time to perform such functions as alarm processing, leak detection, hydraulic analysis, throughput analysis, and other functions deemed critical to the safe operation of the pipeline [7].

2.8 Comparison between CO_2 and Natural Gas Pipelines

It is well known that there are extensive oil and gas pipeline network all over the world; hence it is often thought that these existing pipelines could be used for CO_2 transmission. Also, the maximum design pressure for a natural gas pipeline system is between 50 - 150 bar [56; 59; 64; 67]. Thus the vast majority of existing pipeline which are made of carbon steel will be metallurgically suitable for CO_2 transmission, especially if the CO_2 is sufficiently dry and meets the quality specification required by the CCS. Therefore, the techno-economic experience of natural gas transport can easily be adopted for the CO_2 case.

While this is possible in principle, there are technical limitations. First, oil and gas pipelines are normally operated between pressures above 48 bar (700 psig) and about 82 bar (1186 psig) [7]. In other to keep the CO_2 in dense phase pressures above 80 bar are expected while CO_2 pipelines are expected to operate at pressures within the range of 100 – 200 bar to avoid two phase flow and considering the effect of impurities (existing CO_2 pipelines operate at pressures between 85 - 150 bar).

Second, methane which is the main component of natural gas has critical pressure of 46.2 bar and critical temperature of - 83°C as against critical pressure and temperature of 74 bar and 31°C respectively for CO_2. Thus their flow properties differ considerably. Given the higher critical temperature of CO_2, at typical gas pipeline operating pressure, the CO_2 is in the dense or liquid phase.

Third, the nature and operating condition of CO_2 pipeline will require a tailored hydraulic design and analyses to facilitate the safe operation of the pipeline.

2.9 Cost Evaluation of CO_2 Compression Duty/ Knowledge Gap

The literature is awash with cost prediction models of CO_2 pipeline transport [9]. A segment of CO_2 pipeline transport that has not be given desired attention is the system's energy or compression duty required to move the CO_2 in the pipeline to point of storage. In studies where the analysis of the compression duty is considered, the required energy for compression is either treated as an efficiency penalty on the overall power generation plant's efficiency or assumed to be supplied from the grid. Hence the existing energy cost models are merely a quantification of the power consumption by the compressor or pump through established thermodynamic equations [8; 54; 56; 63; 64; 68].

The power consumption of the compression process is said to be about 40% of the auxiliary load and could constitute 8-12% of the power plants output [60]. Thus, the energy consumed is huge in economic terms and the need for an alternative source of energy becomes desirable. Similarly, the power requirement for booster stations in remote locations along a trunk pipeline where the cost of transporting electricity becomes prohibitive necessitates a suitable prime mover. On the merit of GTs being extensively used in pipelines to compress gas at suitable points, they become prime mover of choice for this duty. While considerable operational experience had been gathered in natural gas transmission, the same is not true for CO_2 transmission especially as the properties and behaviour of the gases are quite different. Furthermore, the CCS as an emerging technology needs to explore all available alternatives to enable an informed decision by policy makers in the near future when its enforcement is envisaged. Sequel, the application of this prime mover for CO_2 pipeline compression duty will require a techno-economic and environmental risk assessment. Thus a gap in knowledge exists which this study attempts to bridge.

2.10 Review of Techno-Economics for Gas Turbine Application in Pipeline

The only known work on this subject is the one by Nasir [69] in Cranfield University, UK. In this study, adapting an existing pipeline profile, the techno-economics of using GT as prime mover on natural gas pipeline was conducted. The optimization of compressor station along the pipeline profile was investigated. The study also made comparative assessment of choosing either a GT or electric motor drive under a given operating condition. The study concludes that the cost of compression reduces with increase in pipe size; however, an increase in pipe size implies pipe material cost rise. Although the converse is true for reduced pipe size; in comparison the increase in operating cost far outweighs the savings in pipe material cost. Also for a specific pipe size and set pressure ratio, the higher the throughput, the higher the compression power requirement. This implies higher operating cost due to more fuel consumption for any increase power demand. The study also concluded that under-sizing a pipe for a particular duty due to anticipated fluctuations in flow capacity is deemed economically unviable and unprofitable in this application. In addition, an economic pipe size is only determined from the interaction between the operating cost and pipe material cost for a particular throughput. Even so, the profitability could be jeopardised if the throughput is below a certain minimum value required to break-even. Also worthy of mention from the study, is the conclusion reached after a comparative analysis of GT emission with the off-site emission from a coal power generation plant that “electric drive may at first seem better in terms of environmental pollution, from the electricity generation today and in the near future, the use of electric motor contributes more to environmental pollution than does GT. Electric drive may not be a viable drive option for an interstate pipeline which often passes through areas where electricity grid is usually not available”.

2.11 Gas Turbine Driven Compression Equipment

Traditionally, gas compression employs the use of high speed reciprocating or diaphragm compressors. However, compression is achieved nowadays by using variable speed electric motor or GT driven centrifugal compressors because of advantage of better efficiency, oil free compression and less maintenance cost [60].

Compared to natural gas, CO_2 has much lower speed of sound due to its higher molecular weight leading to high Mach numbers, thereby reducing the operating range of many compressors [51]. See Figure 2-6 below.

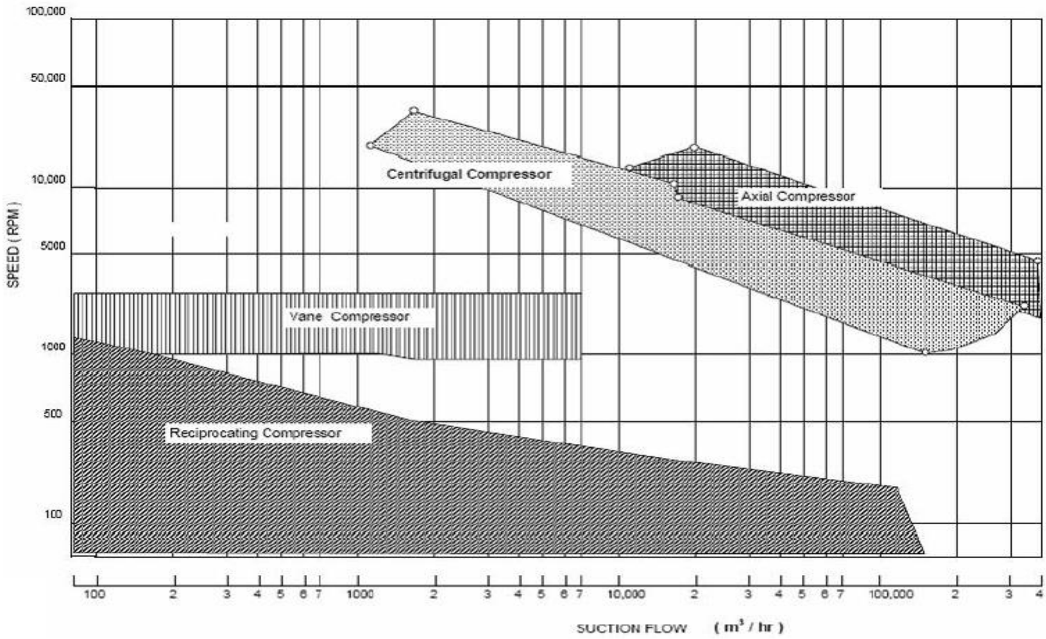

Figure 2-6 Operating Range of Centrifugal Compressor Compared to Other Types

Source: [70]

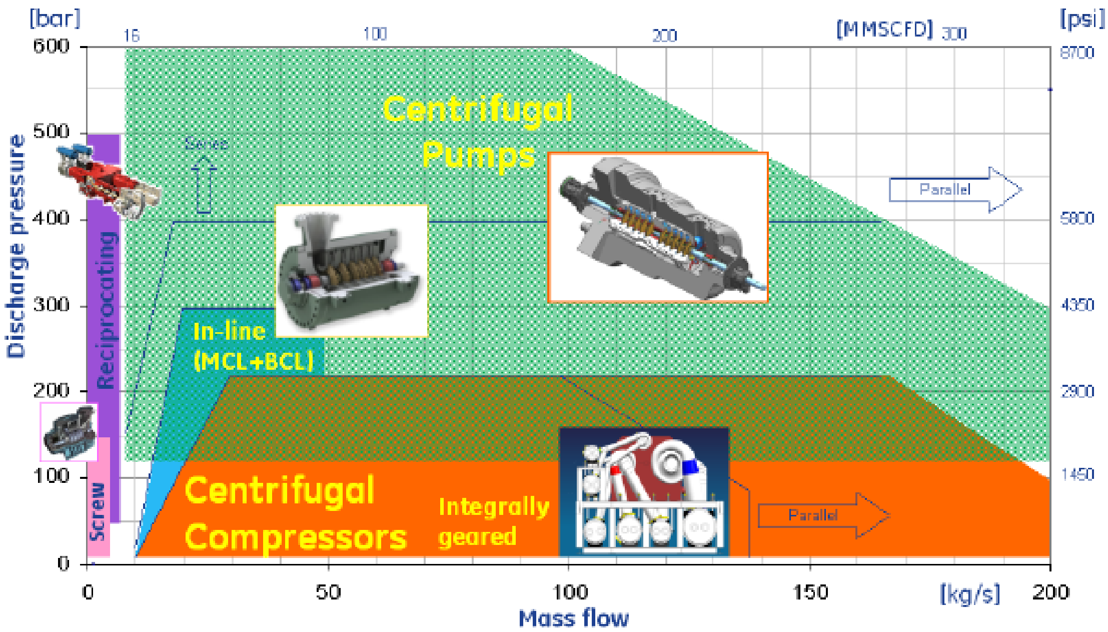

Figure 2-7 Operating Range of Compressors and Centrifugal Pumps

Source: [65]

In order to save compression power, it is proposed that compression of the CO_2 can be done using compressor /pump train. Provided there is sufficient cooling, power saving of about 5 - 15% is said to be achieved when pumps are introduced at compressor discharge pressure of 100 – 130 bar to further compress the CO_2 to very high pressures [54; 63; 65; 71]. Within this pressure range, integrally geared compressor technically gives best economy due to gains from intercooling. Figure 2-7 above shows the operating ranges of compressor/pump train as a function of inlet mas flow for a CO_2 application.

2.11.1 Centrifugal Compressor

Centrifugal compressors belong to the category of compressors known as dynamic compressors or turbo-compressors. It achieves compression (pressure rise accompanied by a decrease in volume and increase in temperature) by imparting inertial force (acceleration, deceleration and turning) to the gas by means of rotating impellers and stationary diffusers. Centrifugal compressors are made up of one or more stages - a stage consists of an impeller and diffuser (vaned or vane less) from the manufacturer's perspective. A stage may also refer to several combinations of impeller/diffuser separated by intercoolers within a compressor. Figure 2-8 below shows a cut-away view of a typical centrifugal compressor.

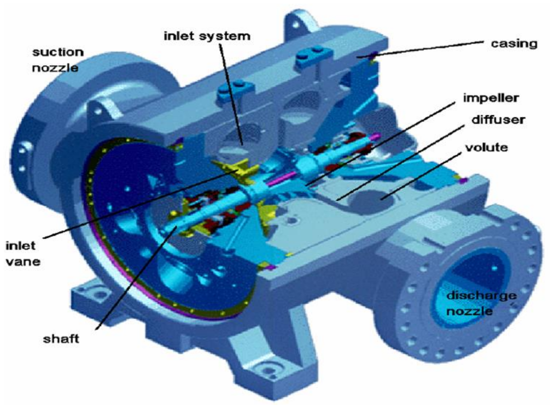

Figure 2-8 Cut-Away View of a Typical Centrifugal Compressor

Source: [45]

The compact nature of centrifugal compressors and the ability to operate at higher speed over a fair range of inlet conditions give them the advantage to cope with the high density of CO_2 and the high flow rate from most CO_2 recovery schemes. In particular, the CO_2 centrifugal compressor has high power density i.e. for the same power consumption compared to the natural gas compressor; it is physically smaller.

Novel internally cooled centrifugal compressors are currently been developed that give some promising features in significant reduction in power consumption. They work on the principle of isothermal compression with integral gear design for inter-stage cooling flexibility and optimization of flow coefficient in the choice of most favourable rotating speed for each pair of impellers at both low and high pressure [40; 54; 60; 72]. Very high performance supersonic CO_2 compressors have equally been proposed to reduce the number of stages to two or three as against the 6-staged conventional ones [73].

2.11.2 Centrifugal Compressor Performance

Compressors are designed to operate at the best efficiency point (see Figure 2-9 below) corresponding to a distinct head (pressure ratio) and actual inlet flow, below or above which losses (aerodynamic and/or mechanical) are incurred; hence reduction in efficiency.

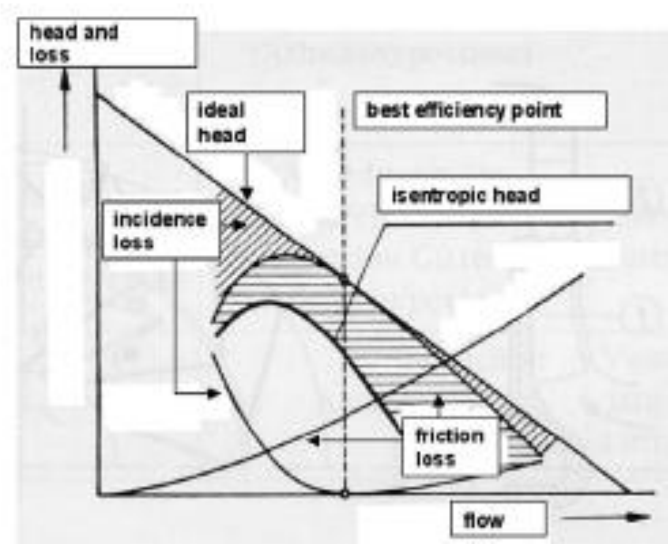

Figure 2-9 Compressor Head versus Flow Relationship at Constant Speed

Source: [45]

A collection of operating points over speed ranges give what is known as the compressor performance map or characteristic shown in Figure 2-10 below. This is always provided by the manufacturer and is obtained from rig test conducted on the equipment. The operational envelope of the compressor is limited by surge (or stall) for lower flows at high head; and choke (or stonewalls) for large flows at low head. The surge control line prevents the compressor from surge while depending on the manufacturer; the choke limit may be relaxed so long as a positive head is maintained. In practice, a choked condition is reached when the head falls below a certain percentage of the head at best efficiency point [45; 74; 75].

The amount of head is also limited by available power, the minimum and maximum speed limit of the compressor, and temperature variation. The speed limit which is imposed by stress limit in the compressor can be improved by the use of beam style compressor as against the overhung design. The addition of inter stage cooling reduces the effect of temperature.

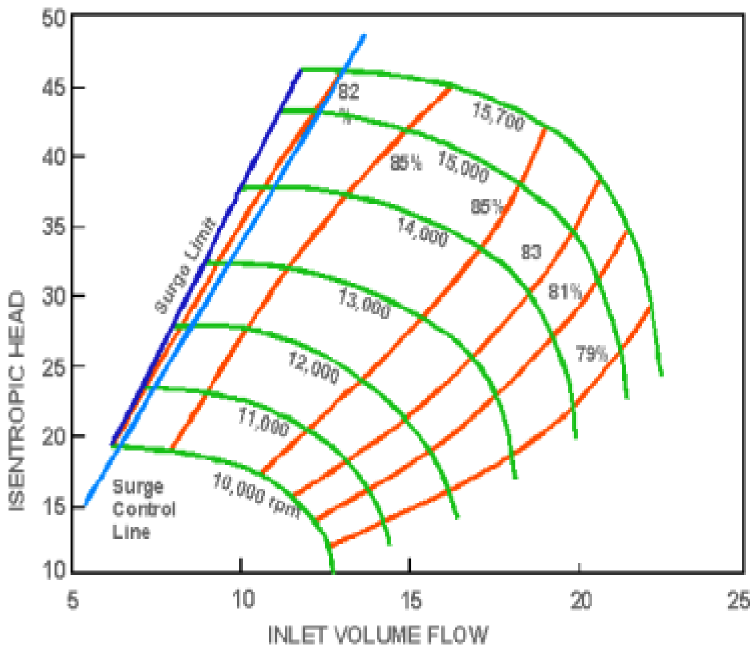

Figure 2-10 Typical Compressor Map (Variable Speed)

Source: [51]

The compressor performance characteristics do not change and its interaction with the system it operates determines the operating point on the compressor. System parameters that determine operating point on the compressor include molecular weight of fluid, suction pressure and discharge pressure [75]. Therefore, the system (pipeline duty) must be carefully matched with the compressor performance to give a cost-effective compressor capacity utilisation [45].

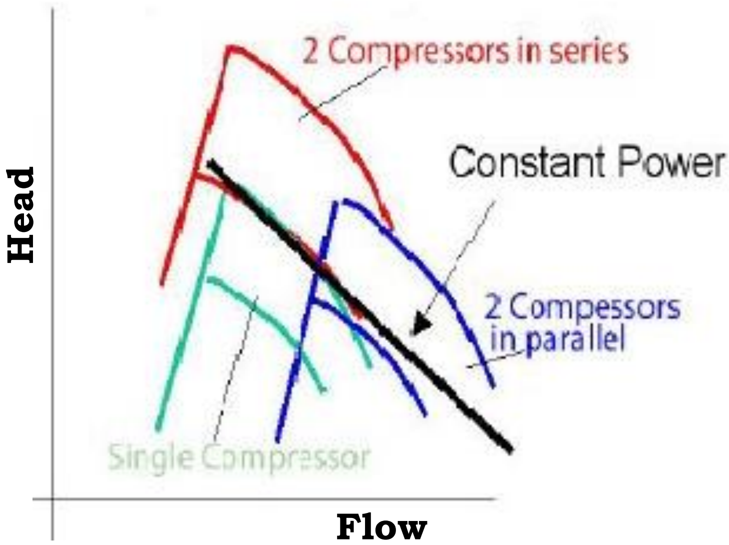

Figure 2-11 Composite Maps for Two Compressors Operating in Series and in Parallel

Source: [74] modified

For a power limited compression system, the range of operations of two or more compressors can be extended by either operating them in parallel or in series(see Figure 2-11 above) to obtain an increased capacity or higher head (pressure ratio) respectively. However, it is often economical to operate the compressor at full load while keeping the others shut down as stand-by than to operate at part-load. The centrifugal compressor is favoured by its ability to operate at full power [45; 51]. Compression power is dependent on the available driver power (often reduced by gear efficiency when a gear connection is used) and the compressor safe operating limits (i.e. head, mass flow and efficiency) or compressor characteristic. Actual power requirement of the compressor is determined by stage – wise (polytropic) analysis to account for the changing thermodynamic parameters influenced by temperature and presence of impurities. The isentropic values can be used for the preliminary selection, while the performance is corrected with the stage wise analysis.

2.11.3 Centrifugal Compressor Selection

Generally, compressors are designed and manufactured to meet the process demand or requirements as specified by the user. The user must have a good understanding of the system resistance or characteristics to guide the recommendation of control system to the manufacturer. Hence they are said to be tailor made. Centrifugal compressor selection requirement include among others:

- Composition of gas to be compressed
- Flow range (mass flow in kg/min or volume flow rate in m^3/min)
- Inlet temperature and pressure ranges for each operating condition
- Pressure ratio required or discharge pressure
- Type of driver and
- Economic consideration (capital and service operating cost)

Centrifugal compressors has typical flow ranges of 500 m^3/hr to 300,000 m^3/hr and multi-stage pressure ratios of over 20 could be attained. The fact that compressors do not always operate at rated best efficiency point due to process demand, families of frames are usually developed by manufacturers that cover range of possible flow rate. This ensures engineering and manufacturing cost optimization [76]. The guidance code for centrifugal compressor selection can be found in API 617.

2.11.4 Centrifugal Pump

Centrifugal pumps fall in the class of pumps referred to as rotor dynamic pumps. They are simple, compact, less expensive, and reliable. Their ability to operate over a wide range of pressure and flows make them well suited for pipeline applications, hence, they are commonly used in the oil and gas industries. Centrifugal pumps are highly efficient and they can operate at very high speeds. Existing centrifugal pumps can give an output of 25 MPa; although GE Oil & Gas claimed to have successfully pushed the discharge pressure to about 60 MPa [65; 76; 77].

Supercritical CO_2 share some physical properties with light hydrocarbon like ethane which have been transported via pipeline using multistage pump of the type shown in Figure 2-12 below. Thus, existing experience with centrifugal pumps used in the petrochemical and process industries have influenced the technological development of centrifugal pump for supercritical CO_2 application. A major design consideration with pumps is sealing. A seal primarily contain the fluid within the pressure envelope of the pump. Dry gas seals have been proven to give excellent results in these pumps while liquid seals are used in very high pressure application [65].

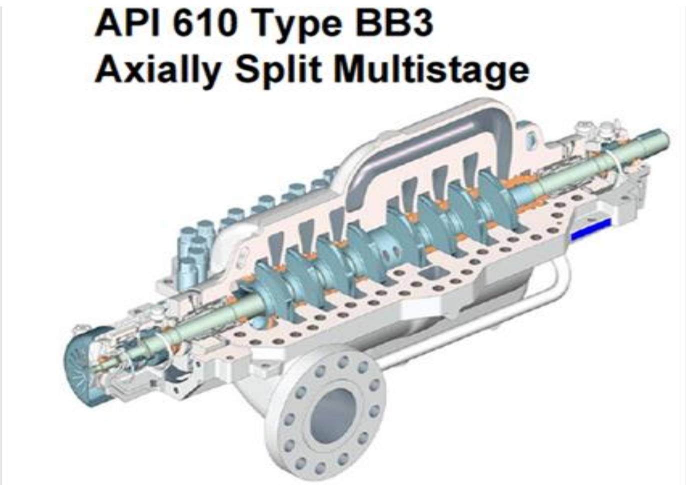

Figure 2-12 A Typical Gas Turbine Driven Centrifugal CO_2 Pump

Source: [77]

2.11.5 Centrifugal Pump Performance

The purpose of the centrifugal pump is to move fluid at a specified flow rate while increasing the head or pressure. Thus, the performance of a centrifugal pump is similar to the centrifugal compressor; the only difference being that the former ‘compresses’ liquid while the latter compresses gas [39; 78].

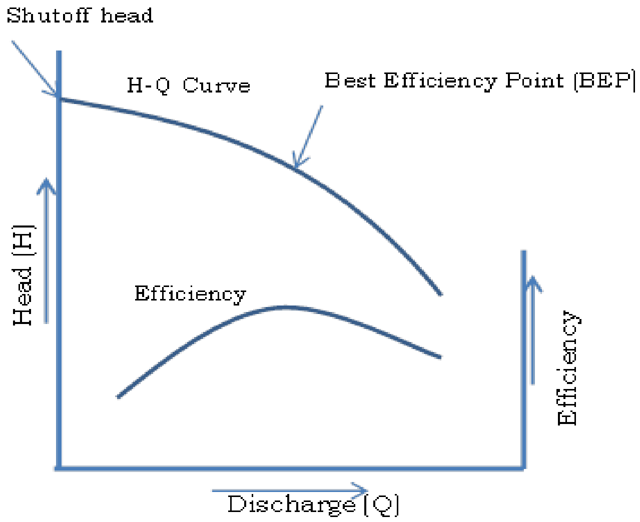

Figure 2-13 Performance Curve for a Centrifugal Pump at Constant Speed

Source: [39] modified

Figure 2-13 above depicts the head – discharge curve of a centrifugal pump. It is essential that they operate close to the optimum or best efficiency point; because similar to the centrifugal compressor, hydraulic losses occur at both extreme of the best efficiency point (BEP). The system or process flow determines the operating point on the pump performance curve. Operating the pump near shut-off leads to vibration causing losses; while operating beyond the optimum leads to losses manifesting in the form of cavitation and water hammer. Therefore, operating the pump within 60 – 120% of the efficiency is considered within safe limits.

Centrifugal pump for CO_2 application differ from the conventional ones due to some thermodynamic considerations accompanying the supercritical nature of the CO_2 [65].

- First, higher speed is required to give the required discharge pressure to meet the constraints in the number of stage as the CO_2 gets warmer.
- Second, the change in density and high compressibility factor due to the rising temperature during pumping must be accounted for in rated

power; hence stage by stage analysis demanding the use of polytropic parameters instead of isentropic ones becomes obvious.

- Third, the effect of impurities a times is to shift the pump duty to a higher pressure differential which implies more power consumption. In addition, impurities like N_2 and CH_4 entrained in CO_2 stream affect performance of seals which is an essential component of the pump. Seals are manufactured and selected based on the suction temperature and pressure, temperature range of pump operation, rotational speed and diameter of the shaft.
- Fourth, the pump will be used downstream of the compressor in this application, so optimization for pump suction pressure should be determined against the constraint of refrigeration or cooling required at pump suction for a cost- effective pumping.

Pumps in CO_2 application are expected to operate within pressure and temperature ranges of 86 – 150 bar and 26 - 35°C respectively. Therefore, net positive suction head (NPSH) is not a requirement even if the pressure falls to about 76bar since cavitation is impossible above critical pressure. NPSH is only a consideration when the pump is applied in cold / subcritical pressure operation [65; 77].

2.11.6 Pump Selection

Pumps must meet the process requirement for efficient performance and cost savings. Optimum selection has significant beneficial effect on the life-cycle cost of a machine and driver. The selected pump must match the system resistance for known duty points. The expected flow variation and pressures must be carefully considered alongside the given pump performance curve.

The driver to be selected equally guides the choice of pump. Factors like adequate NPSH at pump suction for all operating condition, modular arrangement for ease of maintenance and preference for barrel type pumps for multi-stage application are necessary selection criteria [79]. Other factors considered in pump selection include seals, bearings, couplings and procurement [49]. The API 610 provides a guidance to pump design and selection.

2.12 Gas Turbine Emissions

More than ever, there is growing concern about the adverse effect of power generation exhaust on the environment and the need to cut down on this emission. The wide acceptance of GT as prime movers for both mechanical drive and electrical power generation has brought to the fore - front the issue of its emissions. In fact the thrust of this study is predicated on emission reduction issues.

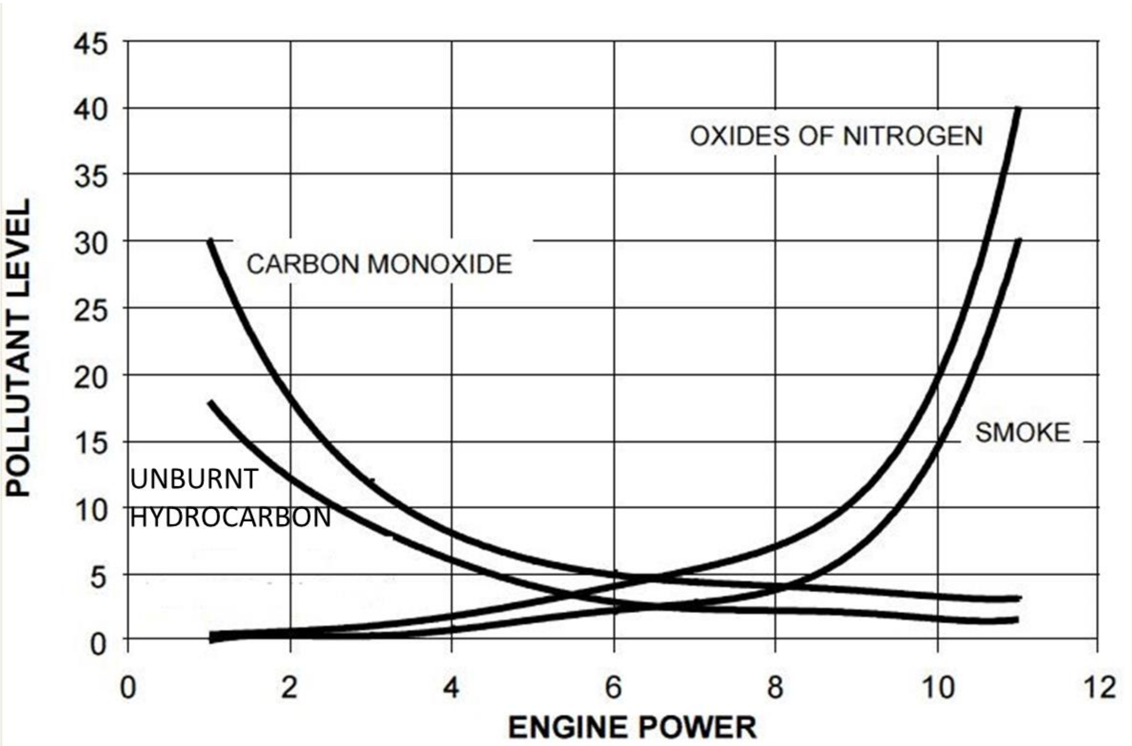

Figure 2-14 Gas Turbine Emissions

(Adapted from [80])

All combustion systems including that of GTs produce pollutants namely carbon dioxide (CO_2), oxides of nitrogen (NO_x), oxides of sulphur (SO_x), Carbon Monoxide (CO), Unburnt Hydrocarbons (UHC) and smoke / soot [17; 80]. SO_x is a major component of acid rain while NO_x in addition to acid rain, causes depletion of the ozone layer leading to incidence of skin cancer. Prolonged exposure to NO_x could cause respiratory illness, impaired vision, headache and allergies.

NO_x also reacts in the presence of sunlight to produce smog (brownish cloud). CO causes asphyxiation (reduction in oxygen carrying capacity of the blood) and fatal if significantly inhaled. UHC can be toxic and could combine with NO_x in a photochemical reaction to form toxic smog. Although some of these pollutants are a small portion of the GT exhaust, the large flow of exhaust gas could lead to large quantities being released to the atmosphere.

Strategies have been developed to reduce the concentration of these GT pollutants. Figure 2-14 above shows the dynamics involved in generating these pollutants. Sulphur content in a fuel can be removed since it determines the emission of SO_x. CO and UHC are concerns at low engine power especially for industrial GT application of this nature where part-load operation is prevalent. However, the development of dry low emission (DLE) combustors in GT has considerably reduced the emission of NO_x, UHC and CO [44].

2.13 Gas Turbine CO_2 Emission

CO_2 is the product of complete and efficient combustion of fuel containing carbon e.g. natural gas which is the predominant fuel of GT.

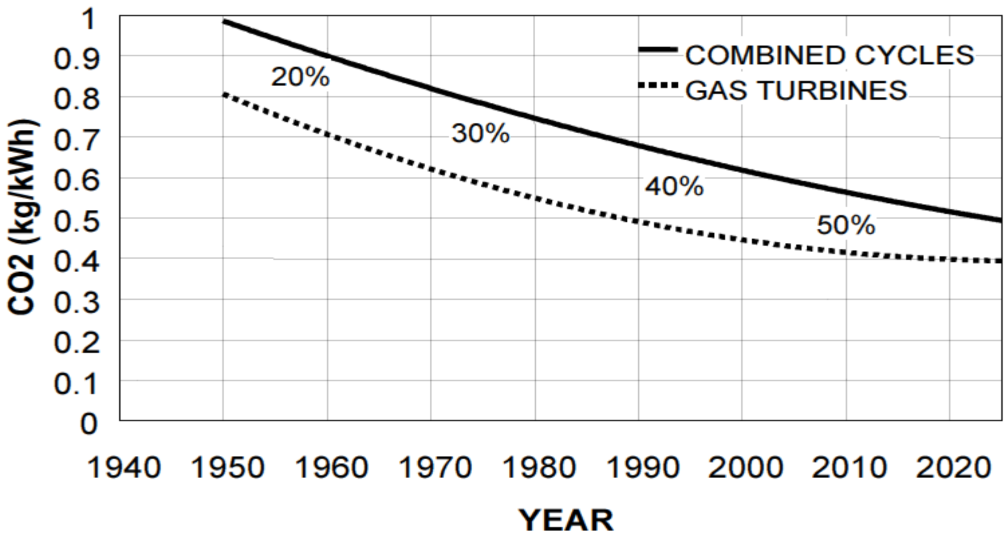

Figure 2-15 Carbon dioxide Emissions

(Adapted from [80])

CO_2 is considered non – toxic; however, it is a GHG associated with global warming. The emission of CO_2 depends primarily on the type, quality and quantity of fuel fired in the GT. The reduction of CO_2 is achieved by improving the thermal efficiency of GT, thereby saving fuel burn. Figure 2-15 above shows a modest projection of CO_2 emission reduction with improved cycle efficiency.

The improvement in thermal efficiency is tied to increase in combustor firing temperature. Incidentally, increase in firing temperature leads to formation of NO_x. Thus, while material technology limit was thought to be a barrier to further improvement in efficiency, emission consideration constitutes another barrier. Apparently, the technological solution for carbon capture will have to be stepped up.

2.14 Gas Turbine Cost Appraisal

Important criteria that affect the economic success of GT in any application include initial cost, running cost (especially fuel cost), life cycle cost and emission. The choice of GT for a particular application has great impact on cost, fuel consumption, operational flexibility, as well as the availability [48].

2.14.1 Capital Cost

The capital cost of GT consists of the initial cost and installation cost. The initial cost is highly influenced by the level of technology which determines the design and the component materials. The requirement to derate the GT mechanical drive due to site ambient condition to provide the required power output is another factor that affects the initial cost. Installation cost include the labour and equipment cost required to install the GT on site. It also comprises the shipping cost and all operational cost incurred to bring the GT to working condition.

2.14.2 Running Cost

This is otherwise known as Operation and Maintenance (O & M) cost. The running cost includes routine maintenance (like change of lube oil and air filters), repairs and overhauls. It also include holding of spares and labour to keep the GT in good working condition. Maintenance is either scheduled or condition based and could involve shut down of the engine.

The cost of fuel contributes a large share to O & M cost in GT operation; in fact it could account for over two-third of the annual operating cost. The cost of fuel depends on the fuel consumption of the engine, the global oil and gas market price and the cost incurred in transporting the fuel to the point of use. In economic analysis, it is usually treated separately.

2.14.3 Life-Cycle Cost

During the life-time of the GT, other cost different from the O & M cost are usually incurred. These include the cost of insurance premiums and depreciation. It may also be in form of incentives as credit tax depending on the prevailing condition.

2.14.4 Emission Cost

The growing concern over the effect of GHG on the environment has seen many governments taking steps to cut down the emission especially from power generation plants. A major pointer to this, is the Kyoto Protocol of 1997 in Japan - where every participating country is saddled with the responsibility of restricting emissions below a certain limit [3]. Thus, regulations have been adopted to give CO_2 (a major GHG) avoidance economic benefits. Owing to release of CO_2 as by- product of combustion in GT engine to the atmosphere, the cost of its emission is considered in economic evaluation.

2.15 Economic Evaluation Appraisal

Generally, the deployment of GT as prime mover is a capital intensive investment. Its attractiveness to any potential investor will be tied to its economic benefit or financial gains.The GT prime movers are mostly designed for a plant life of about 25 years or more; thus, the value of investment projects into the future. The future is beclouded with uncertainties, thereby hinging the value of investment on factors of time, return on initial investment, and risk. The time factor deals with the sustainability of an investment over its entire life cycle taking into account the returns. Returns are measured in terms of cash in–flow and out-flow in the investment taking into account the time value of money. Risk is a measure of uncertainty and volatility of returns because profit does not equal cash. Cash can have different values at different times and hence resulting profit.

The bottom line being that, investment in GT prime mover for this application must be planned considering the dynamics of operability and marketability associated with futuristic projections. Several methods of evaluating the economic viability of projects can be found in literature [81-83]. These include:

a. Net Present Value (NPV) method.
b. Accounting Rate of Return (ARR).
c. Payback and Discounted Payback methods.
d. Internal Rate of Return (IRR) method.
e. Levelized Costing method.

Although all these methods could be used for economic appraisal of projects, in ranking of projects, some are deficient and would not give an accurate assessment. In comparison, the NPV method stands out to be least disadvantaged because it contains fewer assumptions and cannot easily be misapplied. Furthermore, it is easy to interpret and it measures the value of investment which is of interest to the investor [81].

In this study, the NPV method was selected for the economic analysis presented in chapter 6.

2.15.1 The Net Present Value (NPV) Method

This is built upon a discounted cash flow analysis. The NPV takes into cognizance the time value of money and is defined as [81; 83]:

$$\mathbf{NPV} = \sum_{t=1}^{n} \mathbf{C_t} \, [\mathbf{1} + \mathbf{i}]^{-\mathbf{t}} - \mathbf{I} \qquad \textbf{[Eq. 2-2]}$$

C_t is the net cash flow at the end of year t; i is the discount rate; n is the project's life span (in years); I is the initial cost of the investment.

Thus, the NPV is calculated by discounting the yearly cash flows over a project's life time to a present value using a carefully selected rate of discount. The result obtained gives an estimate of the wealth generated by the project. The higher the positive value achieved, the better its attractiveness [82].

2.15.2 Pay Back Period

The payback period (PBP) is an economic index used to evaluate how long it takes to recover the initial investment. In other words, PBP is the time taken for the cash out-flows of a project to equate the cash in-flows. Payback period is a measure of risk and does not measure profitability as it ignores cash inflows after payback [82]. Hence, an investment with a high positive NPV and long pay-back period may be potentially risky. The shorter the pay-back period of an investment the lesser the financial risk.

The payback period based on the discounted cash flow analysis is defined as the minimum value of n that satisfies the equation:

$$\boldsymbol{I - \sum_{t=1}^{n} C_t \, [1+i]^{-t} \leq 0} \qquad \textbf{[Eq. 2-3]}$$

2.16 Concluding Remarks

The basic essentials of the GT prime mover has been highlighted. The requirements of a CO_2 transportation system; physical properties of CO_2 and the performance characteristics of the GT driven equipment required for the compression processes has been reviewed. The very nature of the CO_2 gas impacts on the design of the turbomachinery equipment especially the aerodynamics of the blades. The need to carry out assessment of the GT prime mover for CO_2 pipeline application has also been identified as a gap in knowledge. Equally reviewed were the different economic evaluation criteria; since economic performance of the GT is of interest in this study. The next chapter presents the modus operandi employed to tackle the identified gap in knowledge.

3 Research Methodology

3.1 Introduction

This chapter presents the approach used in carrying out the analyses of this study in order to achieve the desired aim and objectives. It basically gives the description of the TERA frame work adapted to assess the application of gas turbine (GT) as prime mover in the pipeline transmission of CO_2.

3.2 TERA Framework for Gas Turbine- Driven CO_2 Compression

The assessment of GT for this kind of application requires accurate performance data of the selected GT prime mover, a robust economic model and considerations for technical and economic uncertainties (i.e. downtime, life-cycle cost and annual inflation). In this light, a systematic framework for this assessment is embodied in the adapted TERA framework for GT – driven transmission of CO_2 in pipeline summarized in Figure 3-1 below. It consists of four sets of integrated modules: Pipeline / Compression module, Engine Performance module, Emission module, and Economic module. The detailed description of the modules are itemised below.

3.2.1 The Pipeline / Compression Module

This module evaluates the power required by selected compressors and pumps to compress the CO_2 from initial pressure to pipeline operating pressure. Codes based on standard equations for estimating energy required for the GT-driven equipment were developed (detailed analysis presented in the next chapter). The compression duty is of two kinds i.e. the initial pressure boost after capture and the pressure boost at booster station situated along the pipeline profile. Typical CO_2 flow capacities from power generation plants obtained from literature were used for the analysis.

In line with the reasoning adapted from [10; 54; 56; 65], the model assumes that at the initial pressure boost, compressor is used to boost the pressure from 1bar (0.1 MPa) to 100 bar (10 MPa); then a pump is introduced to compress it to 150 bar (15 MPa) which is the assumed pipeline operating pressure. At the booster station, pumps only will be required for recompression.

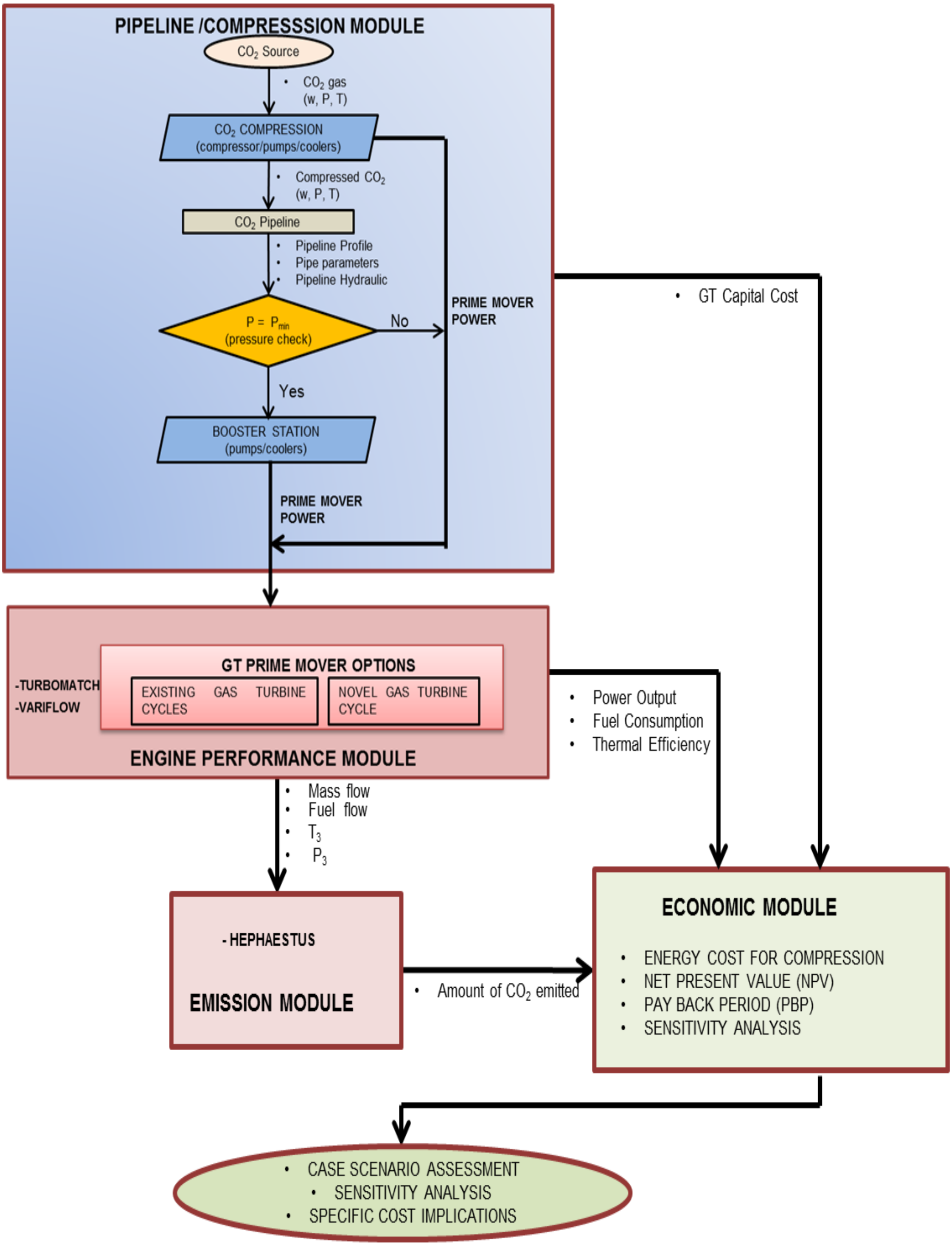

Figure 3-1 TERA Framework for Gas Turbine–Driven CO_2 Compression

However, the need for recompression or otherwise will have to be determined by evaluating the pressure distribution along the pipeline profile through hydraulic analysis of a given pipe diameter and CO_2 throughput at stated conditions. In this regards, the pipeline aspect of the module employs a code developed by the author to simulate the pressure drop along an adapted real life pipeline profile (Sarir oil field to Tobruk terminal in Libya) - Figure 3-2 below to evaluate the pressure drop along the pipeline, hence delineating points of pressure boost. When the pipeline pressure falls below a set value, recompression is triggered. To ensure safety of pumps and in agreement with published works [84-86], 10MPa is set as minimum threshold for recompression.

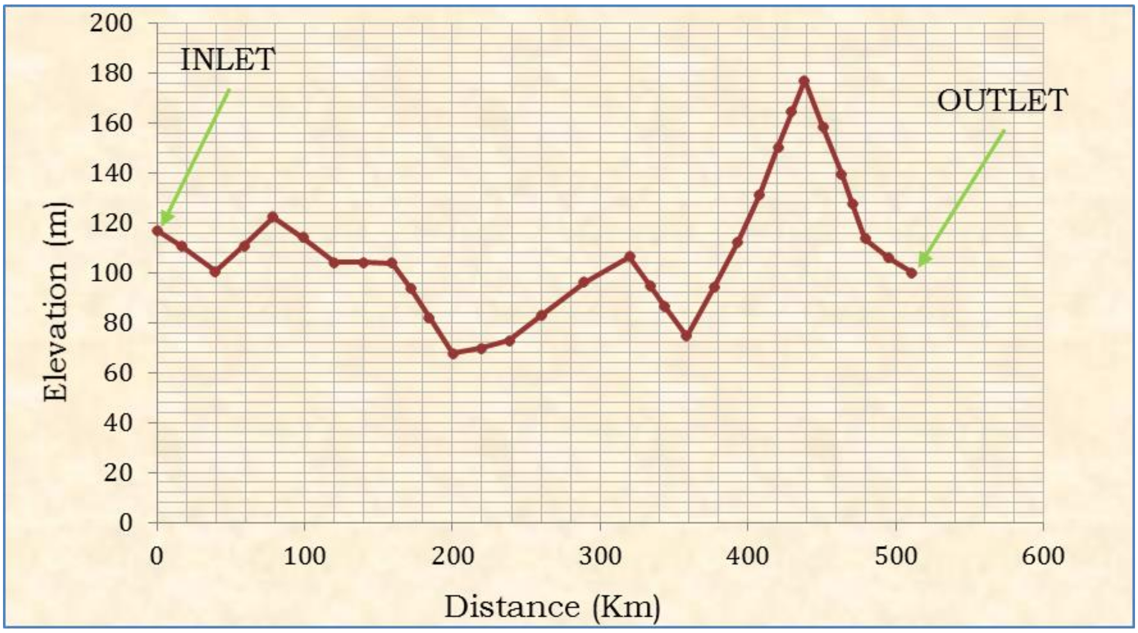

Figure 3-2 CO_2 Pipeline Profile

Source: [87]

3.2.2 The Engine Performance Module

Once the compression duty is established, a GT meeting the power requirement is selected and simulated in this module. The module uses in house GT simulation codes namely **Turbomatch** and **Variflow** to simulate the design point as well as predict off-design performances of the selected GT over expected operating conditions. The GT simulation codes are written in FORTAN; hence the input files for these GTs were formulated in FORTRAN.

The Variflow code is especially useful for simulating purpose-built GT based on a novel GT cycle. The code has also been modified by the author to implement variable stator vanes in order to extend the off design performance simulation of single-shaft GT.

3.2.3 The Emission Module

The environmental impact of utilising GT as mechanical drive with respect to CO_2 emission is addressed in this module. This module requires engine performance parameters (mass flow, fuel flow, combustor inlet pressure and temperature) from GT off-design performance results as input. The amount of CO_2 in "g / kg of fuel" released into the atmosphere by the GT is obtained as output. Other output emission values include that for NOx, CO and UHC which is not considered here due to their current low level in industrial GTs.

The Cranfield University emission code "HEPHAESTUS" is adopted for this work. The code uses a stirred reactor-based approach to predict gaseous emissions from a conventional GT combustor. It essentially quantifies the GT exhaust emission by the use of efficiency correlations and semi – empirical models [88].

3.2.4 The Economic Module

The economic module code developed by the author receives technical data from the other modules to evaluate the profitability or otherwise of deploying GT as a prime mover in this application. A case scenario based evaluation of investment cost / risk in deploying GT using economic indices of net present value (NPV) and payback time (PBP) is conducted. The analysis is based on a year- by –year life cycle cost evaluation using a discounted cash flow model. The associated risk is presented as sensitivity studies on the effect of fluctuation in fuel price and CO_2 throughput.

3.3 Turbomatch Scheme Overview

The Turbomatch Scheme [89] is a GT simulation code developed in Cranfield University by the Department of Power and Propulsion, School of Engineering (now Centre for Propulsion Engineering, School of Aerospace, Transport and Manufacturing) to facilitate the calculation of design point and off-design point performance of existing and novel GT thermodynamic cycles. The code developed in FORTRAN programming language is user friendly and has been continuously improved to meet current design challenges.

The scheme represents the thermodynamic parameters of various components of a GT engine with pre-programmed modular units called “Bricks” using the so called “Station Vectors” (the term “vector” by no means represents magnitude but an ordered set of numbers) and “Brick Data”. The bricks are code named using six capital letters which abbreviates or suggest its purpose e.g. COMPRE (compressor), NOZCON (convergent nozzle), TURBIN (turbine) etc. The bricks linked together using an interface “codeword” provides the ability to simulate the operational state of the engine’s different components. In so doing, result is obtained for the engine’s output power or thrust, fuel consumption, mass flow etc. The scheme assumes the GT uses kerosene fuel with lower heating value of 43 MJ/kg during the simulation.

The detailed information of each component’s performance as well as the gas properties at every station is also provided. The simulation output or performance results are presented in “.txt” files; and in addition has the capability of extracting the parameters of the various components into excel to enable the user carry out desired performance analysis or comparison. The Turbomatch scheme is successfully used for aero, marine and industrial GT performance simulation in majority of projects conducted in Cranfield University

3.4 Concluding Remarks

The TERA framework as presented describes the content of the various modules. The application of the various modules in subsequent chapters to carry out required evaluation and analysis will give detail exposition of the modules. The modules validity / level of confidence is presented in the sixth chapter. The compression aspect of the pipeline / compression module will be presented in the next chapter.

4 CO_2 Compression Modelling

4.1 Introduction

This chapter presents the evaluation of the compression power requirement by the driven equipment (compressor/pump) to compress the carbon dioxide (CO_2) to desired pipeline operating pressure in the dense phase state. An excel code was developed to simulate the required power. The simulation entails multistage compression power evaluation as well as consideration of pressure drop due to intercooling. During the initial compression, the CO_2 is modelled as gas (vapour) up till 100 bar in the compressor and as a liquid in the pump to higher pressure ~ 150 bar. As the CO_2 flow leaves the compressor, it is assumed cooled to $25°C$ at pump suction. Similarly, the CO_2 re-compression at steady state isothermal transmission along the pipeline is modelled as a liquid. Existing works in CO_2 compression / transmission [10; 54; 56; 63] adopted similar line of thought.

Worthy of note however, is that internationally accepted flow simulation programs cannot be easily applied to CO_2 in the dense phase as such programs simulate liquid and gaseous flows only. The properties of the dense phase flow need to be studied and validated on real flow situation to modify these existing programs.

4.2 Model Requirement

The CO_2 is expected to be compressed from atmospheric pressure (1 bar) to the pipeline intake pressure of 15 MPa (150 bar) using a combination of centrifugal compressor and centrifugal pump. At the suction end of the centrifugal compressor, the CO_2 is kept at a maximum temperature of 40°C while the pump's suction temperature is kept at $25°C$. Considering the limitation imposed by maximum head achievable and allowable temperature in the compressor, multistage compression with intercooling is simulated.

4.3 Modelling Assumptions

a. The source of the CO_2 is from fossil- fuelled power generation plants employing MEA (mono-ethanolamine) absorption method for post-combustion carbon capture.
b. The CO_2 is pure ignoring impurities (quality of the captured CO_2 has been specified in the reports on CCS projects and pipeline specification found in references [8; 59; 90-93] with a purity > 95.5 % by volume).
c. The range of estimated CO_2 flow rates in million tonnes per year (MTPY) from power generation plants are as obtained from references [1; 24; 94].
d. The Peng-Robinson equation of state (PR-EOS) is considered.
e. Isothermal pipeline transmission of CO_2 over an adapted pipeline profile.

4.4 Modelling the Centrifugal Compressor Power

Equations for evaluating the head, work or power of a centrifugal compressor in units of Nm/kg, Nm or Nm/sec respectively have been presented in literature. These equations are primarily used to evaluate the adiabatic head [Eq. 4-1]; and with slight modification the polytropic head [Eq. 4-2] as expressed below [45; 74; 76; 95].

$$\boldsymbol{H_{ad} = T_s\, Z_{ave} \frac{R_o}{mw} \frac{\gamma}{\gamma - 1}\left[\left(\frac{P_d}{P_s}\right)^{\frac{\gamma-1}{\gamma}} - 1\right](KJ/kg)} \qquad \textbf{[Eq. 4-1]}$$

$$\boldsymbol{H_{poly} = T_s\, Z_{ave} \frac{R_o}{mw} \frac{n}{n - 1}\left[\left(\frac{P_d}{P_s}\right)^{\frac{n-1}{n}} - 1\right](KJ/kg)} \qquad \textbf{[Eq. 4-2]}$$

Where,

$$\boldsymbol{\gamma = \frac{C_p}{C_v} = \frac{C_p}{C_p - R}} \qquad \textbf{[Eq. 4-3]}$$

$$\boldsymbol{\frac{n}{n-1} = \frac{\gamma}{\gamma - 1}\eta_{poly}} \qquad \textbf{[Eq. 4-4]}$$

Considering the “polytropic head” say, the required compression power or gas power as it is also called is defined by [Eq. 4-5] as

$$\boldsymbol{P_{poly}} = \frac{\boldsymbol{m.H_{poly}}}{\boldsymbol{\eta_{poly}}} \quad (\boldsymbol{KW})$$ **[Eq. 4-5]**

And the shaft or prime mover power is given by [Eq. 4-6] as

$$\boldsymbol{P_{shaft}} = \frac{\boldsymbol{m.H_{poly}}}{\boldsymbol{\eta_{poly}}} \ \frac{\boldsymbol{1}}{\boldsymbol{\eta_{mech}}} \quad (\boldsymbol{KW})$$ **[Eq. 4-6]**

(Mechanical efficiency of about 87% was assumed in the simulation)

In actual application, the compression process is rarely polytropic or adiabatic. Thus, to achieve estimate of the actual compression head, modifications such as the “Schultz polytropic head correction factor”; the Mollier diagrams, or the use of equations of state are readily explored [46; 54; 76; 96].

In the light of the above, the modelling of the centrifugal compressor power presented here was achieved using the PR –EOS [97]. The PR-EOS is said to be among the most precise and proper for engineering application [53; 76]. Further reading about other equations of states, the Mollier diagram and the Schultz polytropic factor can be found in references [76; 96].

[Eq. 4-1] and [Eq. 4-2] can be re-cast in terms of enthalpy difference between the suction and discharge states determined by any two of pressure, temperature and entropy of the compressed gas. Hence the actual compressor head, H_{actual} can be represented as [44; 56; 74; 76];

$$\boldsymbol{H_{actual}} = \Delta \boldsymbol{H} = \boldsymbol{H_d} - \boldsymbol{H_s} \quad (\boldsymbol{KJ/kg})$$ **[Eq. 4-7]**

This is further defined as:

$$\boldsymbol{H_{actual}} = \Delta \boldsymbol{H} = \boldsymbol{H_d} - \boldsymbol{H_s} = \Delta \boldsymbol{H^*}_{\boldsymbol{T_s}} + \int_{\boldsymbol{T_s}}^{\boldsymbol{T_d}} \boldsymbol{C_p}\, \boldsymbol{dT} - \Delta \boldsymbol{H^*}_{\boldsymbol{T_d}} \quad (\boldsymbol{KJ/kg})$$

[Eq. 4-8]

Where, the term ΔH^{*} is called the enthalpy departure function defined by the PR-EOS as:

$$\Delta \boldsymbol{H}^{*} = \boldsymbol{R_O T}(\boldsymbol{Z}-\mathbf{1}) + \frac{\boldsymbol{T}\frac{d\boldsymbol{a}}{d\boldsymbol{T}}-\boldsymbol{a}}{\mathbf{2}\sqrt{\mathbf{2}\boldsymbol{b}}}\ln\left(\frac{\boldsymbol{Z}+\mathbf{2.414}\boldsymbol{B}}{\boldsymbol{Z}-\mathbf{0.414}\boldsymbol{B}}\right) \quad \textbf{[Eq. 4-9]}$$

(the terms of this equation are fully defined in [97]);

$$\boldsymbol{C_p} = \boldsymbol{C_p}\,(\boldsymbol{T}) \quad \textbf{[Eq. 4-10]}$$

(the mathematical expression for [Eq. 4-10] specifically for CO_2 is found in [44].

4.4.1 Modelling the PR- EOS for Compressibility Factor (Z)

To fully solve [Eq. 4-8] for the actual head, the CO_2 compressibility factor needs to be determined in order to evaluate ΔH^{*}. To achieve this, an excel code was developed using the PR-EOS. From [97], compressibility is expressed as:

$$\boldsymbol{Z}^{3} - (\mathbf{1}-\boldsymbol{B})\boldsymbol{Z}^{2} + \left(\boldsymbol{A}-\mathbf{3}\boldsymbol{B}^{2}-\mathbf{2}\boldsymbol{B}\right)\boldsymbol{Z} - \left(\boldsymbol{A}\boldsymbol{B}-\boldsymbol{B}^{2}-\boldsymbol{B}^{3}\right) = \mathbf{0} \quad \textbf{[Eq. 4-11]}$$

Where,

$$\boldsymbol{A} = \frac{\boldsymbol{aP}}{\boldsymbol{R_o}^{2}\boldsymbol{T}^{2}} \quad \textbf{[Eq. 4-12]}$$

$$\boldsymbol{B} = \frac{\boldsymbol{bP}}{\boldsymbol{R_o T}} \quad \textbf{[Eq. 4-13]}$$

$$\boldsymbol{Z} = \frac{\boldsymbol{Pv}}{\boldsymbol{R_o T}} \quad \textbf{[Eq. 4-14]}$$

The terms a, b are fully defined in [97].

4.4.2 Thermodynamic Stage Compression Ratio

The evaluation of the compression head is done in one or more stages of compression depending on the technology which is restricted by the allowable discharge temperature. The stage compression ratio is given by [39; 95; 96]

$$\boldsymbol{r_i} = \left(\frac{\boldsymbol{P_d}}{\boldsymbol{P_s}}\right)^{\frac{\mathbf{1}}{\boldsymbol{N}}} \quad \textbf{[Eq. 4-15]}$$

Considering pressure drop due to intercooling, the stage compression ratio is defined in “Guide to European Compressors and their Applications” referenced in [49] as,

$$\boldsymbol{r_i = 1.05\left(\frac{P_d}{P_s}\right)^{\frac{1}{N}}} \qquad \textbf{[Eq. 4-16]}$$

1.05 is a factor of allowance for the pressure drop through intercooler and pipe work.

In the course of this study, the effect of intercooler pressure drop in prime mover power was investigated; hence the author defines a new relationship for the stage compression ratio given by [Eq. 4-17] as:

$$\boldsymbol{r_i = (1 + \Delta P_{ic})\left(\frac{P_d}{P_s}\right)^{\frac{1}{N}}} \qquad \textbf{[Eq. 4-17]}$$

Where

$\Delta \mathrm{P_{ic}}$ = % pressure drop in the intercooler.

And for ΔP_{ic} = 5%, [Eq. 4-17] reduces to [Eq. 4-16].

4.5 Modelling the CO_2 Thermodynamic Properties

The simulation of the power requirement for CO_2 compression involves thermodynamic parameters that need to be adequately modelled. These include the following:

4.5.1 Density and Viscosity

The excel code of these parameters for CO_2 was obtained from a personal communication with the author of the work in reference [98] who developed set of correlations for the two parameters using experimentally measured data from Kinder Morgan - a well-known CO_2 transporter in USA. The code give density and viscosity estimates over temperature and pressure ranges likely to be encountered in CCS and is thus limited to -1.1 to $82.2\ °C$ (temperature range) and 7.6 to 24.8 MPa (pressure range). Values obtained compares favourably with those obtained from online CO_2 property calculators like NatCarb and National Institute of Standards and Technology [64].

4.5.2 Discharge Pressure and Temperature

The modelling of the discharge parameters from the driven equipment (compressor and pump) begins by specifying the suction (inlet) temperature and pressure. Equally important, is a reasonable assumption of compressor polytropic efficiency (η_{Poly}) and pump efficiency.

The CO_2 from the carbon capture system is assumed to be at atmospheric pressure and cooled at inlet to the first stage of the compressor to about $40℃$ - this temperature being the optimum from the capture process [8; 90; 91]. Constant inlet temperature is assumed at successive stages of the compressor for multi-stage compression.

With known inlet conditions, the discharge pressure and temperature are defined for compressor as [76; 95]:

$$\boldsymbol{P_d = P_s r_i} \quad \textbf{[Eq. 4-18]}$$

Considering pressure drop across intercoolers, the modelling of the suction pressure between stages becomes:

$$\boldsymbol{P_s = (1 - \Delta P_{ic}) P_d} \quad \textbf{[Eq. 4-19]}$$

The average pressure due to non-linearity of flow in the pipeline is defined as [39]:

$$\boldsymbol{P_{ave} = \frac{2}{3}\left[P_s + P_d - \left(\frac{P_s P_d}{P_s + P_d}\right)\right]} \quad \textbf{[Eq. 4-20]}$$

The discharge temperature on the other hand is given by the relationship,

$$\boldsymbol{T_d = T_s r_i^{\frac{n-1}{n}}} \quad \textbf{[Eq. 4-21]}$$

Where the polytropic exponent - $\left(\frac{n-1}{n}\right)$ is as defined in [Eq. 4-4].

At inlet to the pump, the suction temperature is maintained at 25°C through the use of aftercoolers. The temperature rise in the pump upon compression is given by [99] as:

$$\Delta \boldsymbol{T} = \frac{P(KW)*(1-\eta_{pump})}{C_P * Q * \rho} \qquad \textbf{[Eq. 4-22]}$$

The value of $\Delta \boldsymbol{T}$ is relevant to determine whether or not further cooling is required before the high pressure CO_2 fluid is introduced into the pipeline.

4.5.3 Specific Volume and Actual Flow

The specific volume of the CO_2 before and after compression can be evaluated accordingly by the relationship [76];

$$\boldsymbol{v}\ ({m^3}/{kg}) = \boldsymbol{R_O}\ \frac{Z_s}{mw}\frac{T_s}{P_s} \qquad \textbf{[Eq. 4-23]}$$

With a known mass flow rate of the CO_2, the suction volume flow rate can be obtained using the relationship [76];

$$\boldsymbol{Q}\ ({m^3}/{sec}) = \boldsymbol{mv} \qquad \textbf{[Eq. 4-24]}$$

The volumetric flow rate (capacity) is used for compressor specification or size. Its condition is either stated as Normal [*sea level*, $0°C$]; Standard [*101.325 KPa (14.7 psia),* $20\,°C$ or $68\,°F$, *Rel. Humidity-36%*] or Inlet [compressor suction condition] also referred to as Actual. In practice, the inlet condition is specified in units of ICFM (Inlet Cubic feet per minute); ACFM (Actual Cubic feet per minute); and Cubic metre per hour (m^3/hr).

4.6 Modelling the Centrifugal Pump Power

The shaft power required by the pump is simulated keeping in mind the suction and expected discharge pressures, which in this case are 10 MPa and 15 MPa respectively. From [49; 56; 68];

$$\boldsymbol{W_{pump}} = \frac{Q\,(P_d - P_s)}{\eta_{pump}} \quad (\boldsymbol{KW}) \qquad \textbf{[Eq. 4-25]}$$

$\boldsymbol{P_d}$, $\boldsymbol{P_s}$ in MPa; Q in $\mathrm{m^3/sec}$; and $\boldsymbol{\eta_{pump}}$ in %

4.7 Simulating the Required Compression Power

The power required for compression is evaluated for CO_2 flow rate that will meet both available driver power and pipeline compressor actual suction capacity. A maximum of 25 MW driver power is considered while the compressor is expected to operate at best efficiency over the chosen CO_2 flow range. A design actual CO_2 flow rate of **1.5 MTPY** (~**100,000m^3/hr**) at design speed which is within the operational limit specified for pipeline compressor (PCH) as shown in Figure 4-1 below is maintained. Inlet flows above and up to twice this capacity will be assumed driven by the same driver in a parallel flow arrangement. All simulated result is assumed to be at steady state.

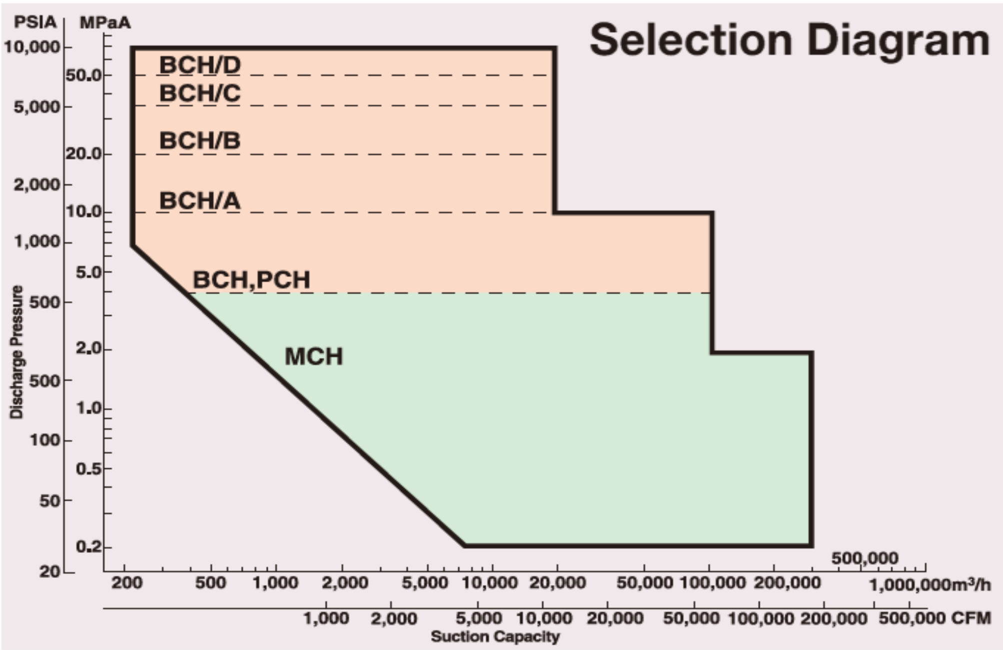

Figure 4-1 Centrifugal Compressor Selection Chart [Courtesy Hitachi Plant Technologies]

A constant compressor polytropic efficiency value of 80% per stage was assumed for the simulation considering existing technology [73; 95; 100; 101]. However, it should be noted that the compressor efficiency value decreases in successive stages for multi-stage compression due to fouling, speed matching and mechanical constraints. A polytropic efficiency value of 90% was also used in the simulation to evaluate the effect of change in efficiency on the compressor head. A Pump efficiency of 75% was assumed in the simulation. A

10% increment on simulated power is applied to meet the API 617 minimum power margin specification for GT driver selection.

4.8 Multi-stage Compression Simulation Analysis

The simulated required CO_2 compression power is presented in a graph of GT power against the CO_2 flow rate for selected multistage compression. The choice of compression stages is guided by the need to evaluate the power requirement of existing CO_2 compressor technology and novel compression processes. It is necessary to recall that compressor inlet temperature "Ts" for each stage is specified as 40 °C and the discharge pressure "P_d" after the last stage as 100 bar. Then, the CO_2 is introduced into the pump after cooling to 25 °C for further compression and cooling to pipeline inlet condition of P_{in} = 150 bar; T_{in} = 25 °C.

The results obtained must be interpreted alongside an understanding of the system demand or characteristic, the compressor characteristic and available power to drive the compressor. The system demand in this case is such that a constant discharge pressure of 100 bar is required, thus the compressor must operate against a fixed head or discharge pressure irrespective of the amount of CO_2 being compressed.

To achieve this, consideration is given to the performance characteristic of the centrifugal compressor which to a very large extent is governed by the "Affinity laws" or "Fan-laws". According to these laws, the behaviour of the centrifugal compressor at speeds other than design is such that the capacity varies directly as the speed; the head developed as the square of the speed and the power required as the cube of the speed. Thus, the compressor meets the process demand per se operating as close as possible to its best efficiency point by means of controls determined by the choice of driver. The controls include [48] : (i) variable speed control (refer to Figure 2-10), (ii) adjustable inlet guide vanes (iii) suction or discharge throttling, and (iv) recycle

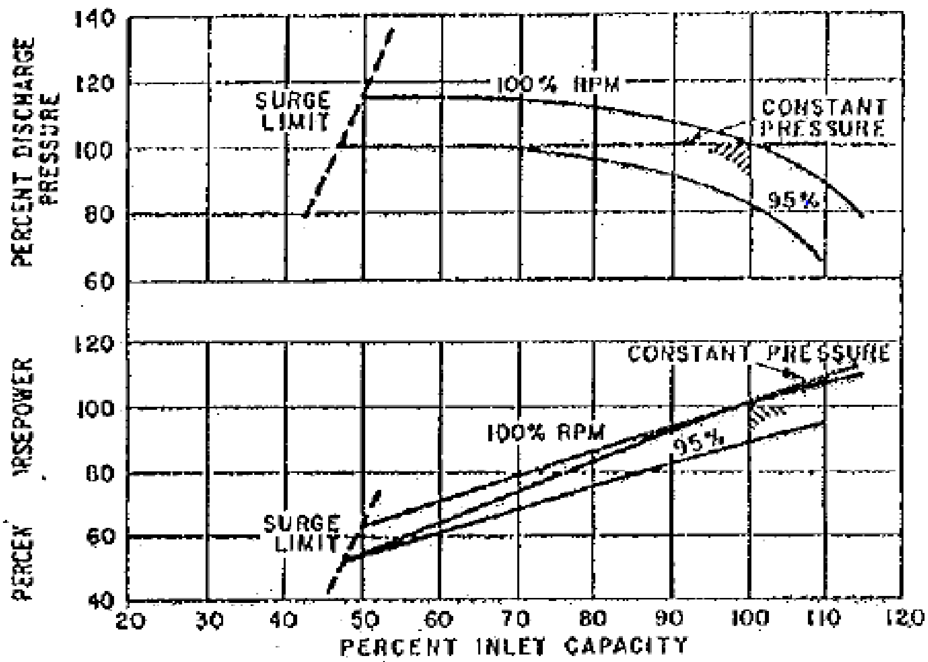

Figure 4-2 Constant Discharge Pressure Control for a Gas Turbine Driven Compressor

(Adapted from [102])

Figure 4-2 above brings to light the issues being highlighted. Assuming a 10% head rise from the compressor's design point to surge; the Affinity law dictates an approximate minimum speed of 95% of design speed to keep the discharge pressure constant with reducing capacity. The implication being that the compressor will operate safely at design operating point down to a minimum of about 50% rated flow to maintain desired system requirement.

4.8.1 Validation of the Simulated Compressibility Factor (Z)

The compressibility factor is crucial to the evaluation of required compression power. Therefore, the simulated CO_2 compressibility values from the code developed using the PR-EOS was validated against the values obtained using Aspen HYSYS – commercial pipeline simulation software with built-in PR-EOS library.

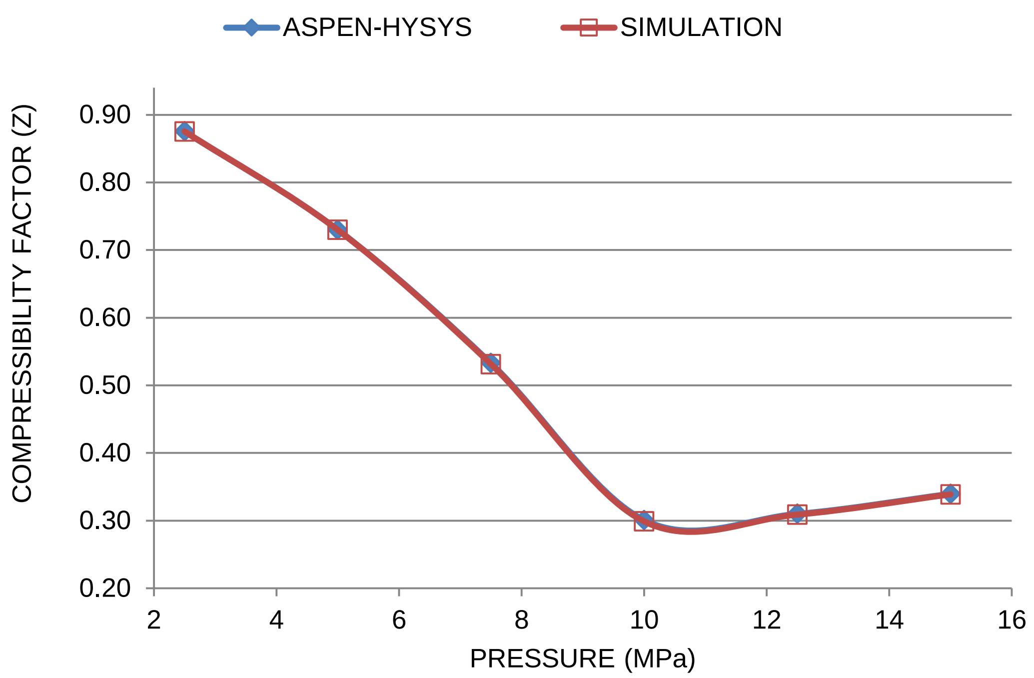

Figure 4-3 Comparison between Simulated Compressibility Factor with PR-EOS Property Table from Aspen-HYSYS for Pure CO_2 at 40°C

The simulated values show good agreement with Aspen-HYSYS as evident from the plot obtained in Figure 4-3 above. However, the PR-EOS like all other equations of state has limits of applicability. Thus, the code will give useful results from zero pressure and temperature to pressures and temperatures of about 30 MPa and 589.29 K respectively.

The limitation highlighted above implies that the compressor head evaluation with [Eq. 4-8] will become inaccurate when fewer compression stages are analysed due to the very high discharge temperature. Interestingly, as one approaches this limit, there is no noticeable discrepancy between the result obtained with [Eq. 4-8] and [Eq. 4-7] which is without the enthalpy departure function term [Eq. 4-9].

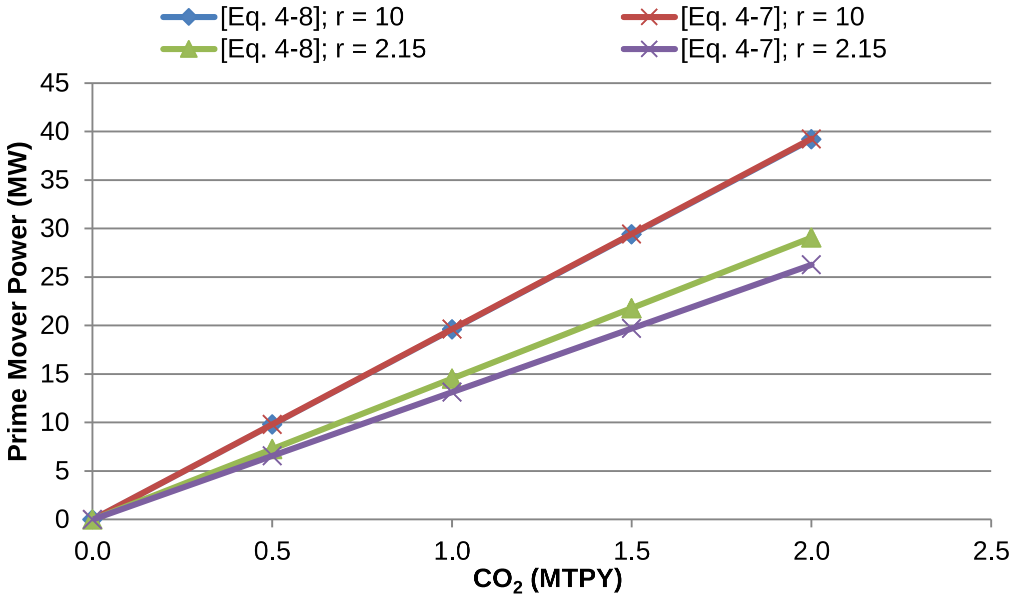

Figure 4-4 Comparison between [Eq. 4-7] and [Eq. 4-8] for Evaluating CO_2 Compression Power at High and Low Compression Ratio (r = 10 & 2.15 respectively)

Figure 4-4 above shows a comparison between results of compressor head obtained using [Eq. 4-7] and [Eq. 4-8] for a two-stage ($\boldsymbol{r = 10}$) and six-stage ($\boldsymbol{r = 2.15}$) compression.

The deviation between the two equations at the lower compression ratio can easily be observed due to the effect of the enthalpy departure factor [Eq. 4-9] accounted for in [Eq. 4-8]. However, at a higher compression ratio ($\boldsymbol{r = 10}$) with discharge temperature - "T_d = 380K", the two equations give similar results. Therefore, results obtained at temperatures above 589.29 K, when the PR-EOS become inapplicable is adjudged reasonable. It is worthy of mention that the maximum pressure required for the CO_2 pipeline is 15 MPa which is within the EOS applicability range.

4.8.2 Compression Power Saving with Compressor and pump Combination

A saving in compression power requirement is said to be achieved when the CO_2 compression is undertaken using a combination of compressor and pump.

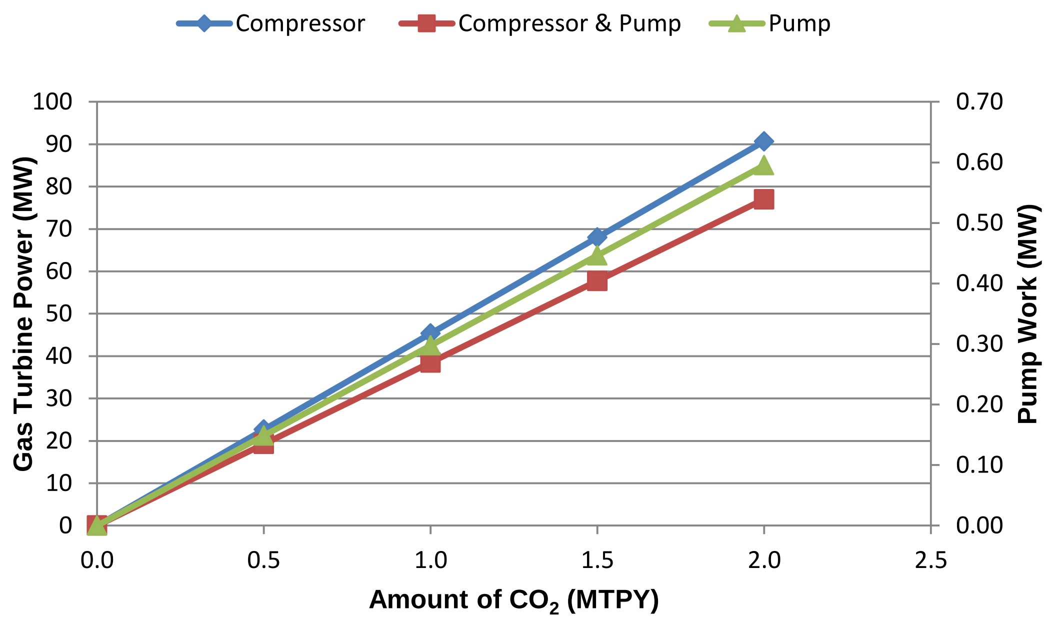

Figure 4-5 Comparison between 1- Stage Compression Power Requirement using Compressor only and in Combination with Pump

Indicated compression power saving of 5-15% is mentioned in literature for the compressor and pump combination [54; 63; 65; 71]. Ignoring material and seal restrictions, Figure 4-5 above technically demonstrates the proof of concept from the huge difference in GT power requirement when a compressor alone is used compared to its combination with pump. Analysis of the results obtained show a pump power of about 400 KW is required at maximum design CO_2 capacity and 15.2% saving in power with such combination.

4.8.3 Power Requirement for 10, 8, and 6 Staged Compression and Effect of Intercooler Pressure Drop

Figure 4-6, Figure 4-7, and Figure 4-8 below depict the GT power required for 10, 8 and 6 compressor stages of CO_2 compression respectively within the operating conditions earlier highlighted. The process requirement is to give a constant discharge pressure which is accomplished by varying the speed of the GT driver. In so doing, the compressor efficiently develops a constant head by lowering its speed at flows below the rated flow. As a result it can be observed from the graphs that as the CO_2 mass flow rate increases, the power requirement increases.

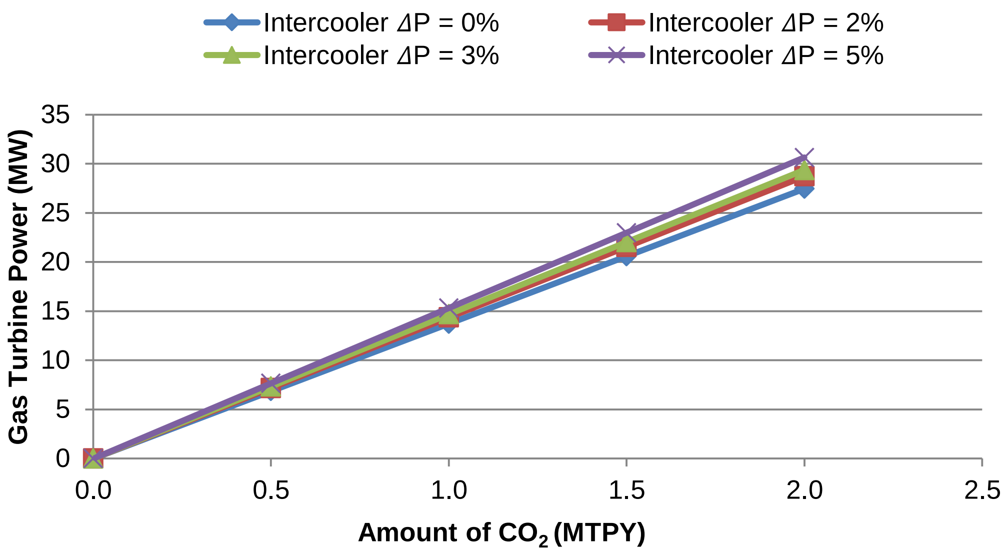

Figure 4-6 Gas Turbine Power Requirement for a 10 - Stage CO_2 Compression

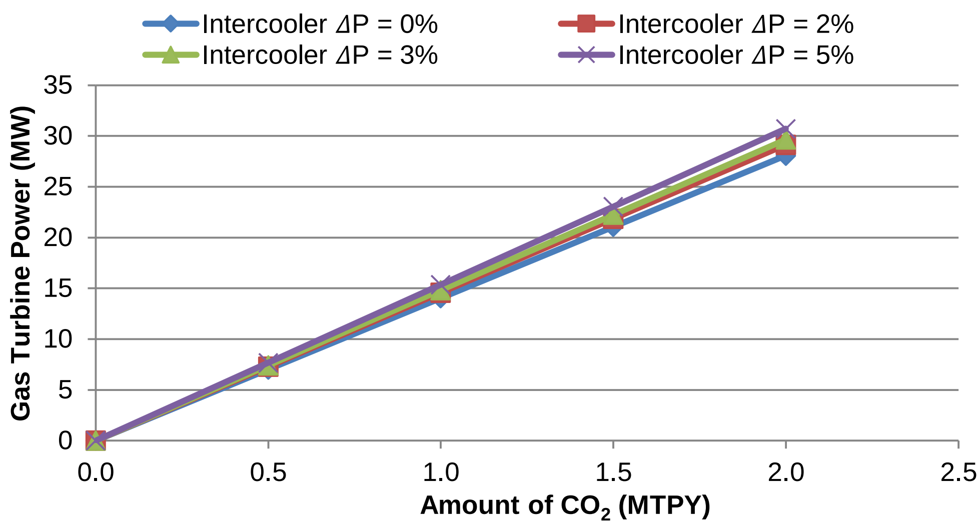

Figure 4-7 Gas Turbine Power Requirement for an 8 - Stage CO_2 Compression

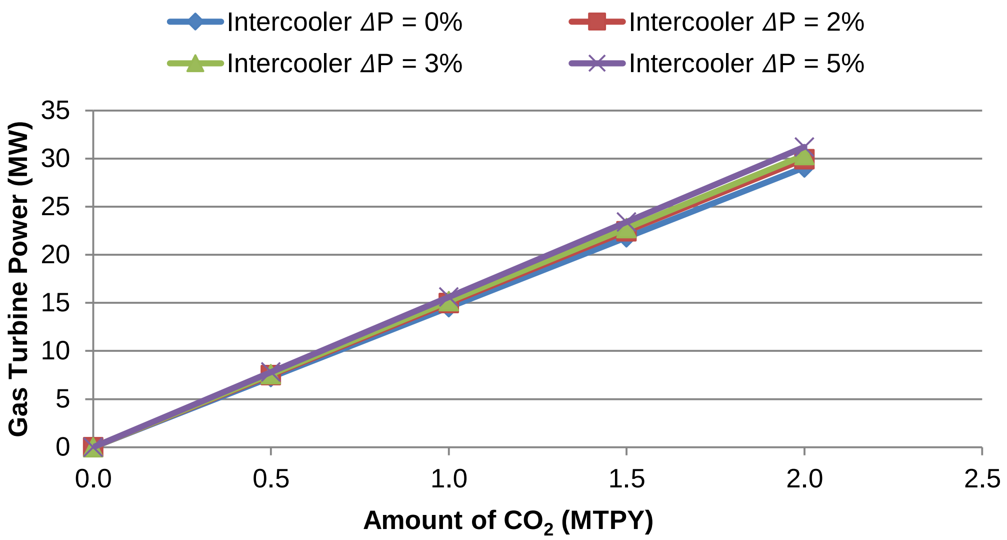

Figure 4-8 Gas Turbine Power Requirement for a 6 - Stage CO_2 Compression

Also deducible from the plots is the steady increase in compression power requirement as the intercooler pressure drop increases. This is necessary to ensure the required discharge pressure is attained during the compression process.

At the rated flow of 1.5 MTPY, the power required considering 5% intercooler pressure drop (say) are 22.98 MW, 23.03 MW and 23.41 MW for the 10, 8, and 6 compression stages respectively. The values indicate about 2% increase in power requirement as the number of stages reduces from 10 to 6 (compression ratios varying from 1.58 - 2.26). The rise in compressor head or otherwise rise in stage compression ratio with fewer stages accounts for this development.

From the operational point of view, the increase in required power means increase in energy cost. However, this is a trade-off for the compactness and comparative light weight of centrifugal compressors with fewer numbers of stages which is advantageous during installation and scheduled maintenance. Another merit of the increase compressor head from the GT driver performance point of view is the ability to operate near design speed at rated compressor operating point, thus minimizing part load operation. This is so considering the fact that higher head per stage is achieved at higher compressor speed.

4.8.4 Power Requirement for 3 and 2 Staged Compression and Effect of Intercooler Pressure Drop

Ignoring seal restriction with hope of advancement in material and manufacturing technology the power required for a 3 and 2- staged CO_2 compression as simulated are shown in Figure 4-9 and Figure 4-10 below.

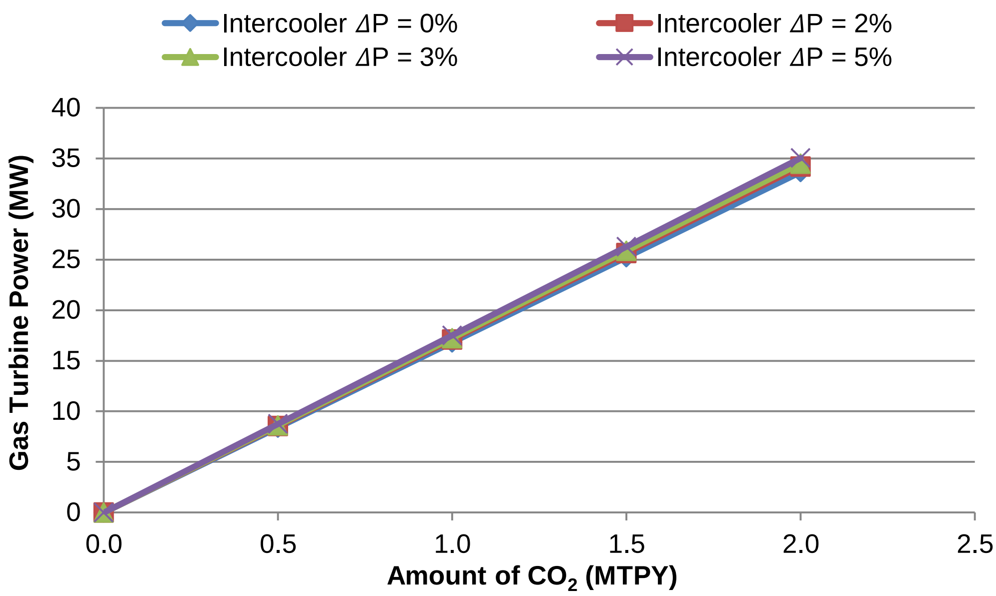

Figure 4-9 Gas Turbine Power Requirement for a 3 - Stage CO_2 Compression

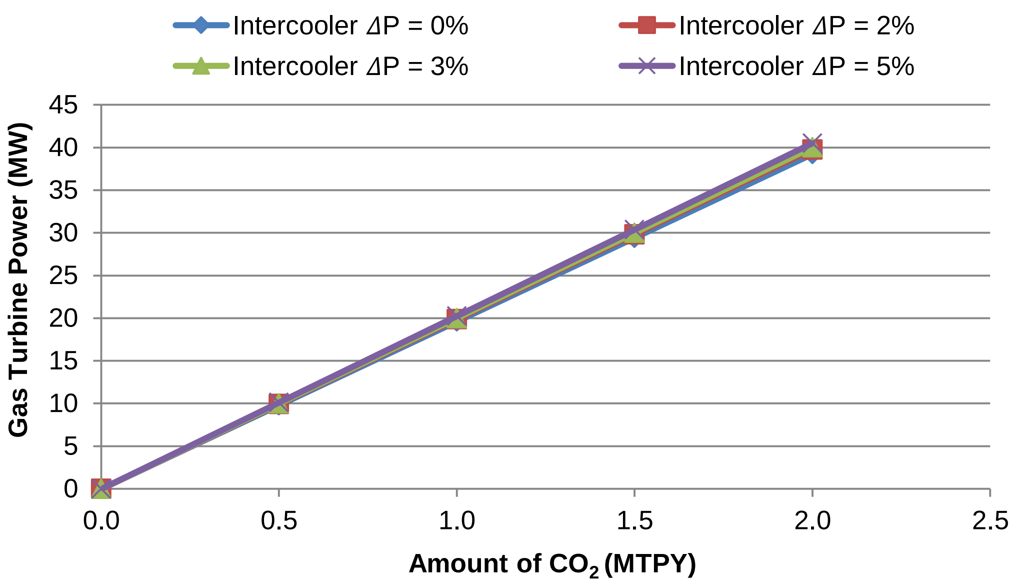

Figure 4-10 Gas Turbine Power Requirement for a 2 - Stage CO_2 Compression

The major motivation of going to this extreme is that this concept is being pursued by some CO_2 centrifugal compressor manufacturers in recent times [73].

The 3-stage compression process gives a discharge temperature of about 198 - 211°C over stage pressure ratio of 4.64 – 4.87. On the other hand, the 2-stage compression whose stage pressure ratio varied from 10 - 10.5 has discharge temperature varying from 309 - 324°C respectively. The discharge temperatures from these processes are quite high. Assuming the CO_2 is stable in the temperature range, given an effective cooling system, the amount of heat released could be tapped for other beneficial use e.g. in heat pumps. However, this is a subject that is beyond the scope of the current study.

At the assumed rated compressor operating point and considering 5% intercooler pressure drop, the GT power required is 26.29MW and 30.38MW for the 3 and 2 –staged compression respectively. Although the power requirement is relatively high, its compactness and the competitive low price advocated by the OEMs will be a huge advantage over the conventional multistage centrifugal compressors. The required power could be lowered with better efficient designs making it more attractive for CO_2 compression application.

It could also be inferred from the plots obtained that compared to the 10, 8 and 6 - staged compression, the effect of intercooler pressure drop is not so evident. This could be attributed to reduction in the number of intercoolers.

4.8.5 Effect of Polytropic Efficiency on Compression Power

Most centrifugal compressor technologies have efficiencies above 75% but below 90% [95]. The more efficiently the compression is achieved the lower the power required from the prime mover. The plot obtained in Figure 4-11 below buttress this point as a marked reduction in compression power is observed with increase in efficiency. In this particular efficiency variation, there is a remarkable 20% difference in the required compression power which is huge in economic terms.

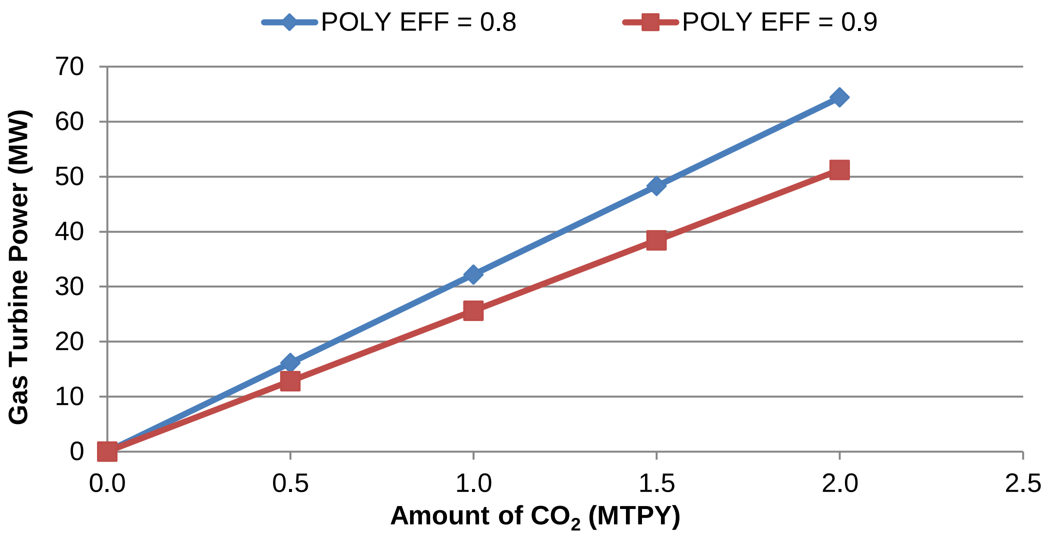

Figure 4-11 Influence of Compressor Efficiency on Gas Turbine Power

4.9 Concluding Remarks

The requirement to compress CO_2 to supercritical pressure brings the need for intercooling in multistage compression process. Thus, issues around the heat exchanger would require all necessary investigation to determine an optimum design for the CO_2 system. The pressure drop across the intercooler need to be considered in the preliminary analysis of the required GT power. Although pure CO_2 has been assumed, the impact of impurity on the compression power is still an area that should be researched. It will also be worthwhile to consider the use of other equations of state in order to compare values.

The next chapter presents a modification of one of the in-house GT simulation code used for performance analysis.

5 Modification of Variflow Code

5.1 Introduction

Variflow is an in-house gas turbine (GT) performance simulation code developed in Cranfield University. Over the years it has been subject to improvement through modification of its subroutines to cater for specific needs. The need to further extend the off-design performance prediction of single shaft GT using this code prompted the modification discussed in this chapter. The modification done here is the implementation of variable stators.

5.2 Performance Simulation of Industrial Gas Turbine

The ability to represent GT engine data with models or codes that reflect the thermodynamic behaviour of such engines has become a necessity to OEM and users. The use of GT simulation codes or software in the design studies, performance prediction and diagnostic as well as life cycle analysis is a common trend nowadays. Performance prediction methods are well described in reference [17] and they form the basis of any GT simulation software / code.

The area of interest in the current study, is performance prediction of industrial GT for mechanical drive application. The GT simulation consists of design point and off-design performance evaluation of selected GT from the public domain. The engine manufacturers will normally provide performance specification of GT at ISO (International Organisation for Standardisation) condition i.e. at 101.325 KPa (14.696 psia), sea level, ambient pressure; $15\,°C$ ambient temperature; 60% relative humidity; and zero installation pressure losses [44]. Information usually provided include the power output, thermal efficiency (or heat rate), compressor pressure ratio, exhaust gas flow (or mass flow) and exhaust gas temperature. The design point performance of a selected engine is simulated to closely match its specification at ISO. Once the GT engine leaves the manufacturer’s bench, all measured performances are off-design. Hence, informed decisions or analysis are made based on the ability to predict the performance of the GT over its expected running range. A major off-design variable is ambient temperature change whose influence over the GT power output is of utmost importance to the user.

Economic consideration borne out of the likely nature of the CO_2 pipeline operation requiring the GT driver to cope with varying power settings brings to fore the necessity for performance analysis.

5.3 The Variflow Code

The initial version of the Variflow code was developed and validated by a doctoral researcher in Cranfield University to enable the performance analysis of single shaft GT power cycles [103]. It is used for design point and off-design performance analysis and has the capabilities of handling different fuels and working fluids. In fact, the code was used extensively in Cranfield University to conduct research related to CO_2 abatement in GT power cycles for the “IEA Greenhouse R&D Programme". The detailed simulation procedures and assumptions for the design point and off-design calculations can be found in reference [103].

The Variflow code is written in FORTRAN and has undergone several modifications over the years. The most recent modification (in 2006) is the inclusion of water injection and division of the compressor into four parts by an MSc student [104] in Cranfield University. In the existing (as handed) version, the code consists of a main program which calls twenty-eight (28) other subroutines to carry out performance simulation of single shaft GT cycle.

5.4 Modelling Variable Geometry Compressor in the Variflow Code

The existing Variflow code is only able to simulate GT performance with a fixed geometry compressor. In order to extend the off-design performance prediction capabilities of the code, the author modified the existing Variflow code by developing subroutines to implement variable stator vane angles. The details of this modification will be discussed in section 5.5. However, it will be worthwhile to highlight some basic principles considered in developing the algorithm for implementing the variable geometry in the GT axial compressor as well as the effect on performance.

5.4.1 Performance Enhancement Using Variable Stators

The GT mechanical drive is expected to perform satisfactorily within the required power variation envisaged in the pipeline CO_2 compression application. At low power settings and during start up, there is high propensity for surge in the GT axial compressor with undesirable effect if left unchecked. One of the methods employed to ensure an acceptable surge margin for start and load acceptance conditions is the use of variable inlet guide vane and stators (VIGVs and VSVs) or simply variable stators (VS) in axial compressors (others include multi-spooling and use of blow-off valves)[105]. The use of VS essentially involves repositioning stator blade passages (vane angles) to control the amount of air mass flow into the compressor [106]. In so doing, the compressor characteristics or geometry is altered; hence such compressors are referred to as variable geometry compressors [36]. IGV adjustment, enhance the part-load performance of the GT drive as it gives better matching of various components in industrial GT. In addition, the application of variable stators is especially useful in single shaft GT as a power control strategy for improving combined cycle efficiency through the control of turbine exhaust gas temperature [36; 107].

5.4.2 Compressor Map and Beta Line

In order to consider variable stators in the performance simulation, compressor performance maps for several vane angles were employed. These maps relate the basic thermodynamic parameters that define the overall characteristics of a GT axial compressor which are usually represented using “non-dimensional groups” (the groups are actually quasi-dimensional). These are:

- Pressure ratio (PR)
- Non-dimensional Mass Flow $(W\sqrt{T}/P)$
- Non-dimensional speed $(N/\sqrt{T})$
- Isentropic efficiency (η)

It is usual practice to refer (or correct) the parameters to standard inlet condition of 101.325 KPa and 288.15 K. Hence the pressure and temperature terms in these non-dimensional groups are replaced by δ and θ respectively and are otherwise known as referred or corrected parameter groups. Therefore,

$$\boldsymbol{\delta = P/101.325} \qquad \textbf{[Eq. 5-1]}$$

$$\boldsymbol{\theta = T/273.15} \qquad \textbf{[Eq. 5-2]}$$

Implementing compressor maps in a computer model of this nature is facilitated through the use of an auxiliary parameter known as Beta line. Beta lines are arbitrary lines drawn parallel to the surge line whose point of intersection with the speed lines serves as an array address (see Figure 5-1 below).

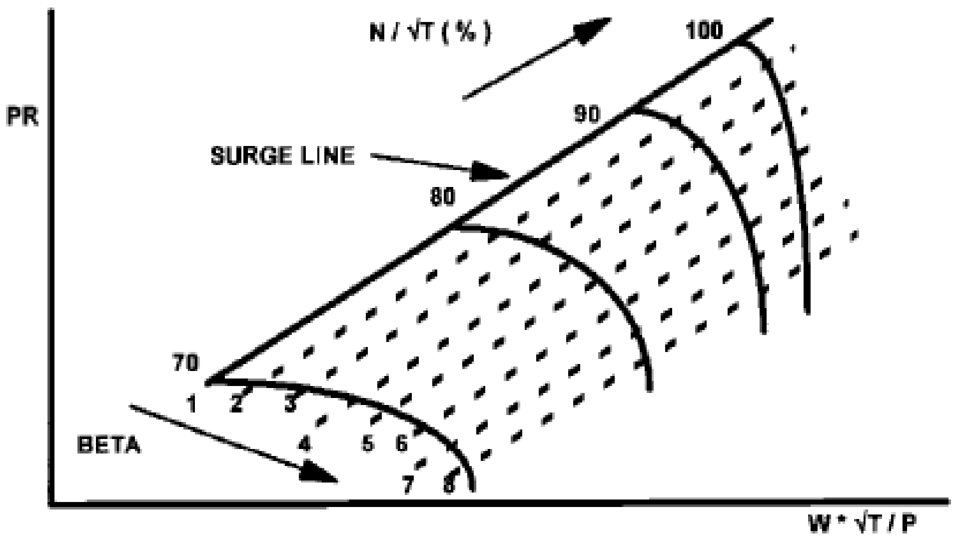

Figure 5-1 Compressor Map Fitted with Beta Lines

(Adapted from [44]**)**

In addition, the introduction of beta lines helps to overcome the problem associated with defining points at high values of non-dimensional speed for low values of PR. A feature of the compressor characteristic is that at such points, the speed lines become vertical. Thus for a given value of ($W\sqrt{T}/P$) and $N/\sqrt{T}$ there are several values of PR.

5.4.3 Compressor Characteristics Estimation

Compressor map data, like all other GT component data are highly proprietary to the manufacturers; which are not usually provided for sake of marketing competitions. However, at intervals of time, component characteristics may be published in the open literature. Similar to all other GT components, the GT compressor is designed to specific performance at the nominal design point of the engine. Adequate estimate of compressor data relative to the design condition is obtained as follows [50]:

$$\boldsymbol{PR_R = (PR - 1)/(PR_{DP} - 1)} \qquad \textbf{[Eq. 5-3]}$$

$$\boldsymbol{W_R = \left(\frac{W\sqrt{T}}{P}\right) \Big/ \left(\frac{W\sqrt{T}}{P}\right)_{DP}} \qquad \textbf{[Eq. 5-4]}$$

$$\boldsymbol{N_R = \left(\frac{N}{\sqrt{T}}\right) / \left(\frac{N}{\sqrt{T}}\right)_{DP}} \qquad \textbf{[Eq. 5-5]}$$

$$\boldsymbol{TR_R = (TR^{\epsilon} - 1)/(TR_{DP}^{\epsilon} - 1)} \qquad \textbf{[Eq. 5-6]}$$

where $\boldsymbol{\epsilon = \gamma/\gamma - 1}$ **[Eq. 5-7]**

(PR, W, N, and TR are the pressure ratio, mass flow, speed and temperature ratio respectively; TR is related to PR through the isentropic efficiency of the compressor. Subscripts DP and R refer to design and relative values).

5.4.4 Scaling Factors

The code has compressors maps representative of different technologies preloaded into it. Usually, the simulation require generation of component maps other than the default ones in the code depending on the engine been modelled. Although component characteristics are engine specific, the default compressor maps can be adapted to simulate other engines by scaling the corrected parameters using factors defined thus [106; 108]:

$$\boldsymbol{SFPR = \left(PR_{DP} - 1\right)/\left(PR_{DP\,Map} - 1\right)} \qquad \textbf{[Eq. 5-8]}$$

$$\boldsymbol{SFADW = \left(\frac{W\sqrt{\theta}}{\delta}\right)_{DP} \Big/ \left(\frac{W\sqrt{\theta}}{\delta}\right)_{DP\,Map}} \qquad \textbf{[Eq. 5-9]}$$

$$\boldsymbol{SFEFF = \eta_{DP}/\eta_{DP\,Map}} \qquad \textbf{[Eq. 5-10]}$$

($SFPR$, $SFADW$ and $SFEFF$ are the pressure ratio scale factor, corrected non-dimensional mass flow scale factor and efficiency scale factor respectively. Subscript DP is the design point values of scaled components and $DP\ Map$ is the design point map values of known component).

The Variflow code employs the above method to scale the compressor parameters during design point and off-design performance. However, it is desirable to keep the pressure ratio scale factor to as near unity as possible.

5.5 Developing the Subroutines to Implement Variable Stators

A major task in modifying any existing simulation code such as the Variflow code is the need to understand the algorithm of the main program and all subroutines used in building the code. The Variflow code is written in FORTRAN; hence all modifications were carried out in FORTRAN. However, it is worthy of note that the code as currently modified is in the FORTRAN free form source (FORTRAN 90) as opposed to the fixed form source (FORTRAN 70) of the existing one. This implies that non-conforming syntaxes were modified to suit the current format in the absence of which the code will fail to compile.

In the course of modifying the existing code, entirely new subroutines were created while some of the existing ones where amended where necessary.

5.5.1 Modification of the Compressor Map

The anchor of this modification is on the compressor map. The existing Variflow code works with five different in built compressor maps numbered one to five over a pressure ratio range of 1.7 to 11.0 respectively [104]. Each map is characterised by ten speed lines, with each speed lines defined by five points.

The current version of the code is equally fitted with five different compressor maps of variable geometry characteristics obtained from the Map Library of the Cranfield University Turbomatch 2.0. The range of pressure ratios for these maps (1 – 5) are shown in Table 5-1 below.

Table 5-1 Map Numbers and Corresponding Pressure ratios

Map Number	1	2	3	4	5
Maximum Pressure Ratio	2.0	4.5	7.0	11.0	15.0

Each map contains twenty relative speed lines with each speed line defined by twenty points' parameter co-ordinates for pressure ratio, corrected mass flow and efficiency. The compressor characteristics are defined for parameter values at stator vane angles of 0°, 10°, 20°, 30°, 40° and 50°. The compressor is assumed to be fully opened and fully closed at stator vane angles of zero degrees and 50° respectively. Figure 5-2 below depicts map no. 3 showing the twenty speed lines (6.5% - 130% relative speed).

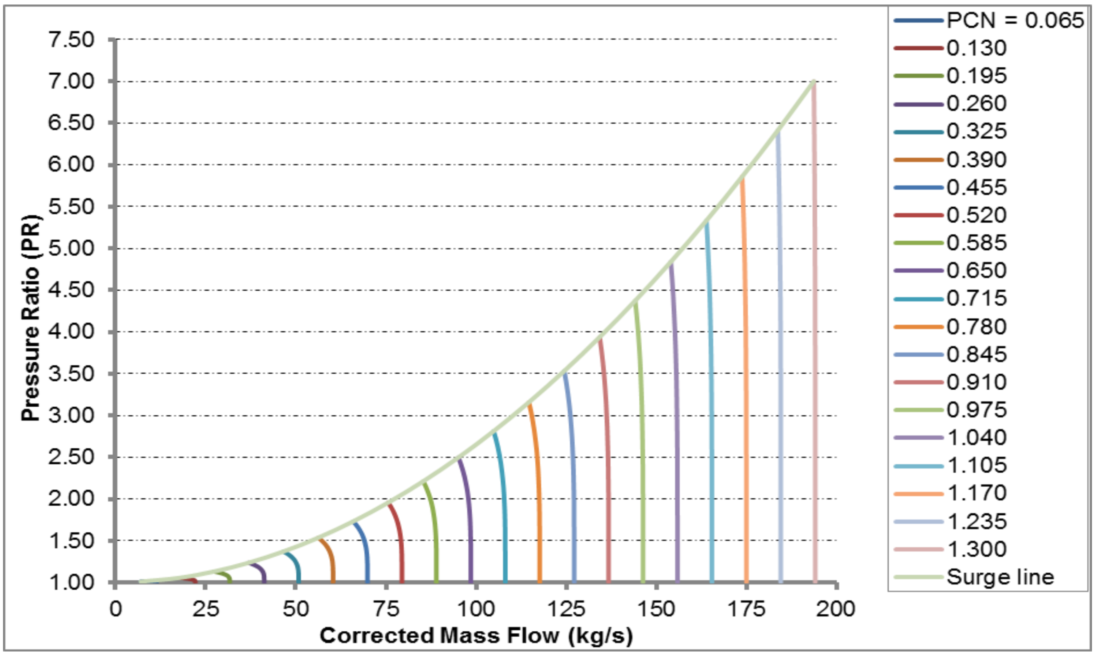

Figure 5-2 Compressor Map 3, stator angle 0°

Similarly, Figure 5-3 below shows the changing geometry of compressor map no. 3 for the different stator vane angle positions (the speed range shown is for 50% - 130% relative speed).

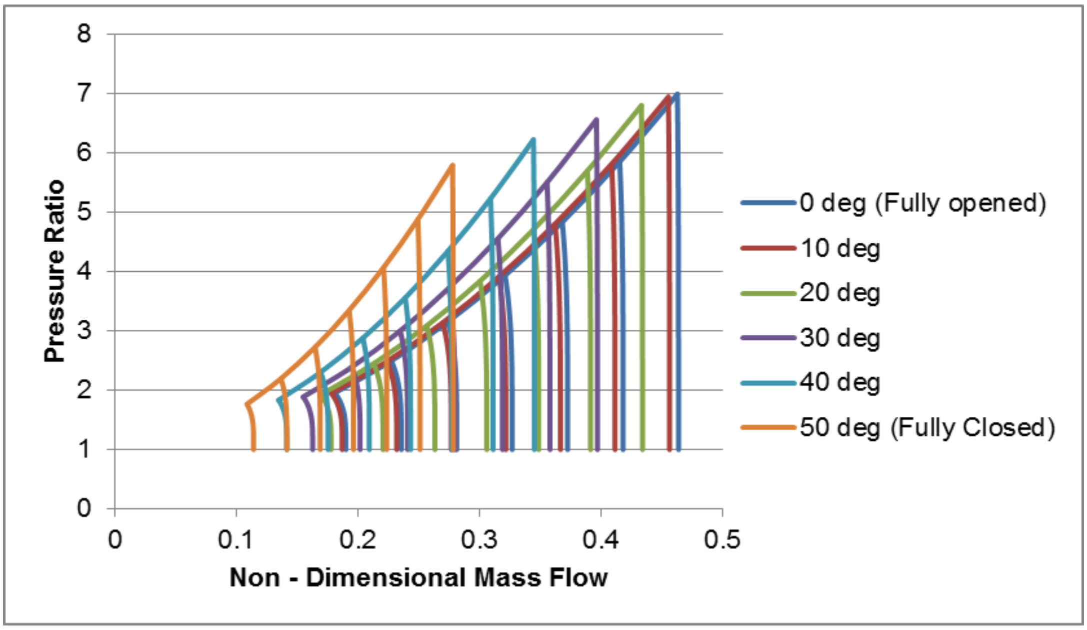

Figure 5-3 Compressor Map 3 characteristics with VSV angle

5.5.2 Implementing the New Compressor Maps

The parameters of the existing compressor map was implemented using a 10 x 5 two - dimensional array. Owing to the need to represent the six variable geometries of each compressor map as shown above, the current maps were implemented in the code using a three – dimensional array of size 20 x 20 x 6. In order to implement the new compressor maps the following tasks were executed:

a. Input data files for each of the compressor maps along with their variable geometries were created. Parameter values for the pressure ratio and non-dimensional flow were captured as elements for each speed line and replicated for the different variable stator vane angles. Hence each file contains six different parameter values for the six variable stator vane positions of 0°, 10°, 20°, 30°, 40° and 50°. The maps characteristics are known as COMPCHICVAR1, COMPCHICVAR2, COMPCHICVAR3, COMPCHICVAR4, and COMPCHICVAR5 respectively.
b. The existing subroutine "FILEREADER" that reads the map at the start of the code was modified to reflect the change in array size for each map. Similarly modified existing subroutines include "COMPMAP",

"CPRGUESSMAP", "JACOBIAN1S", "MATCHINGVALIS", "Comp_Evap_Converge", and "SCALEFACT".

c. In addition to the modification above, the entire algorithm of the subroutine "SCALEFACT" where the parameters scale factors are calculated was modified. It enabled the subroutine accept the user defined surge margin for a chosen map design point parameters from a newly created subroutine known as "POINT_LOCUS". The surge margin is defined by specifying the value of beta.
d. The subroutine "POINT_LOCUS" is used at the design point calculation to enable the user enter the compressor map with known values of speed and beta to obtain the corresponding map design point values of pressure ratio and non-dimensional mass flow. The beta lines is defined such that for a given speed line, it has a value of "1" at maximum pressure ratio and "0" at minimum pressure ratio.
e. The subroutine "VREAD" created to calculate the values of beta across the speed lines and convert the VSV angles into radians.
f. The subroutine "V_FILEREADER" is equally added to the code which functions like the existing "FILEREADER" subroutine but at off-design simulation. It combines the function of the "FILEREADER" subroutine with the ability to select compressor maps at other variable stator vane angles. The subroutine has capabilities of generating compressor maps for variable stator vane angles between the fully opened and fully closed positions.
g. Generation of compressor map parameters for user defined input of variable stator vane positions (angles between 0° and 50°), speed and beta values. This is accomplished through linear interpolation of the compressor map parameters taken advantage of the close intervals between the speed and beta lines.

5.5.3 Program Controls and Error Messages

In order to protect the code from crashing when put to use, the newly developed programs or subroutines exert some measure of checks on the user input. The subroutines employs a control that prevents beta from exceeding 0.999 and it also prevents the user from selecting variable stator vane angle other than 0° at design point simulation. Furthermore, at off-design the program restricts the user to select variable stator vane angle within the defined fully closed and fully opened positions. Similarly, restriction is placed on user input speed less than or equal to the minimum speed and greater than or equal to maximum speed of the compressor map. In all cases, the code will not run, instead an error message will be displayed that will enable the user fix the problem.

5.6 Validation of the Variflow Code Modification

In order to validate the modifications made, the running of the current code must give results that compares favourably with established trend when variable stators are used. In order to achieve this, the code was tested with the simulation of EL200 industrial GT inspired by the 7.68 MW (10,300 bhp) SGT200 GT engine. Using a fuel with lower heating value of 48.17 MJ/kg, the design point simulated output is shown in Appendix A. The design point parameters obtained from the public domain [109] in comparison with simulated result are shown in Table 5-2 below.

Table 5-2 Comparison of Simulated Design Point Performance Parameters with OEM

Performance Parameters	PR	EGT (K)	Efficiency (%)	Exhaust Gas Flow (kg/s)	Heat Rate (KJ/kwhr)	Power (MW)	TET(K)
OEM	12.6	762.59	33.50	29.50	10 740.92	7.68	-
Variflow	12.6	764.01	33.74	29.50	10 669.43	7.68	1326.15
% Error	-	-0.19	-0.72	-	+0.67	-	-

Having simulated this engine, off-design performance simulation at varying variable stator vane angles were performed and the results obtained are as shown in Figure 5-4 to Figure 5-10 below.

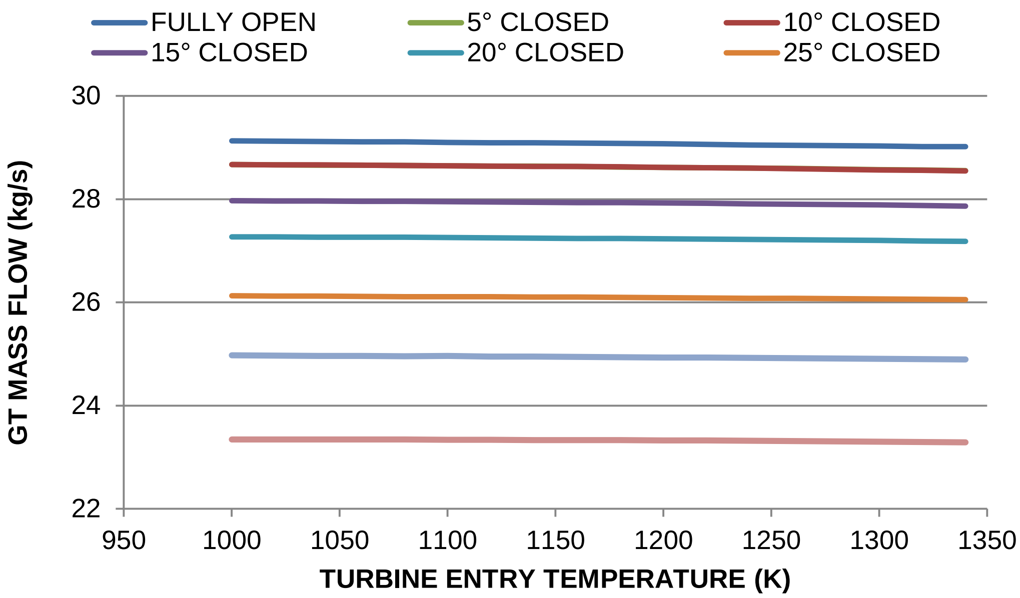

Figure 5-4 Variation of Gas Turbine Mass Flow for Varying Variable Stator Vane Position across Different Power Settings

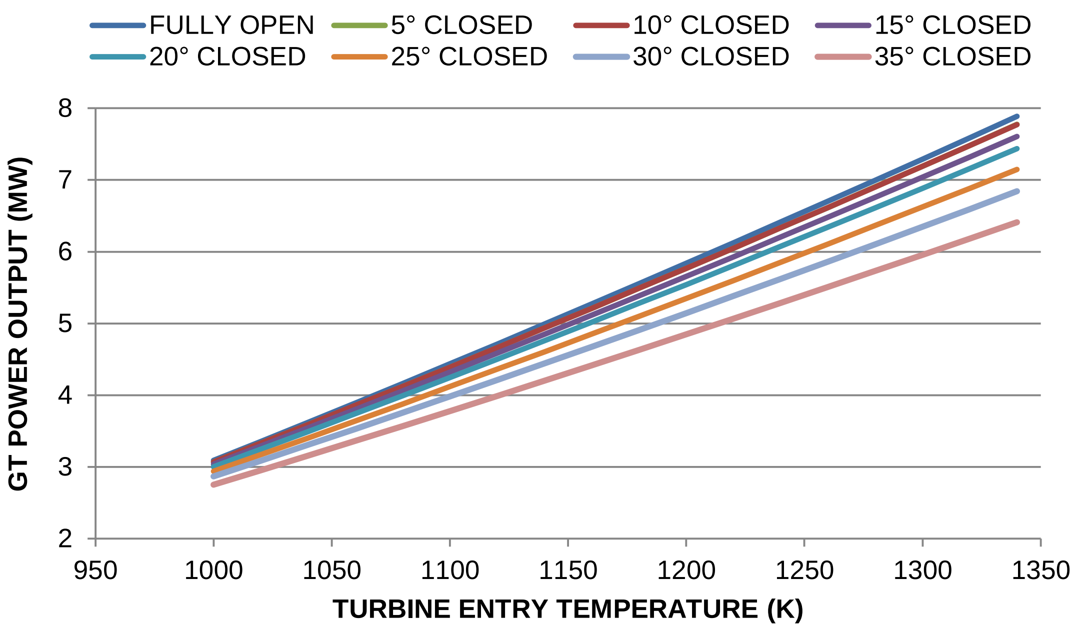

Figure 5-5 Variation of Gas Turbine Power Output at Different Power Settings for Varying Variable Stator Vane Position

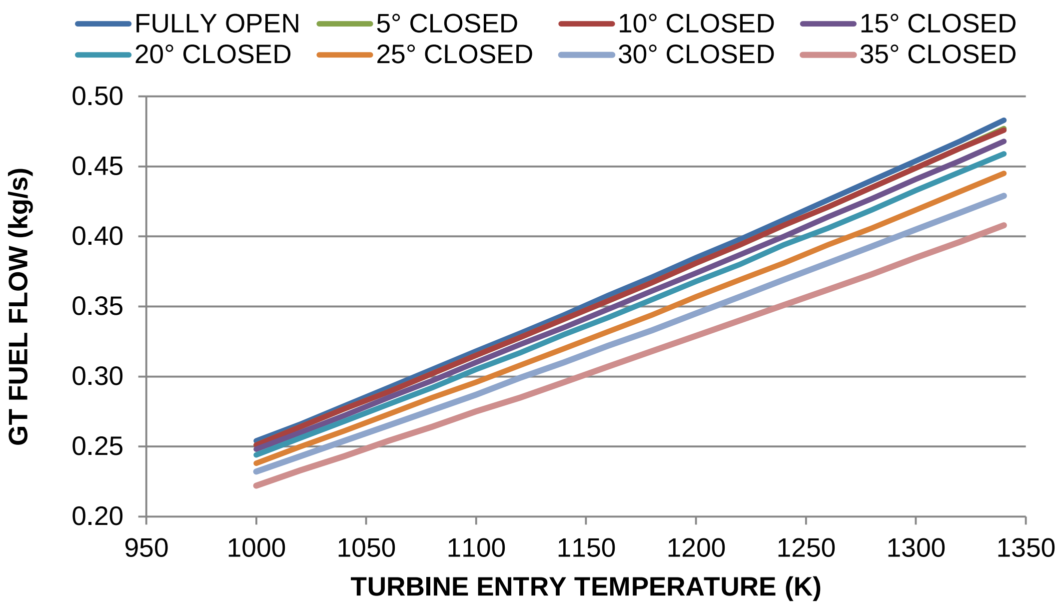

Figure 5-6 Variation of Gas Turbine Fuel Flow at Different Power Settings for Varying Variable Stator Vane Position

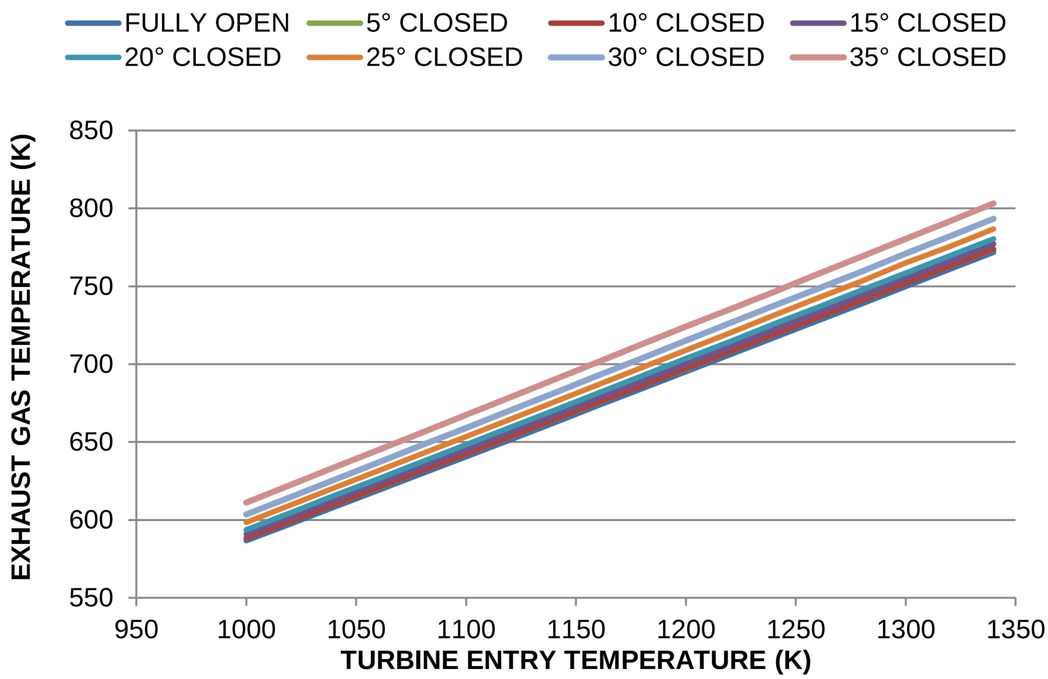

Figure 5-7 Variation of GT EGT at Different Power Settings for Varying Variable Stator Vane Position

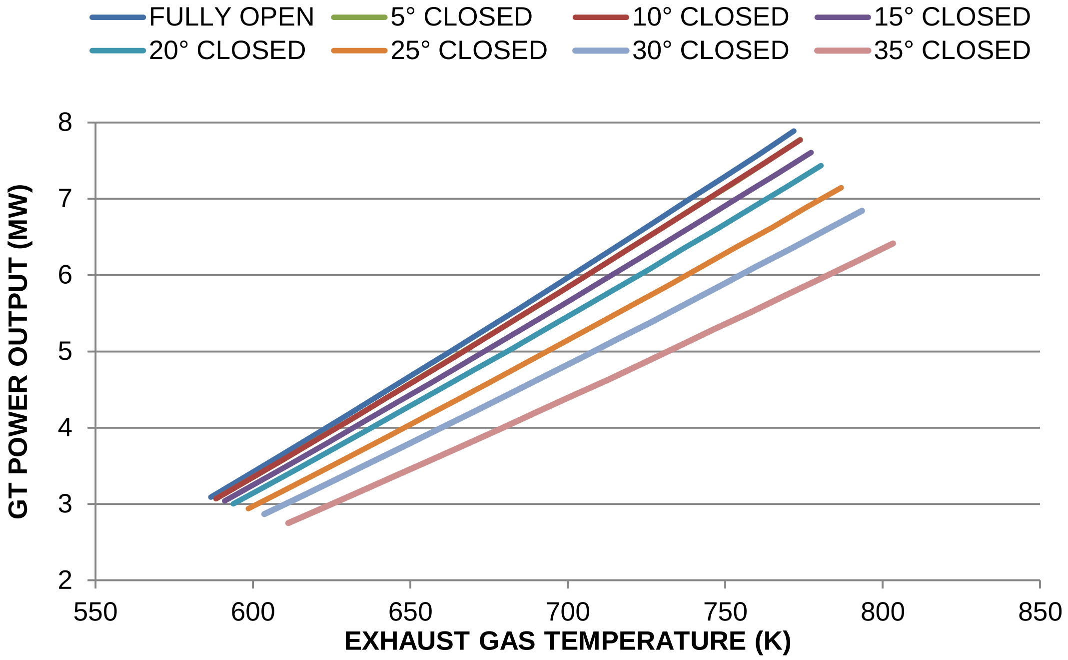

Figure 5-8 Comparison of Gas Turbine Power Output with EGT for Varying Variable Stator Vane Position

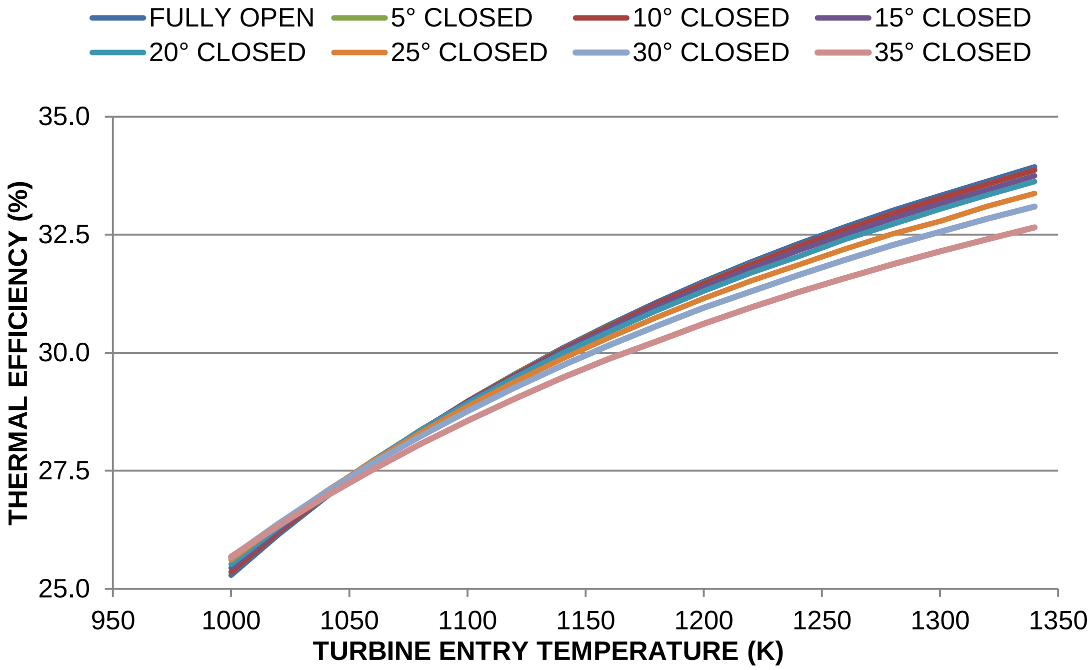

Figure 5-9 Variation of GT Thermal Efficiency at Different Power Settings for Varying Variable Stator Vane Position

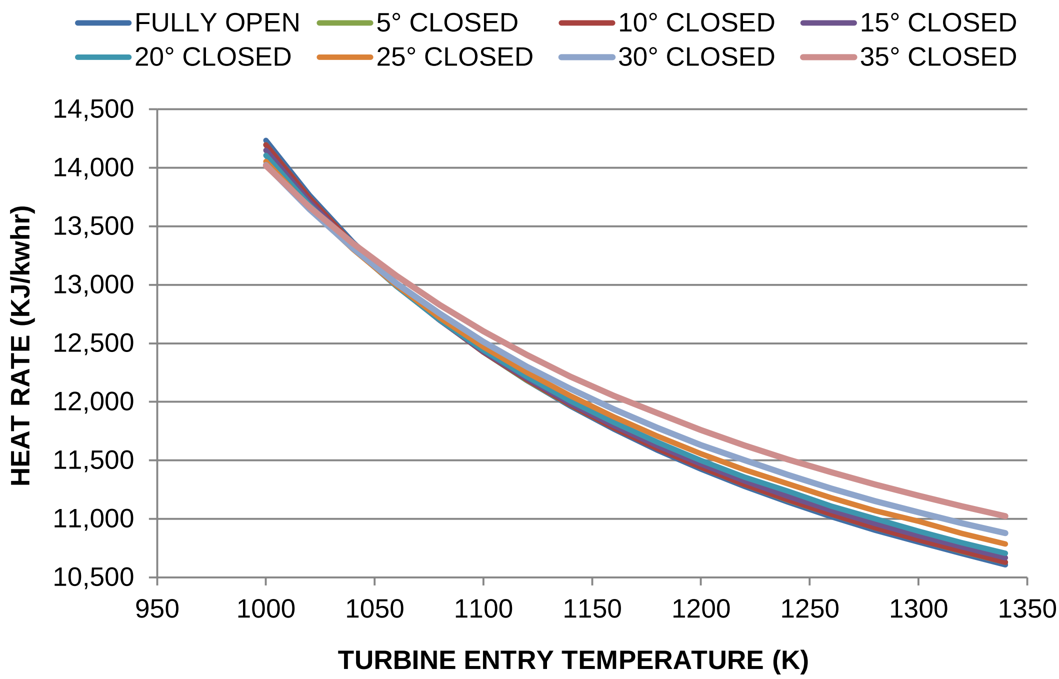

Figure 5-10 Variation of GT Heat Rate at Different Power Settings for Varying Variable Stator Vane Position

Figure 5-4 above depicts the displacements of the curves for the different stator vane position across the GT power settings. This is in conformity with expected trends since the application of variable stator vanes in single shaft GT is primarily to control the flow through the compressor. At fully open position the GT air swallowing capacity is at design mass flow of 29.03 kg/s and it reduces to about 23.34 kg/s with closing of the vanes at 35°. The plot also depicts a fairly constant non-dimensional mass flow with increase in power settings or pressure ratio. The plot equally shows the codes' ability to interpolate compressor map parameters for variable stator vane angles other than the six compressor maps currently implemented in the code.

The trending in the variation of power output, fuel flow and the exhaust gas temperature with the variable stator vanes position depicted by Figure 5-5, Figure 5-6 and Figure 5-7 above respectively is in agreement with the fact that flow compatibility dictates that any reduction in compressor flow (or non-dimensional flow) results in a decrease in compressor pressure ratio. Consequently, the GT power output, fuel flow and EGT will be reduced as shown. There is this advantage of maintaining a constant EGT in employing variable stators during part load performance of GT used in combined cycle application or co-generation. This trend is shown in Figure 5-8 above where the GT is conveniently run at part-load by changing the variable stator vane angles.

Figure 5-9 and Figure 5-10 above show similar trends with respect to the thermal efficiency and heat rate of the GT. GT thermal efficiency and heat rate are inversely proportional to one another, hence the opposing nature of their curves. However, two characteristic features in the figures are noteworthy. First, it is noted from the plots, that at very low power settings (1000 – 1045 K), the thermal efficiency or heat rate is best for the variable stator vane position at 35° closed. This is a benefit derived from using variable stator vanes for the overall improvement of surge margin at low power settings or low engine speed which especially useful in idle speed and decreasing starting power requirement of the GT [17; 36; 50; 47].

The second noticeable feature is the inflexion exhibited in the curves. This is a phenomenal movement of the compressor operating point towards surge as the GT power setting is increased or during acceleration to full power. This is said to happen with rise in firing temperature (hence turbine entry temperature) accompanied by increase in fuel flow before the increase in speed required to increase the mass flow [17]. This inflexion is often thought to occur in high performance axial compressors when they operate with several of the early stages stalled.

Finally, mention must be made that the code is unable to converge at extreme closing of the variable stators unless the design point temperature is increased which is an obvious limitation in this modification.

5.7 Concluding Remarks

The foregoing has demonstrated the ability of the code to predict the effect of variable stators especially for a single shaft GT within the limits of the implemented program. Furthermore, the highlighted limitations give room for future improvement on the code. The development of subroutines to implement variable stators in this code is one of the contribution of this work. The next chapter presents the main outcome of implementing the TERA framework in analysing the use of GT prime movers in pipeline transportation of CO_2 for sequestration.

6 Gas Turbine Mechanical Drive Performance and Economics

6.1 Introduction

The thrust of this work is on the technical, economic and environmental risk assessment of gas turbine (GT) in application to CO_2 compression for pipeline transmission. In doing this, the methodology described in chapter 3 will be employed in two different case scenarios analyses. The case scenarios are predicated on the demand to meet the CO_2 compression duty of the GT driven equipment at two points. Firstly, to compress the captured CO_2 (at source) assumed to be at atmospheric pressure to pipeline operating pressure of 150 bar using a combination of compressor and pump; and secondly for recompression along a trunk pipeline which is assumed to convey the CO_2 to the point of sequestration.

The analyses were facilitated by carrying out design and off-design point simulation of selected GT mechanical drives to meet the yearly compression duty for CO_2 throughput spread across four (4) seasons within the year against a projected plant life of twenty-five (25) years. Then the economic performance as well as the associated risk of deploying the GT drives were assessed using an economic model based on the net present method (NPV) and payback period (PBP) developed by the author. A code was also developed to carry out the hydraulic analysis of the CO_2 flow in the pipeline to determine the point of recompression upon the pressure dropping to a set minimum value.

6.2 Case Scenario Description

The case scenarios are built across an adaptation of the 2040 electricity generation projection from the United States' Annual Energy Outlook 2014 shown in Figure 6-1 below. The outlook shows a decline in coal fired power plants and an increase in natural - gas fired power generation plants due to concerns over emission of GHG. Similarly, it takes into account the improvement in harnessing renewable energy (e.g. solar) for power generation. With the nuclear power plant out of the scenario, the trend shown in the forecast can easily be adapted to suit the Nigerian electricity generation projection. Nigeria is signatory to the Kyoto Protocol and pursues the vision of GHG abatement through the adoption of clean development mechanism (CDM). Nigeria also has abundant sunshine, oil and gas as well as a fair share of coal. Furthermore, Nigeria is a developing economy with a growing population which translates into increasing demand in electricity. Against these facts, the projection per se could well represent a typical forecast for Nigeria. Two case scenarios were considered and explained below.

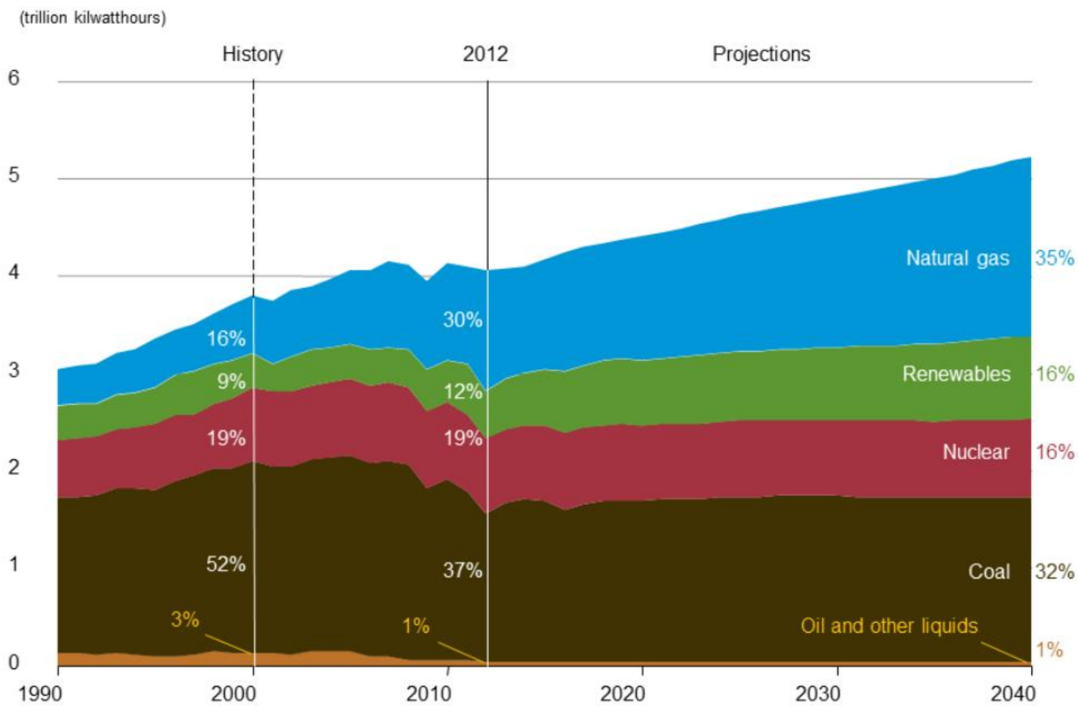

Figure 6-1 Electricity Generation by Fuel, 1990 – 2040

Source: US EIA

6.2.1 Case Scenario I

This case scenario deals with the GT providing power to a turbo-compressor for initial compression of CO_2 captured from power generation plants from atmospheric pressure to 100 bar. The control employed to meet the output pressure requirement and the ability of the turbo-compressor to meet the changing capacity of the CO_2 being compressed will impart greatly on the off-design performance of the selected GT mechanical drives being employed. Thus, imparting on the economic performance of the investment.

The amount of CO_2 generated by the power plant is dependent on the type of power plant [1]. Similarly, the amount of captured CO_2 available for compression is dictated by the power plants' load swing since demand varies on an hourly, daily, monthly or seasonal basis. In addition to the influence of ambient conditions (especially temperature) on the off-design performance of the GT mechanical drive it is increasingly important at the very least to evaluate all season conditions for techno-economic assessment of this kind. Hence, with the site location in mind, this case scenario was analysed considering CO_2 captured from four types of power generating plants at four different seasons experienced in Nigeria within a year shown in Figure 6-2 to Figure 6-5 below. The four different seasons are:

- Hot Season (March – May)
- Early Rain Season (June- August)
- Late Rain Season (September - November)
- Harmattan Season (December- February)

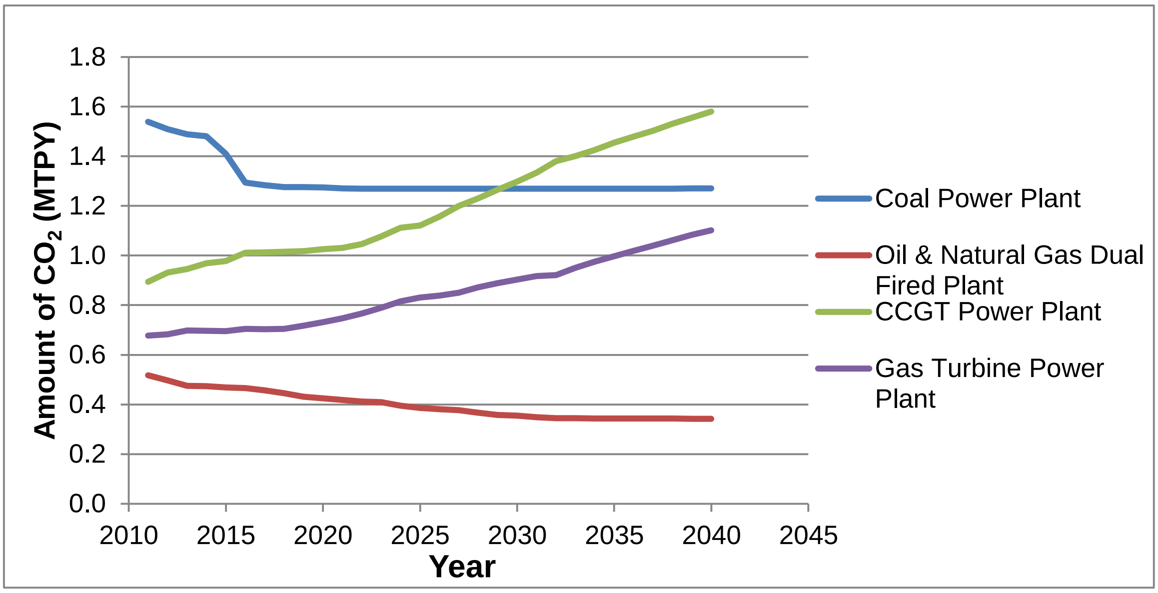

Figure 6-2 2040 Projected Average CO_2 Captured from Four Different Fossil Fired Power Plants for the Hot Season (Max. Temperature = $38°C$)

Source: [116]

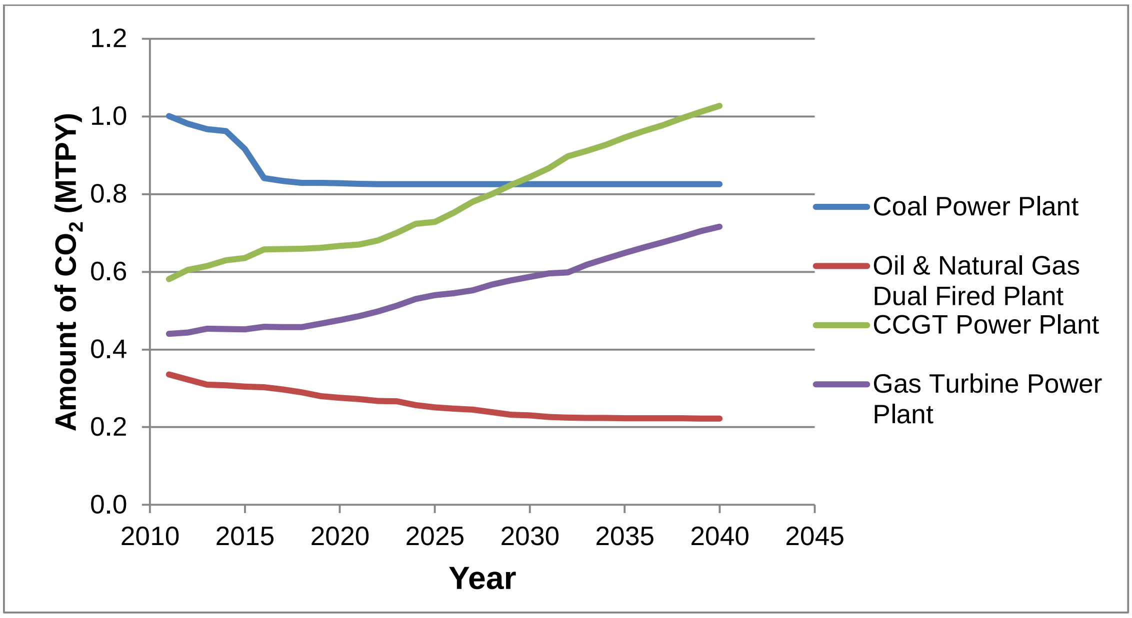

Figure 6-3 2040 Projected Average CO_2 Captured from Four Different Fossil Fired Power Plants for the Early Rain Season (Max. Temperature = $34°C$)

Source: [116]

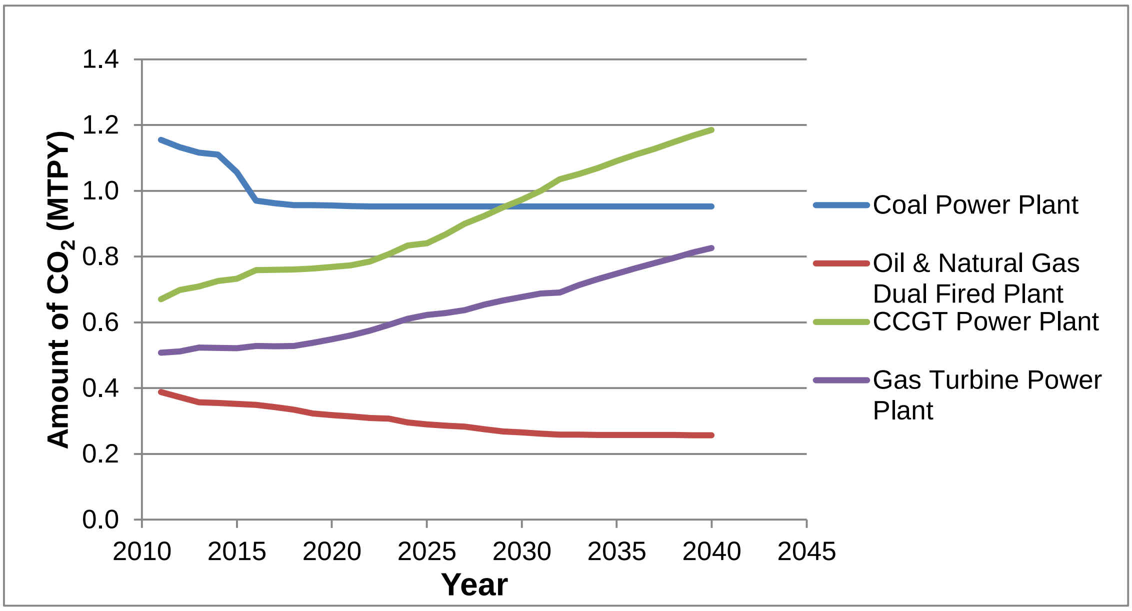

Figure 6-4 2040 Projected Average CO_2 Captured from Four Different Fossil Fired Power Plants for the Late Rain Season (Max. Temperature = $34°C$)

Source: [116]

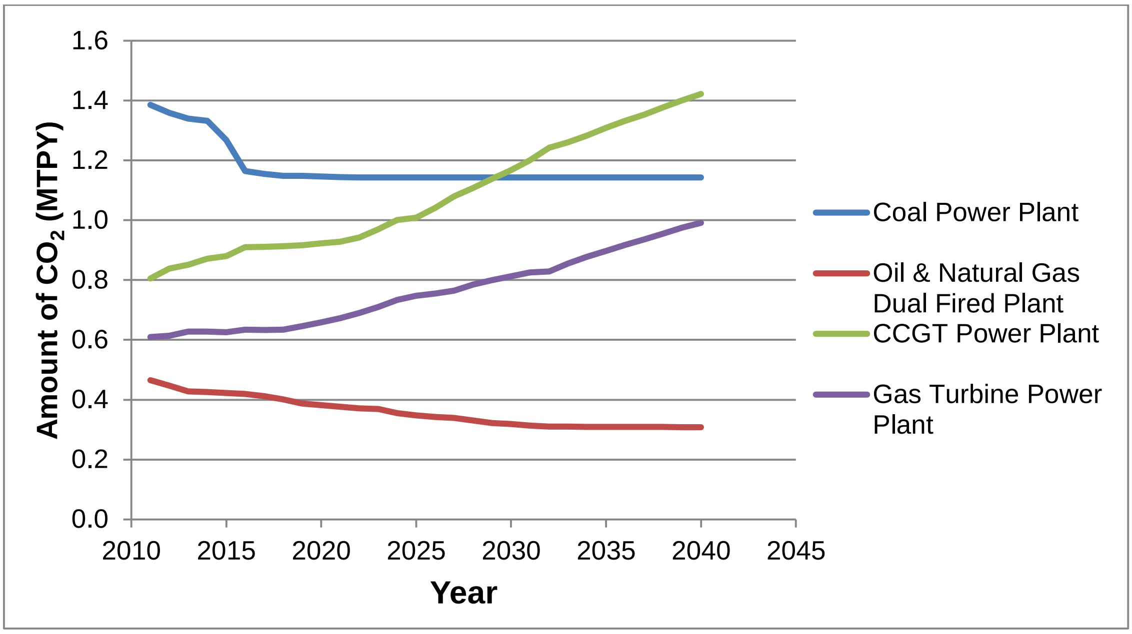

Figure 6-5 2040 Projected Average CO_2 Captured from Four Different Fossil Fired Power Plants for the Harmattan Season (Max. Temperature = $33°C$)

Source: [116]

6.2.2 Case Scenario II

This case scenario involves recompression of CO_2 transmitted along the pipeline profile (Figure 3-2, page 45) in a booster station. Here the GT driven equipment is a CO_2 centrifugal pump. The profile is a buried onshore 511 km pipeline laid across a terrain of about 67 m to 180 m elevation above sea level. The factors affecting the temperature profile for buried pipeline include the pipe's material thermal conductivity, insulation, the soil temperature, overall heat transfer coefficient and the fluid temperature. However, a simplified assumption of isothermal flow and a constant CO_2 flowing temperature of $25\ °C$ ($77\ °F$) were considered. Since the pipeline operating pressure is fixed at 150 bar due to economic consideration as specified in literature; the need for recompression or otherwise will be determined by evaluating the pressure drop along the pipeline profile through hydraulic analysis of a given pipe diameter and CO_2 throughput.

6.3 Gas Turbine Operating Condition

The primary use of the GT prime mover is to transmit torque to the driven equipment. The available GT power and efficiency is majorly impacted by the prevailing ambient condition among other factors. The variation of site specific ambient pressure and temperature are factors that must be considered on both performance and operation. The altitude at which the GT is being operated affects the performance of the GT because it influences sites' ambient pressure and temperature. Ambient temperature decrease of 1.98 $°C$/1000 ft. (304.8 m) is established by the International standard Atmospheres [17]. Given the maximum pipeline profile elevation of about 180 m for this study, temperature change is not a concern. However, the reduction in atmospheric pressure with altitude is a thing of concern since it affects the pressure ratio across the power turbine and hence power output.

It is an established fact that GT performance is better in cold days compared to hotter days [17; 36; 44]. Therefore, variation in site temperature must be considered. In the current study, the operating ambient temperature is for a site located in Kano – a city in the Northern part of Nigeria shown in Figure 6-6 below. Typically, the temperature is highest (about 38 $°C$) during the months of March – May and lowest (about 14$°C$) between December - January. Like most tropical countries, the electricity demand is highest during the hottest months

Figure 6-6 Monthly Average Temperature Variation at the Operational Site [110]

due to massive air-conditioning and refrigeration requirements; hence most power plants are operated at their maximum output which means more CO_2 is expected to be captured from the power plants. This fact is reflected in the relative higher CO_2 flow rate shown in Figure 6-2 above.

A major economic concern about the GT operating condition is the fuel consumption, which varies depending on whether the operating conditions impacts negatively or positively on the thermal efficiency. The concern during operation is always aimed at achieving low fuel consumption for a given compression duty. In this regard, it must be stressed that the performance of the GT and the driven equipment must be such that enables the best overall efficiency. Thus the main goal of the close coupling of the compression system with the GT is to achieve the highest overall package efficiency to enhance lower fuel consumption.

6.4 Gas Turbine Design and Off-Design Simulation

GT performance prediction to establish the operating parameters for a desired power output is pertinent to this analysis. This is especially useful in predicting the fuel consumption which is a major contributor to the overall operating cost (OPEX). Once the compression power requirement is established, the GT mechanical drive is selected and simulated to extract necessary performance data (fuel flow, thermal efficiency & power output) for the techno-economic assessment. As earlier highlighted in the methodology, the GT engine simulation carried out in this study were accomplished using in-house GT simulation code - Turbomatch developed in Cranfield University.

The design point performances were inspired by selected GT from the open domain that meet the required compression power. In order to provide the varying compression duty, selected GTs were de-rated using the maximum site ambient temperature. Mention must be made though that GT can produce far more power at colder ambient temperatures; so design based on worst case ambient conditions may not be optimal in some situation. However, considering the fact that more volume of CO_2 is compressed during the hottest period of the

GT operation, the de-rated power output used for the GT selection may be within optimal design.

Having established the design point, the effect of different compression duty requirement and varying ambient condition on the GT parameters were simulated to provide the off-design performance.

6.4.1 Selected Gas Turbine Design Point Performance

In this particular case scenario, two GT mechanical drives were simulated namely: 33.9 MW simple cycle DLE GT, EL2500RD, inspired by GE LM2500RD and 9.4 MW advanced cycle GT (recuperated) EL1200-R inspired by THM1304-10R [109]. The choice of these two GTs stem from two major considerations. First, the required operational flexibility: in that the power generation plant with the lower CO_2 emission or throughput will be catered for using the 9.4 MW EL1200-R GT while the higher CO_2 throughput will be driven by the 33.9 MW EL2500RD GT. Second: the recuperated cycle is carefully chosen to take advantage of the improved performance (fuel saving) associated with the use of heat exchanger in small power plants where the pressure ratio is low. The heat exchanger essentially enables the pre-heating of the GT compressed air with the exhaust gas before entering into the combustion chamber. Also, the choice of the dry low emission (DLE) GT was informed by environmental concern for lowest available emission rate (LAER).

The Turbomatch input files are shown in Appendix B while the simulated design point performance parameters in comparison with the OEM [109] are presented respectively in Table 6-1 and Table 6-2 below. The fuel lower heat value is 43 MJ/kg.

Table 6-1 Simulated Design Point Performance Parameters for EL2500RD Compared with OEM

Performance Parameters	PR	EGT (K)	Thermal Efficiency (%)	Mass Flow (kg/s)	Power (MW)	TET(K)
OEM	23.0	798.2	39.7	91.2	33.9	-
Simulation	23.0	797.2	39.6	91.2	33.9	1550.0
% Error	-	-0.1	-0.3	-	-	-

Table 6-2 Simulated Design Point Performance Parameters for EL1200-R Compared with OEM

Performance Parameters	PR	EGT (K)	Thermal Efficiency (%)	Mass Flow (kg/s)	Power (MW)	TET(K)
OEM	10.0	-	36.3	45.4	9.4	-
Simulation	10.0	748.71	36.3	45.4	9.4	1200.0
% Error	-	-	-	-	-	-

6.4.2 Off Design Performance of EL2500RD and EL1200-R

The selected GT mechanical drives should effectively cope with the changing demand of the compression equipment. Assuming the turbo-compressor are operated at best efficiencies, in order to assess the ability of the prime movers in providing the required compression duty, the influence of changing ambient temperature and different power settings on the performance of the GTs (EL2500RD and EL1200-R) were simulated. In so doing, the site (profile) altitude of 100m at the point of initial compression was put into consideration. Performance simulations at ambient temperatures of 5°C, 10°C, 15°C, 20°C, 25°C, 30°C, 35°C and 40°C represented by deviation from the design point temperature (15°C) - ISA DEV, as “ISA DEV -10”, “ISA DEV -5", “ISA DEV 0”, “ISA DEV 5”, “ISA DEV 10”, “ISA DEV 15”, “ISA DEV 20”, and “ISA DEV 25” respectively were conducted. The simulation for the maximum site temperature of 38°C (ISA DEV 23) is also included. The turbomatch input files are appended in Appendix C while the performances are presented below.

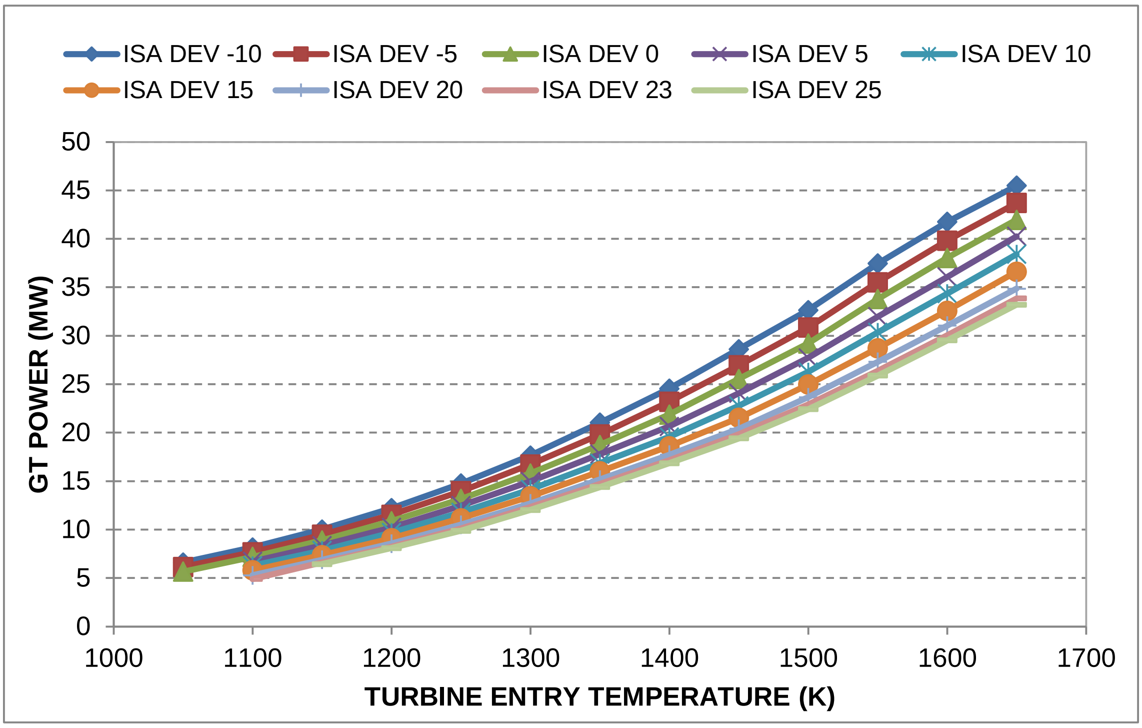

Figure 6-7 Variation of EL2500RD GT Power Output with Change in Ambient Temperature at Different Power Settings

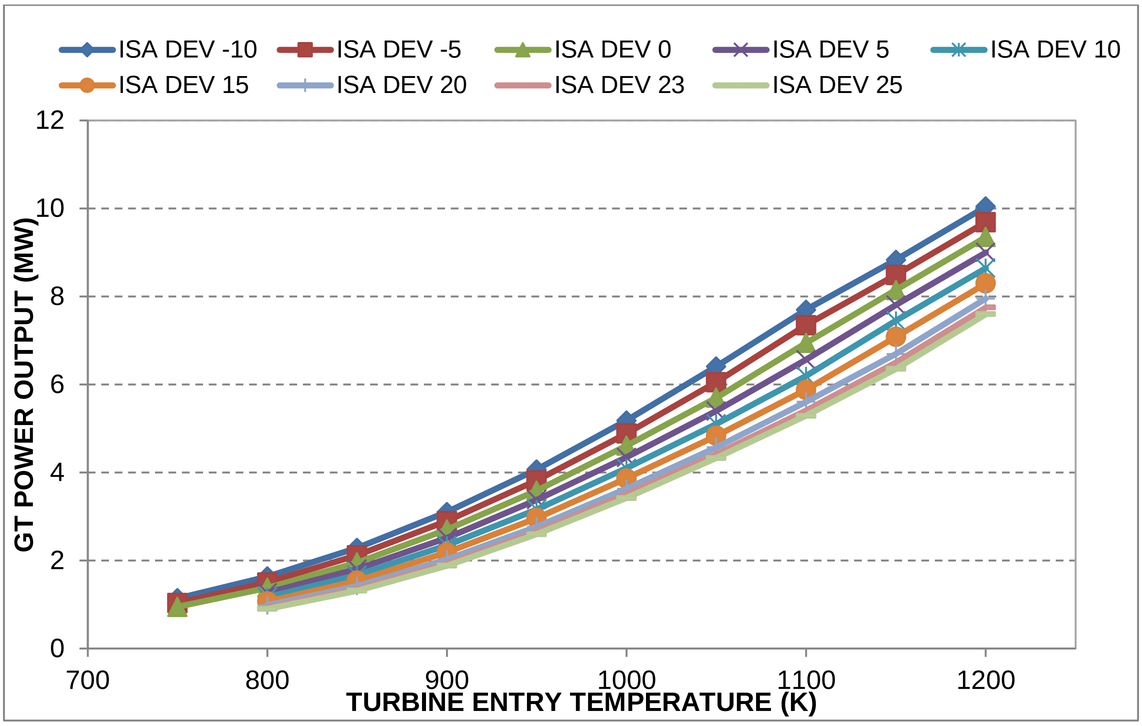

Figure 6-8 Variation of EL1200-R GT Power Output with Change in Ambient Temperature at Different Power Settings

Figure 6-7 and Figure 6-8 above depicts the variation of the GT power output for different power settings or TETs as the ambient temperature changes for the simple cycle GT (EL2500RD) and the advanced cycle GT (EL1200-R) respectively. As expected, the power output increase with increasing TET and decreasing ambient temperature. In GT engines, different power settings or TETs are usually accomplished by the control of fuel flow to the combustor. This is depicted by the similar trends obtained in Figure 6-9 and Figure 6-10 below for EL2500RD and EL1200-R respectively for the fuel flow at different power settings with changing ambient temperatures.

The plot in Figure 6-7 shows the possibility of firing the GT above its design firing temperature for increased power output. While this could be achievable within limits, it is highly detrimental to the creep life of the turbine blade, hence should be avoided.

The GT power output is limited by the firing temperature or TET and the maximum gas generator speed. With increase in fuel flow, both the TET and gas generator speed increase, until one of the two operating limits is reached. At the design point temperature (ISA), both limits are attained at the same time. However, at ambient temperature below ISA the speed limit is reached first while at ambient temperatures above ISA the TET becomes the limiting factor. Thus, although at ambient temperature below ISA the power output increases; such increase is constrained by the speed limit. Similarly, at ambient temperature above ISA it is noteworthy from Figure 6-7 and Figure 6-8 above that there is a minimum firing temperature below which the GT is unable to give power output. This phenomenal occurrence is due to a minimum speed limit constraint which unless attained the GT will not give a useful power output thereby fixing the minimum firing temperature or TET.

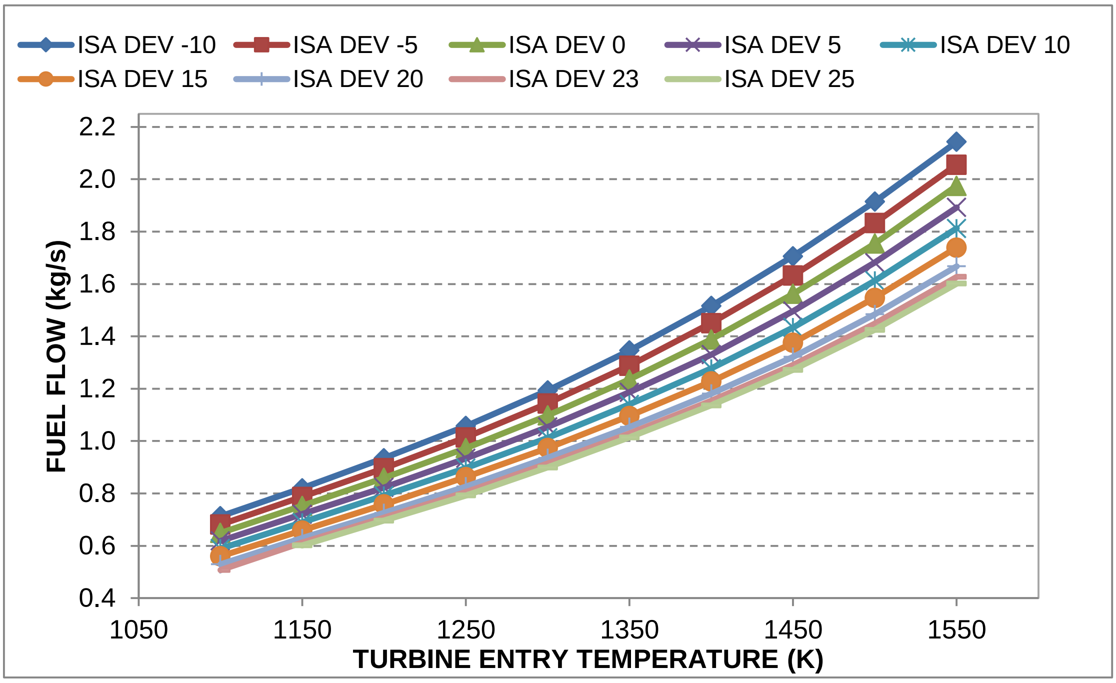

Figure 6-9 Variation of EL2500RD GT Fuel Flow with Change in Ambient Temperature at Different Power Settings

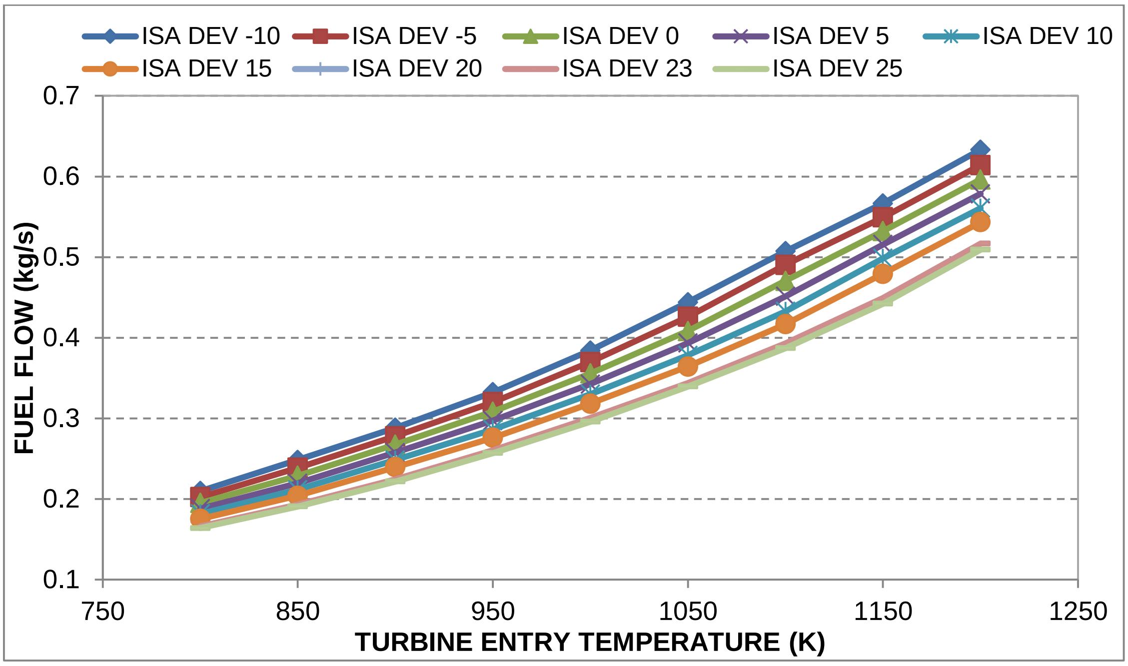

Figure 6-10 Variation of EL1200-R Fuel Flow with Change in Ambient Temperature at Different Power Settings

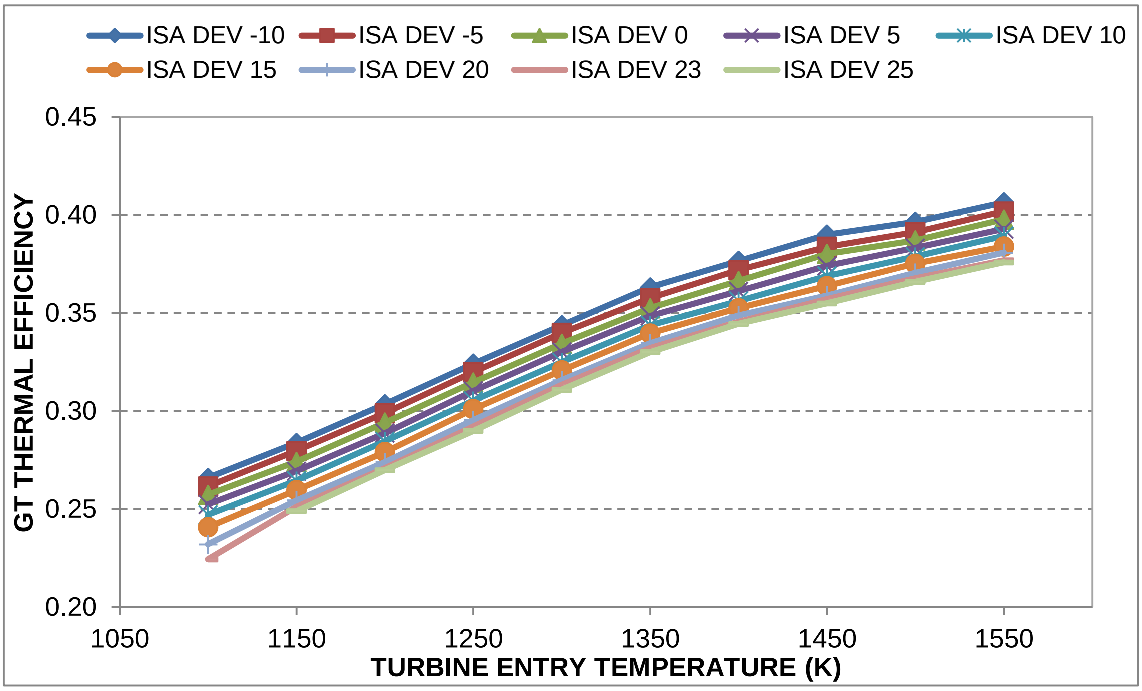

Figure 6-11 Variation of EL2500RD GT Thermal Efficiency with Change in Ambient Temperature at Different Power Settings

[Figure 6-11 - Figure 6-12] and [Figure 6-13 - Figure 6-14] show the thermal efficiencies and heat rates for the designated GT mechanical drives respectively. At ambient temperatures below the design point (ISA) the thermal efficiencies are on the increase because less compression work is required by the gas generator compressor due to the reduced inlet air temperature, hence much of the energy of combustion will be converted into useful work or power output. Similarly with increasing TET, there is increase in heat input and so the thermal efficiency will obviously increase.

The trend depicted by the plots for the heat rate is such that increase in power output (increasing TET or decreasing ambient temperature) causes a reduction in the heat rate. This is expected as heat rate is inversely proportional to thermal efficiency.

$$\boldsymbol{Heat\ Rate = 3600/Thermal\ Efficiency} \qquad \textbf{[Eq. 6-1]}$$

The concept of heat rate is usually quoted by GT manufacturers as a measure of efficiency because it enables the evaluation of fuel cost directly since fuel prices are usually quoted in £/MJ or $/MJ. However, it is worth noting that inefficiencies in the GT driven equipment will result to more power demand from the GT which implies more fuel consumption. Hence for economic evaluation, the end user should ignore the heat rate and deal with the direct value GT fuel consumption.

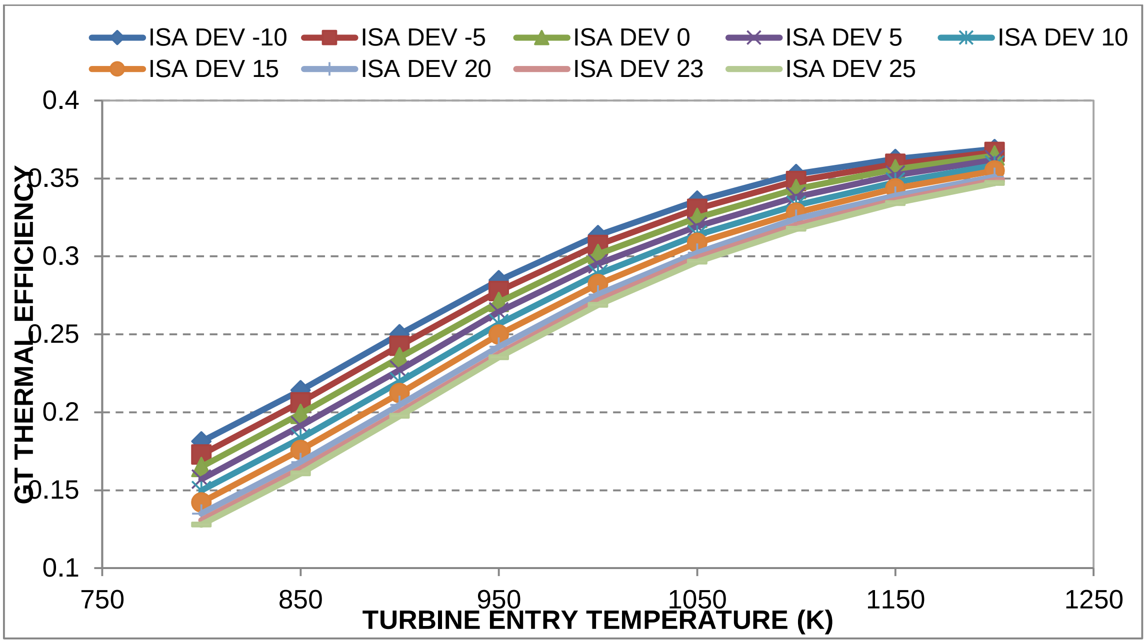

Figure 6-12 Variation of EL1200-R GT Thermal Efficiency with Change in Ambient Temperature at Different Power Settings

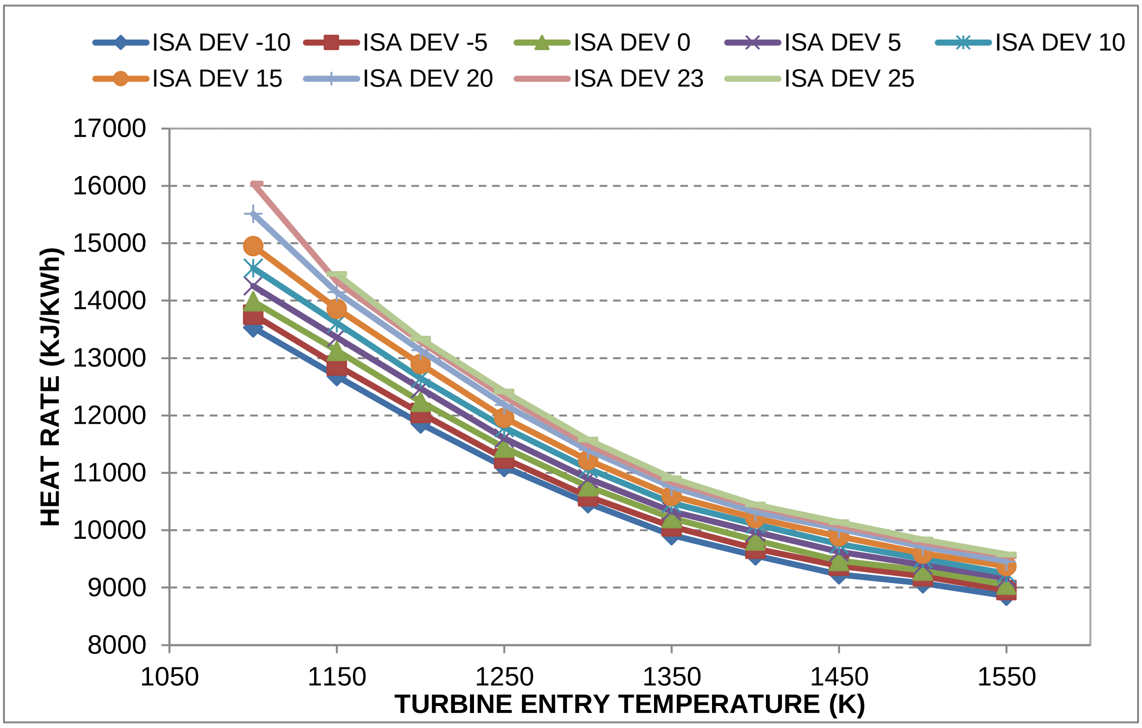

Figure 6-13 Variation of EL2500RD GT Heat Rate with Change in Ambient Temperature at Different Power Settings

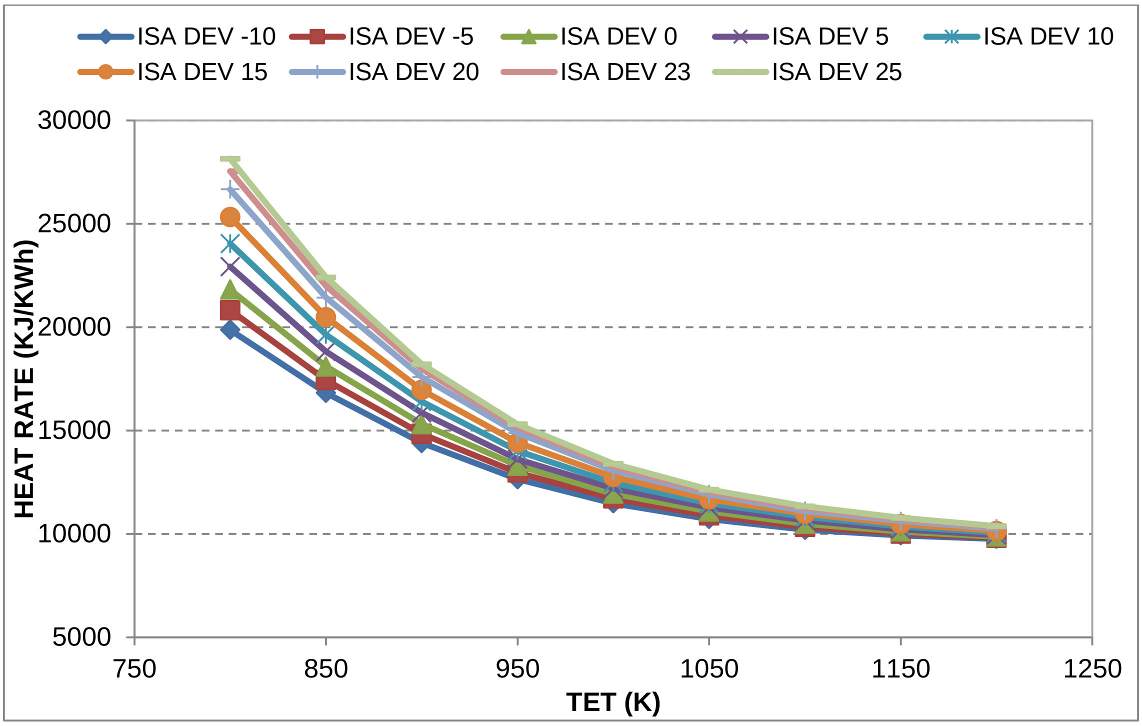

Figure 6-14 Variation of EL1200-R GT Heat Rate with Change in Ambient Temperature at Different Power Settings

6.4.3 Summary

The trends of the selected GTs performances are in agreement with established results during operation. Having established the expected performance of the GT mechanical drive, the next step in the techno-economic and environmental assessment is to apply this in the case scenarios economic evaluation. Therefore, the simulated performances will now be narrowed down to the average site operating ambient temperature of $38°C$ to extract all necessary parameters for the required analysis.

6.5 Gas Turbine Economic Performance Evaluation

The investment in GT mechanical drive for CO_2 compression comes at a cost. The cost associated with the deployment of GT prime mover will consist of the capital cost of the GT (initial and installation costs) and cost incurred throughout the entire service life of the plant. Therefore, like any other investment, this must be driven by economic returns. It is a statement of fact that the future is hard to predict, however, some predictions and scenario studies could be useful in risk reduction of the investment. In this light, a discounting technique based on NPV method is presented for the initial assessment of the economic performance of GT prime movers for CO_2 transmission in pipeline. All cost data were converted to 2013 British Pound Sterling value using UK GDP deflator [111] and annual currency exchange rate [112].

6.5.1 Capital Cost Estimate

The capital cost varies with the capacity of GT and can be obtained from references [27; 113] which are updated annually. Where the price of the engine is not quoted directly, the cost is scaled to the capacity of interest using the generic scaling relation [114]:

$$\boldsymbol{Cost\ C = Cost\ C_0 * \left({}^{S}/_{S_0}\right)^{f}} \quad \textbf{[Eq. 6-2]}$$

Where cost of a component, $Cost\ C$ of size S is related to the cost of a reference component, $Cost\ C_0$ of size S_0 by means of a scaling factor f. Installation cost could be estimated at around 10% of the initial capital cost [115].

6.5.2 Operation and Maintenance Cost Estimate

During the service life of the GT prime mover, the operation and maintenance (O & M) cost comprises of fixed and variable charges. O & M cost to some extent depends on the decision taken during the design and manufacturing phase of the equipment. Likewise, personnel cost which contributes to the O & M costs depends on the equipment size and degree of automation. Maintenance cost depends on such factors as the technology of the GT prime mover, operating environment, operation cycle and type of fuel. Insurance covers for equipment failure, loss of income, loss of savings and business interruption also add to the operation costs. The cost of insurance could vary depending on the GT performance history, system design and operating mode. It may be within the range of 0.25 – 2% of the initial capital cost while it could be covered by the operator’s overall insurance program at no additional cost.

Among the variable cost is the unit - fuel cost which is usually accounted separately because it contributes the largest cost item (fuel contributes about 70% of the total O & M cost). The O & M cost in essence depends a lot on the price of fuel which in turn is subject to the global price of oil. Consequently, oil price is one of the important factors that determine the profitability of deploying the GT mechanical drives. These prices are influenced by the laws of demand-supply, global political scene and the stock market deals. In specific terms, the amount of fuel consumed is determined by the efficiency and operating range of the GT prime mover subject to the driven equipment's performance.

In the absence of any available data, the fixed and variable (except fuel) O & M cost is usually estimated as percentage of the capital cost. However, in this study, the O & M costs estimates are 2012 figures quoted from the updated capital cost estimates for utility scale electricity generating plant by the US Energy Information Administration [116]. Similarly, despite the fact that fuel price is dynamic, a simplistic approach of assuming a fixed price in the analysis was adopted.

6.5.3 Other Cost Estimate

The total capital invested in line with the Electric Power Research Institute (EPRI) cost estimating guidelines has in addition to the above, annual taxes paid on emission (CO_2 levy), income and credit as well as depreciation cost [117].

In this study, the emission tax is represented by carbon tax assumed to be proportional to the mass of CO_2 emission. The income tax is assumed while a straight-line method is assumed for the depreciation cost estimate. The credit tax is assumed to be zero.

6.5.4 Revenue and Economic Performance Modelling

The revenue is the market value of the power consumed for the CO_2 compression. In order words, the power generated by the GT mechanical drive is assumed to be sold at current electricity price. In this work, the electricity price is carefully selected having considered industrial electricity prices published in the 2014 quarterly energy price in the UK [118].

The economic performance analyses were simulated using a FORTRAN based computer code developed by the author shown in Appendix D. The code input files are simulated compression power requirements and corresponding turbomatch or Variflow simulated fuel flow for the power required (typical input file also shown in Appendix D). The input file parameters (GT power and fuel flow) were implemented in the code using a three – dimensional array of size 4 x 25 x 2. Here the “4” represents the four seasons highlighted above; “25” represents the number of years or assumed plant life; and “2” represents the input parameters. Thus the GT power is defined by 4 x 25 x 1 while the corresponding fuel flow is defined by 4 x 25 x 2. Each power generation plant has a separate input file that is read into the program during simulation. The economic performance was modelled using the following calculation steps [114; 119]:

If I is the Capital Cost of the investment and subscript t is the year count $(t = 1,2, \ldots., n)$

i. Revenue

$$\boldsymbol{R_t = EP_t * P_t * lf_t * 8760 * 1000} \qquad \textbf{[Eq. 6-3]}$$

Where R = Revenue (£)

EP = electricity price (£/kWh)

P = GT nominal output (MW)

lf = load factor

ii. Expenses

- Fuel Cost

$$\boldsymbol{fuel_t = cf_t * ff_t * lf_t * 8760 * 3600} \qquad \textbf{[Eq. 6-4]}$$

Where cf = cost of fuel (£/kg)

ff = GT fuel flow (kg/s)

- Operation and Maintenance Cost (other than fuel)

$$\boldsymbol{Variable\ Cost_t = vc_t * P_t * lf_t * 8760}$$ **[Eq. 6-5]**

Where vc = variable cost (£/MWh)

$$\boldsymbol{Fixed\ Cost_t = fc * P * 1000}$$ **[Eq. 6-6]**

Where fc = fixed cost (£/kW-year)

- Other Cost

$$\boldsymbol{Depreciation\ Cost_t = d * I / 100}$$ **[Eq. 6-7]**

Where d = annual depreciation (%)

$$\boldsymbol{Emission\ cost_t = EI_t * ET_t * P_t * lf_t * 8760}$$ **[Eq. 6-8]**

Where EI_t = Emission index ($kgCO_2/kWh$)

ET_t= Emission tax (£ /ton CO_2)

Total expense is the sum of all expenses given by:

$$\boldsymbol{Exp_t = fuel_t + Variable\ Cost_t + Fixed\ Cost_t + Depreciation\ Cost_t + Emission\ Cost_t}$$ **[Eq. 6-9]**

Where Exp_t = Total expense at year t

iii. Operating Income (OI_t)

$$\boldsymbol{OI_t = R_t - Exp_t}$$ **[Eq. 6-10]**

iv. Profit (Pr_t)

The profit yielded for the given year is the operating income less the annual capital pay back.

The capital cost of investment is converted into stream of equal annual payment within the life span of the project by using the annuity factor defined as:

$$\boldsymbol{anf} = \left[\frac{\boldsymbol{i} * (\mathbf{1}+\boldsymbol{i})^{\boldsymbol{n}}}{(\mathbf{1}+\boldsymbol{i})^{\boldsymbol{n}}-\mathbf{1}}\right] \qquad \textbf{[Eq. 6-11]}$$

Where i = interest rate

anf = annuity factor (yr $^{-1}$)

n = project life span (years)

Thus,

$$\boldsymbol{inv_t} = \boldsymbol{I} * \boldsymbol{anf} \qquad \textbf{[Eq. 6-12]}$$

Where inv_t= annualised capital investment (£/yr)

$$\boldsymbol{Pr_t} = \boldsymbol{OI_t} - \boldsymbol{inv_t} \qquad \textbf{[Eq. 6-13]}$$

v. Net Cash Flow

Considering taxable income or tax on profit, the net cash flow at the end of year t becomes

$$\boldsymbol{CF_t} = \boldsymbol{Pr_t} * \left(\mathbf{1} - \frac{\boldsymbol{rt}}{\mathbf{100}}\right) - \boldsymbol{inv_t} \qquad \textbf{[Eq. 6-14]}$$

Where rt = profit tax rate (%)

vi. Net Present Value

Discounting the cash flow to the present using a discount rate (interest or inflation rate are most often used), the net present value is given as:

$$\boldsymbol{NPV} = \sum_{t=1}^{n} \boldsymbol{CF_t} \, [\mathbf{1} + \boldsymbol{i}]^{-t} - \boldsymbol{I} \qquad \textbf{[Eq. 6-15]}$$

Where i = discount rate

vii. Payback Period

The payback period is defined as the minimum value of n that satisfies the equation:

$$\boldsymbol{I} - \sum_{t=1}^{n} \boldsymbol{C_t} \, [\mathbf{1} + \boldsymbol{i}]^{-t} \leq \mathbf{0} \qquad \textbf{[Eq. 6-16]}$$

6.6 CO_2 Emission Prediction

The environmental aspect of this assessment deals with the quantification of CO_2 emission from the GT mechanical drives and placing a price penalty on it (carbon tax). The means of predicting the amount of CO_2 emission has been described in section 3.2.3 of this compendium where an in-house prediction code (Hephaestus) is used for the simulation.

The simulated average CO_2 emission index for EL2500RD GT is 0.49 $kgCO_2/kWh$ while that of EL1200-R is 0.56 $kgCO_2/kWh$. The fuel consumption by the small GT (EL1200-R) is about 37% relative to the consumption of the large GT (EL2500RD). However, the power output of the small GT is about 28% of the output from the large GT. In order words, the small GT consumes more fuel per kWh compared to the large one. Hence, the CO_2 emission index of the small GT should be higher than that of the large GT as revealed by the numbers mentioned above. Furthermore, the engines which inspired EL2500RD and EL1200-R were manufactured in 2007 and 1980 respectively [109]. This means there is a huge technology gap between the efficiency of the two combustors. The "efficiency" factor affects CO_2 emission in GT engines.

At this juncture, it is necessary to mention that CO_2 emission index value of 0.21 $kgCO_2/kWh$ is quoted for natural gas fuel used by GT in Defra – a UK Department of Energy and Climate Change document for calculating GHG emission [120]. However, this value is for a fuel lower heating value of 47.37 MJ/kg as against 43 MJ/kg used for the simulation. More so, the reader must bear in mind that de-rating the GT prime movers to meet site ambient condition and part-load performances will negatively impacts on the GT thermal efficiency and hence the CO_2 emission index (refer to Figure 6-11 and Figure 6-12).

In economic terms, the emission cost burden adversely impacts on the profitability of the venture. The comparative nature of analysis carried out will naturally distribute this impact across board; thus minimising the error in the findings which in itself are estimates and could be highly subjective. Notwithstanding though, this value was used in the simulation to assess its impact on the GT economic parameters especially as it applies to the worst case obtained.

6.7 Economic Evaluation of Case Scenario I

The first step in this analysis is to establish the power required for compression across the four seasons highlighted above, representing the varying power demand. The simulation of the compression power requirement for different CO_2 throughput had been presented in section 4.8 of this work. The power evaluation is based on six–staged intercooled integral geared centrifugal compressor with 5% inter-stage pressure drop consideration (2.3 pressure ratio per stage). The next step is the data and assumptions considered for the analyses which are presented below.

6.7.1 Data and Assumptions

For this case scenario, the following key assumptions in the operation of the GT prime movers should be considered:

a. The smaller GT mechanical drive is used to provide the required compression power for the CO_2 emission captured from the Oil and Natural Gas Dual fired power plant while the larger one is used for the CO_2 emission from the remaining power plants.
b. The availability of the GT mechanical drives are such that they are shut down for maintenance within the early rain season (June – August) when the power demand from the power generation plants are relatively low.
c. 25 years projected plant life - from 2013 to 2037.

The values and parameters required to analyse the economic performance of the GT mechanical drives are shown in Table 6-3 below.

Table 6-3 Assumptions for the Economic Analyses

Parameter	Values
Plant Designation	EL2500RD; EL1200-R
Plant Capital Cost(£/kW)	283.67; 377.62
Plant life (years)	25
Discount rate, Interest rate (%)	15
Plant load factor	0.90
Plants capacity (MW)	33.90; 9.40
Auxiliary consumption	Nil
Fixed O & M cost (£/kW-year)	4.80; 5.01
Variable O &M cost (£/MWh)	7.08; 10.54
Fuel cost (£/kg)	0.12
Fuel Lower Heating Value (MJ/kg)	43
Annual Depreciation (%)	2.5
CO_2 Emission Indices ($kgCO_2/kWh$)	0.49, 0.56
Emission tax (£/ton CO_2)	50
Profit tax rate (%)	20
Electricity price (£/kWh)	0.12

Fuel cost values of £0.09/kg and £0.06kg as well as discount rates of 6, 8, 10, and 12% were considered for sensitivity studies.

6.7.2 Results and Analysis

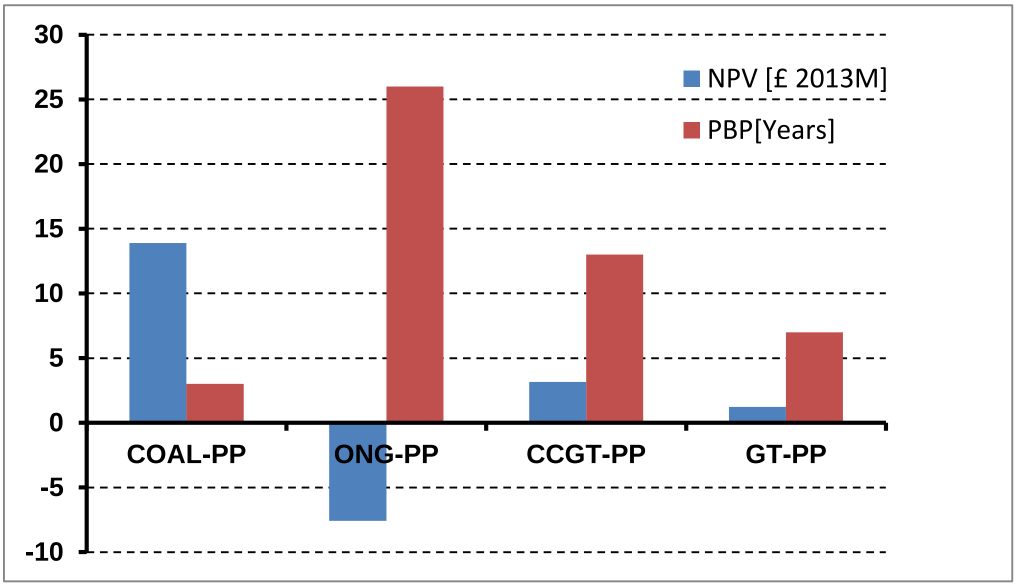

Figure 6-15 Economic Performance of the GT Mechanical Drive in the Four Power Generation Plants (25 Years Plant Life)

Figure 6-15 above show the main economic results obtained using the parameters itemised in Table 6-3 above for deploying GT mechanical drives in compressing CO_2 captured from four power generation stations. A typical result is shown in Appendix E. The capital cost of the two GT prime movers or mechanical drives are about £9.6M and £3.6M for the 33.9 MW and 9.4 MW respectively. Clearly, the economic performance of the GT mechanical drive applied to the CO_2 throughput from the “Coal Power Plant (COAL-PP)” stands out to be more attractive (NPV of about £13.9M) while that of the “Oil and Natural Gas Dual Fired Plant (ONG-PP)” is most unattractive leaving the investor with a deficit NPV of about £7.6M at the end of the 25 years plant life. The GT mechanical drive’s NPV for the Combined Cycle Gas Turbine Power Plant (CCGT-PP) is about £3.2M which makes it relatively attractive. However, the return on investment in the GT application to the COAL-PP is higher and offers the best choice.

The investment on GT driven compression of the CO_2 throughput from the Gas Turbine Power Generation Plant (GT-PP) is equally attractive as a NPV value of about £1.2M is obtained. It should be noted here that the GT prime mover used in compressing CO_2 from the GT-PP (EL1200-R) is different and smaller in size compared to that (EL2500RD) used for the other three power generation plants. Therefore, comparison will depend on the GT prime mover in relation to the power generation plants.

In terms of investment risk or time taken to recoup the initial capital invested in the GT prime movers, the COAL-PP shows a payback period (PBP) of three years. The period is quite short hence making it less risky and further enhanced its attractiveness. On the other hand, the PBP for the GT prime mover application to the CCGT-PP is thirteen years which is just a year above half of the prime mover’s life. Although a relatively profitable investment, the PBP put the investor at a medium risk level. In the case of the GT prime mover used in the GT-PP, its PBP is seven years; and were we to judge the risk associated with it, in comparison to the GT size and capital cost it could pass as a low risk investment.

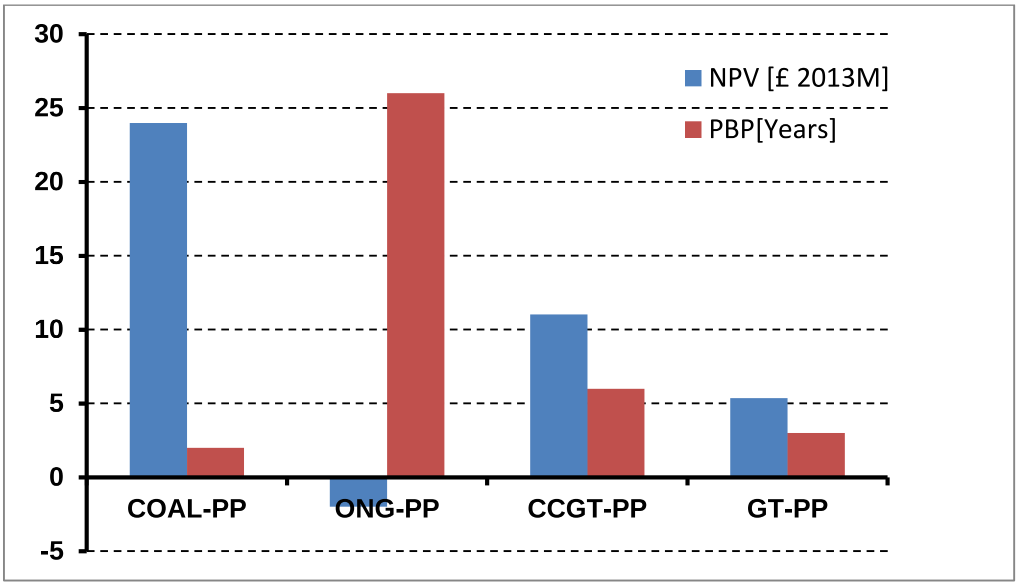

Figure 6-16 Economic Performance of the GT Mechanical Drives Considering Emission Index of 0.21kgCO_2/kWh (25 years Plant Life)

Assuming an equal emission index of 0.21 kgCO_2/kWh; the investment analysis brings to bear the impact of cut in expenses as can be deduced from Figure 6-16 above. The economic performance trend is similar to the one above despite the fact that emission cost was paid on a very low CO_2 emission index compared to that of Figure 6-15 above; especially the fact that the ONG-PP still returns a negative NPV. However, one can observe that the reduction on expense has virtually placed the prime movers' application to the remaining three power generation plant in a favourable economic climate. The NPV and PBP are respectively over £24M and two years; £11M and six years; and £5.3M and three years for the GT compression duties in the COAL-PP, CCGT-PP and GT-PP respectively. This is a huge improvement.

The economic performance of the 33.9 MW GT mechanical drive in application to the three power generation stations (COAL-PP, ONG-PP and CCGT-PP) as shown above could also be analysed in terms of changing CO_2 throughput (refer to CO_2 throughput in Figure 6-2 to Figure 6-5 above). Assuming the CO_2 being compressed is captured from a single power generation station with load swings which changes the CO_2 throughput. This possibility especially comes to play considering the fact that renewable power generation will force most power plants to work at intermediate load condition. One could easily infer that the more the CO_2 throughput decreases from the design maximum throughput, the lower the NPV, the higher the risk and in the worst case scenario (considering the ONG-PP CO_2 throughput) the investment would operate at a loss. Given the necessity of CO_2 abatement from power generation plants as earlier established, some initial investment projections could guide government's policy in creating a healthy economic climate to prospective investors. This takes us to the influence of two major cost components that affects revenue generation assuming constant electricity price i.e. discount rate and fuel cost.

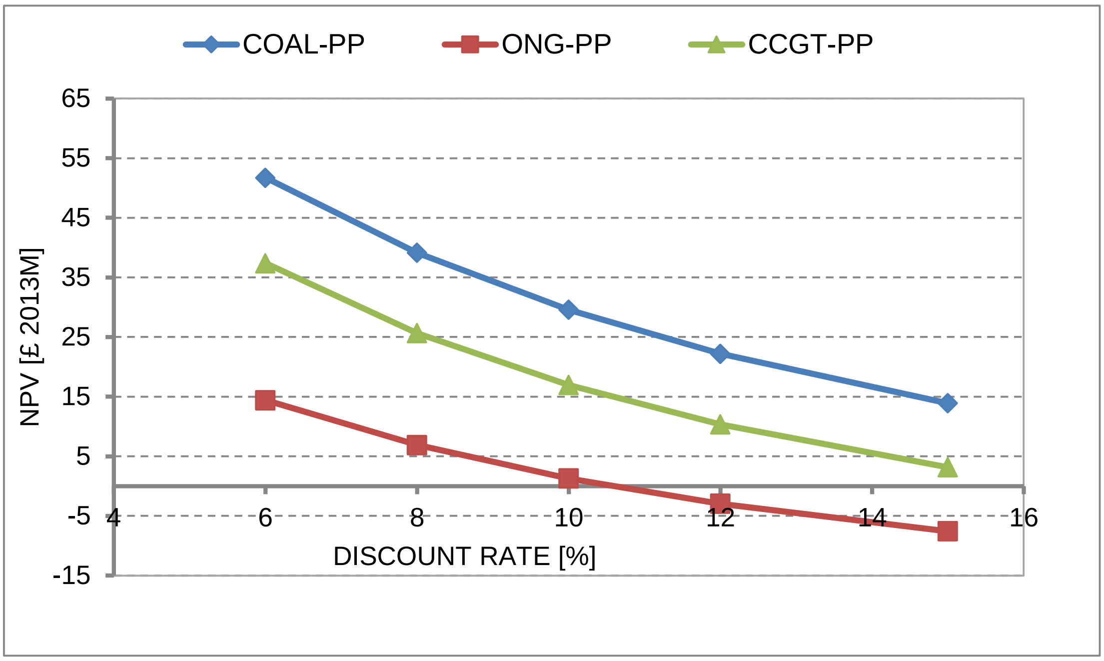

Figure 6-17 Effect of Discount Rate on the Net Present Value of the 33.9 MW GT

Figure 6-17 above and Figure 6-18 below show the influence of discount rate on the NPV and PBP respectively of 33.9 MW GT compression duty. The lower the discount rate the better the NPV and PBP for the investment. At very low CO_2 throughput typified here by ONG-PP, a maximum discount rate of 10% will ensure the economic performance by the GT prime mover is such that the capital invested is recouped just before the end of the plant life. In this light, government policy aimed at reducing inflation and interest rates to a single digit percentage will be an encouragement to investors in the CO_2 pipeline transport.

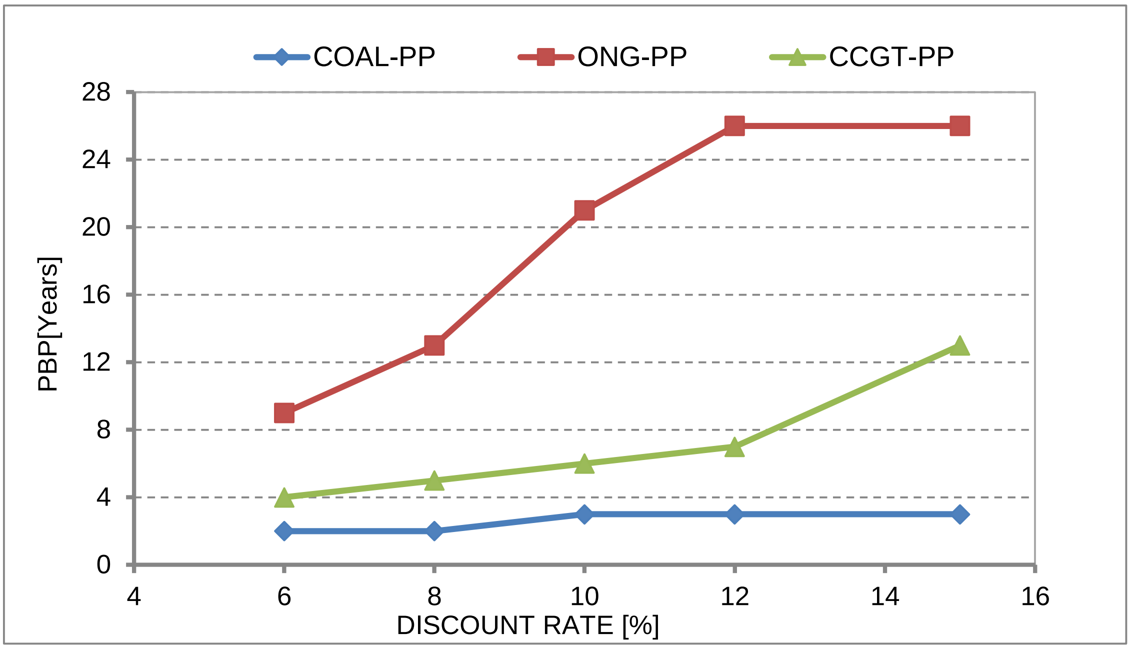

Figure 6-18 Effect of Discount Rate on Payback Period of the 33.9 MW GT

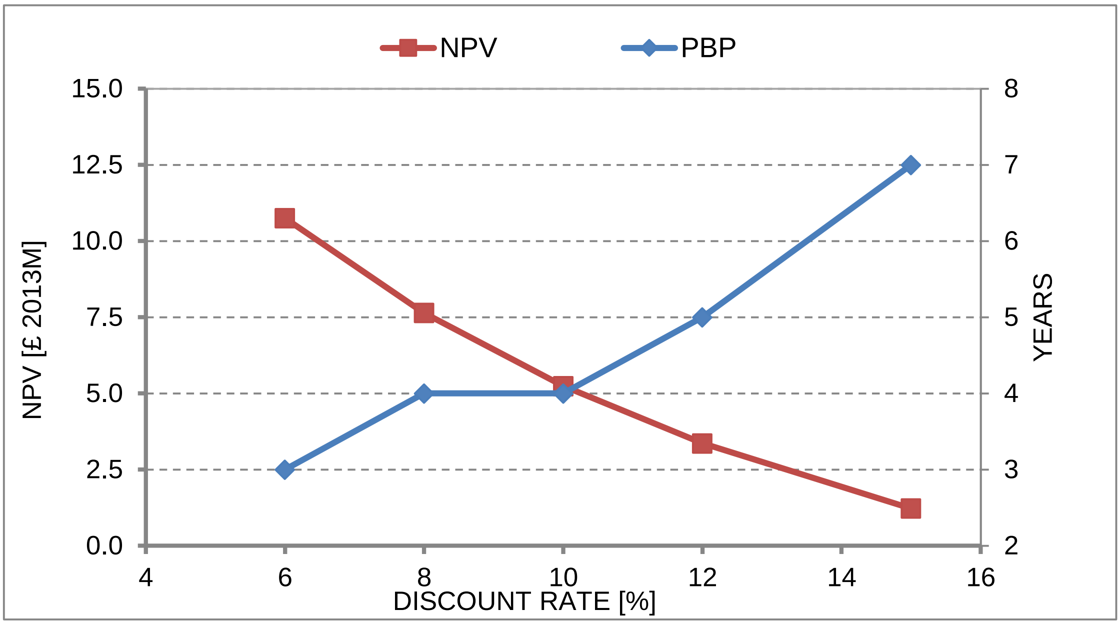

Figure 6-19 Effect of Discount Rate on the NPV and PBP of the 9.4 MW GT

Figure 6-19 above depicts the influence of discount rate on the NPV and PBP of the GT mechanical drive used in compressing captured CO_2 from the GT-PP.

The trend is similar to the ones presented above. Owing to the fact that this GT drive was originally selected to meet the low CO_2 throughput from the power generation station; in comparison to Figure 6-17 and Figure 6-18 above, the economic performance in the worst case here i.e. 15% discount rate is still appreciable. At this rate the NPV is over £1.2M while the PBP is seven (7) years compared to a deficit NPV of £7.6M at the end of the plant life for the 33.9 MW GT compression duty of the ONG-PP. In this light, one might be pushed to suggest that two different sizes of GT prime mover could be provided to cope with changing CO_2 throughput. However, the prohibitive cost of owning and operating a GT engine will hardly give room for such arrangement especially as applied to CO_2 pipeline whose investment knowledge is still at infancy stage.

Reduction in variable operation and maintenance cost whose major contribution is from fuel consumption is another alternative explored here.

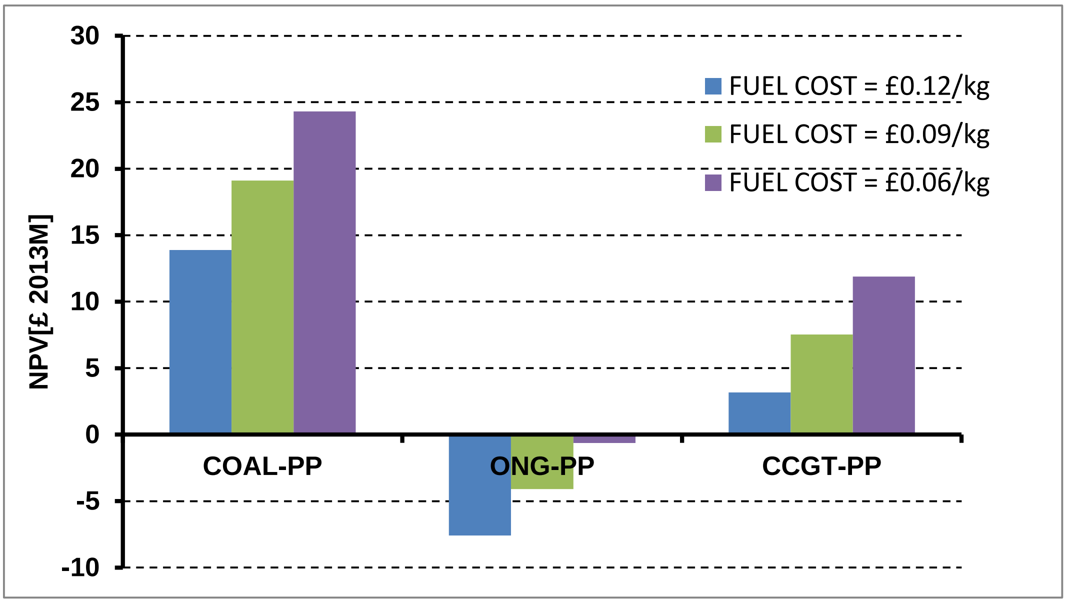

Figure 6-20 Effect of Fuel Cost on the Net Present Value of the 33.9 MW GT

Figure 6-20 above and Figure 6-21 below show the effects of changes in fuel price on the NPV and PBP respectively of the 33.9 MW GT mechanical drive. Similarly, Figure 6-22 below depicts the effect of fuel price on the NPV and PBP of the 9.4 MW GT mechanical drive. Three natural gas prices were assumed: a current price of £0.12/kg and two other prices at 25% and 50% reduction considering the current trend of fall in oil and gas price at the global market (note that these prices are mere assumptions for the purpose of analysis only and by no way represent current market prices).

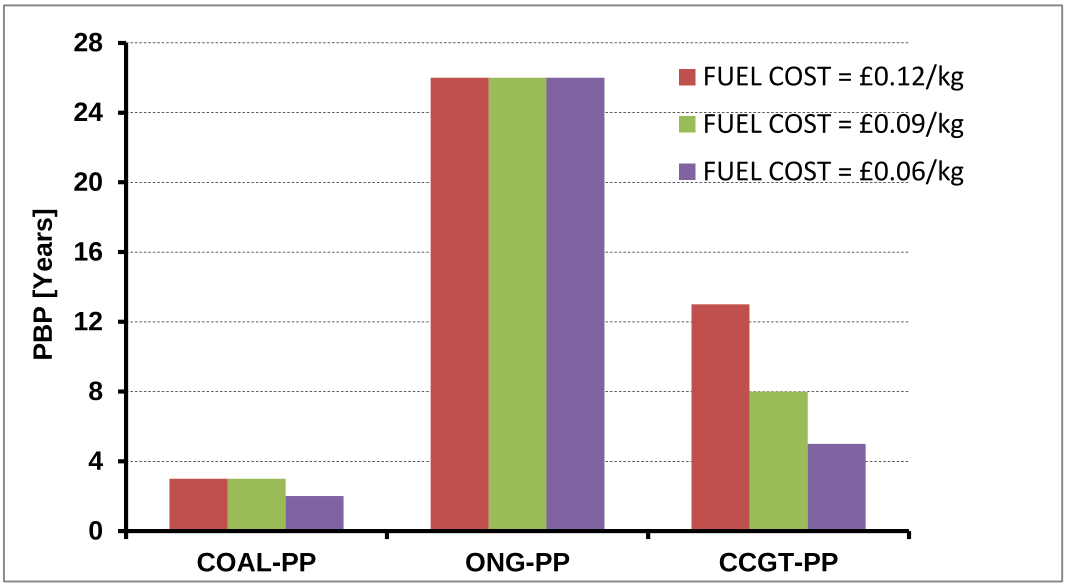

Figure 6-21 Effect of Fuel Cost on Payback Period of the 33.9 MW GT

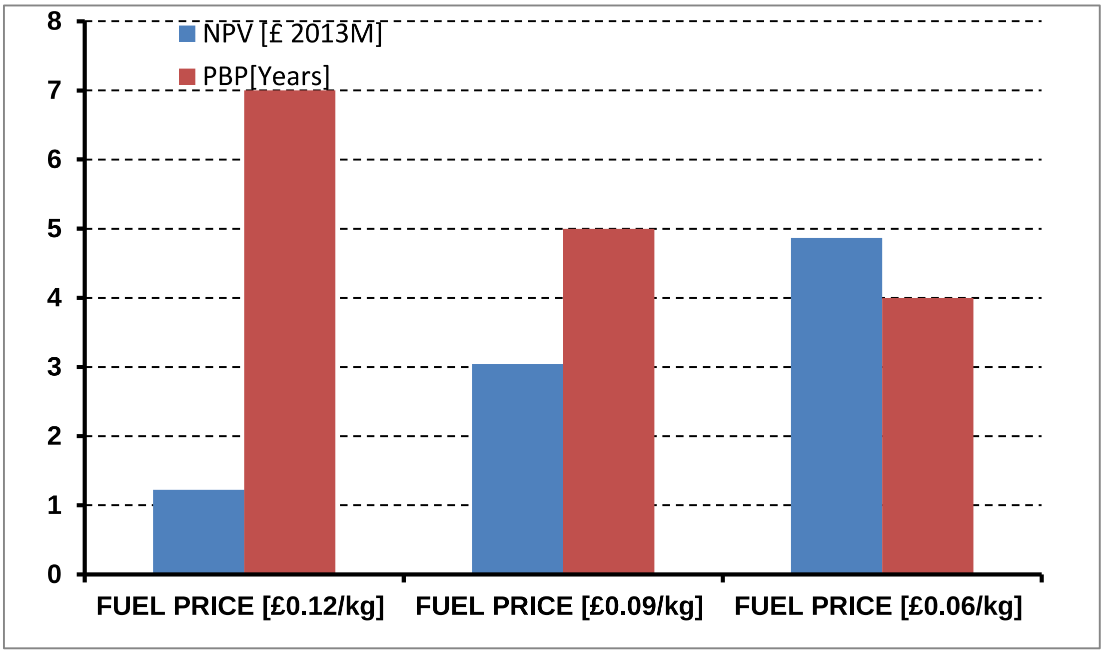

Figure 6-22 Effect of Fuel Cost on the NPV and PBP of the 9.4 MW GT

In Figure 6-20 and Figure 6-21 above, with decreasing fuel price, the economic performance of the GT mechanical drive increase in its value save for its application on the ONG-PP which is still in deficit. However, the deficit in the ONG-PP reduced from about £7.6M to about £0.6M with 50% reduction in the price of fuel. In application to the remaining two power generation plants, at 50% reduction in the price of fuel, the NPV with the COAL-PP nearly doubled its value (£13.9M - £24.3M) while that of the CCGT-PP nearly quadrupled (£3.2M - £11.9M). The high impact of fuel cost on the profitability of this investment is highlighted by this trend. Similarly, the lowering of fuel prices also impact positively on the risk of recouping the initial investment as observed in the downward trend of the PBP especially with the CCGT-PP where it improved from thirteen to five years.

In the particular case of the 9.4 MW GT mechanical drive, the NPV appreciated from over £1.2M to about £4.9M in value with 50% reduction in the assumed current fuel price. The PBP is also lowered to nearly half of the initial time frame. In the medium price range, the NPV nearly tripled its initial value while the PBP reduced to five years from the initial seven years.

One economic parameter that has been kept constant so far is the price of electricity which is the only source of revenue ascribed to this model. The NPV and the PBP are positively influenced by increase in electricity price all other cost being constant. The selling price of electricity may naturally increase because of the additional cost incurred in making the power generation plants carbon capture ready. However, the envisaged reduction in natural gas price might force it to remain steady keeping the economic performance trending as above. Nevertheless, an increase in electricity price will affect the economic performance of the GT prime mover in same manner as a decrease in discount rate or fuel price explained above

6.7.3 Summary

The success of GT application to CO_2 pipeline is hinged on the ability to maintain the available CO_2 throughput as close as possible to the rated throughput that guided the selection of the prime mover at onset. Cut in expenditure in the form of reduced discount rate and fuel cost will in no measurable terms ensure desired profitability is achieved because return on investment is increased.

6.8 Economic Evaluation of Case Scenario II

The case description has already been presented in section 6.2.2. In order to evaluate the economic performance of the GT prime mover, the hydraulic analysis of the pipeline profile for a given pipe diameter and CO_2 throughput need be carried to delineate the positioning of the booster station. With a pipeline inlet pressure of 150 bar, 100 bar is set as minimum threshold for recompression to avoid two-phase flow which ensures that pumps are safely used for boosting the pressure. This 100 bar minimum pressure limit set for the CO_2 flow along the pipeline is in accord with recommendation in literature [64; 68; 84-86].

6.8.1 Hydraulic Analysis Data and Assumptions

The dense-phase CO_2 flow in the buried pipeline is assumed to be isothermal at a constant flowing temperature of $25°C$ $(77°F)$. Since the pipeline operating pressure is fixed at 150 bar, the prevailing factors that will drive this analysis (i.e. to determine the pressure drop during transmission) will be the pipe size (internal diameter), the CO_2 throughput, the minimum pressure limit and the elevation change (due to the high density of the CO_2). The CO_2 throughputs were corrected to standard conditions (14.696 psia; 60°F) while pressure loss due to pipe connections and valves were neglected.

It was earlier revealed that the dense phase have a viscosity similar to that of a gas and a density near that of a liquid, hence literature has documented the pressure drop modelling of CO_2 flow in the liquid phase using Bernoulli equation [68] while the gaseous phase has been employed by the vast majority [2; 10; 55; 57]. The mere fact that the CO_2 flow in pipeline still undergo expansion and contraction; in order words compressibility, inevitably dismisses analysing the pressure drop as an incompressible flow or liquid.

The influence of impurities in the hydraulic analysis of the captured CO_2 per se is a matter of concern. However, in order to simplify the hydraulic analysis, the current study assumes the CO_2 is pure while employing established gas flow equations to evaluate the pressure drop caused by the high density CO_2 flow. The compressibility properties were estimated using the PR-EOS - widely used for CO_2 pipeline flow analysis and said to be among the most precise and proper for engineering application [9; 53; 55; 63].

Three API 5L X-70 line pipe sizes of 323.9 mm (12¾″), 355.6 mm (14″) and 610 mm (24″) were assumed. Meanwhile, the 12¾″ and 14″ pipe sizes were specifically chosen in order to keep the CO_2 flow velocity as close as possible to within 1.5 – 4m/s said to be the most cost effective velocity for dense phase CO_2 transmission [8; 9]. Furthermore, these two pipeline sizes are similar to the existing Weyburn CO_2 trunk pipeline connecting USA and Canada where the CO_2 is used for enhanced oil recovery (EOR) [2; 55]. The pipe wall thickness $t(m)$ for a pipe of outside diameter $D(m)$ is evaluated using the relationship specified in the US Code of Federal Regulation (CFR 2005) as [55; 85]:

$$\boldsymbol{t = \frac{P_{max} D}{2SFE}} \qquad \textbf{[Eq. 6-17]}$$

Where P_{max} (MPa) is the maximum operating pressure of the pipeline which is chosen as $15.3 MPa$; S (MPa) is the minimum yield strength of the pipe specified as $483\ MPa$ for the API 5L X-70 line pipe selected; F is design factor for safety taken as 0.72 and E is the seam joint factor set to 1.0 based on seamless weld pipe assumption [39; 55; 68]. The summary of the pipeline parameters are shown in Table 6-4 below.

Table 6-4 Parameter Values of Pipeline

Parameter	Values
Maximum Operating Pressure(MPa)	15.3
Initial Pipe Inlet Pressure(KPa)	15000,10500
CO_2 Density at Inlet (kg/m^3)	876.3
CO_2 Viscosity at Inlet(Pas)	9.2605×10^{-5} [98]
Isothermal Flow Temperature(°C)	25
Base Temperature(°C)	25
Base Pressure (KPa)	101.325
Pipe Outside Diameters (mm)	610, 355.6; 323.9
Pipe Outside Diameters (inch)	24; 14; 12¾
Minimum Yield Stress (MPa)	483
Longitudinal Joint Factor	0.72
Design Factor	1.0
Absolute Pipe Roughness (mm)	0.045 [49; 121]
Pipe Thickness (mm)	13.45; 7.84; 7.14
CO_2 Throughput (MTPY)	3.70; 2.41; 2.78; 3.33
CO_2 Specific Gravity	5.12
Profile Horizontal Distance (km)	511

6.8.2 Modelling the Pipeline Flow

The flow of the CO_2 in the pipeline was modelled using the fundamental flow equation cast in two different forms as obtained from literature. The first form of the equation (in SI units) said to be widely used in the pipeline industry is cast thus [122]:

$$\boldsymbol{Q = 47880 * \frac{T_b}{P_b} E * D_{in}^{2.5} \left[\frac{p_1^2 - p_2^2 - \frac{0.0375 * SG * (H_2 - H_1) * p_a^2}{Z * T_a}}{SG * Z * T_a * L * \lambda_{DW}} \right]^{0.5}} \qquad \textbf{[Eq. 6-18]}$$

The second form of the equation also in SI units and considering the effect of elevation difference between the upstream and downstream ends of the pipeline segment is cast as [121]:

$$\boldsymbol{Q = 5.7473 * 10^{-4} * F \frac{T_b}{P_b} * D_{in}^{2.5} \left[\frac{p_1^2 - exp^s p_2^2}{SG * Z * T_a * L_{eq}} \right]^{0.5}} \text{ [62][62]} \qquad \textbf{[Eq. 6-19]}$$

Or

$$\boldsymbol{Q = 11.4946 * 10^{-4} * \frac{1}{\sqrt{\lambda_{DW}}} \frac{T_b}{P_b} * D_{in}^{2.5} \left[\frac{p_1^2 - exp^s p_2^2}{SG * Z * T_a * L_{eq}}\right]^{0.5}}$$ [62][62] **[Eq. 6-20]**

With:

$$\boldsymbol{P_a = \frac{2}{3}\left[P_1 + P_2 - \left(\frac{P_1 P_2}{P_1 + P_2}\right)\right]}$$ **[Eq. 6-21]**

$$\boldsymbol{s = \frac{0.0375 * SG * (H_2 - H_1)}{Z * T_a}}$$ **[Eq. 6-22]**

$$\boldsymbol{L_{eq} = \frac{L(exp^s - 1)}{s}}$$ **[Eq. 6-23]**

And for n number of segments $L_1, L_2, L_3, \ldots, L_n$ that make up the pipeline length L, the equivalent length is defined as:

$$\boldsymbol{L_{eq} = j_1 L_1 + j_2 L_2 e^{s1} + j_3 L_3 e^{s2} + \cdots + j_n L_n e^{s(n-1)}}$$ **[Eq. 6-24]**

Where

Q = gas flow rate at standard conditions (m^3/day)

T_b = base temperature (K)

P_b = base pressure (KPa)

T_a = average flow temperature (K)

P_a = average pressure (KPa)

P_1 = upstream pressure (KPa)

P_2 = downstream pressure (KPa)

H_1 = upstream elevation (m)

H_2 = downstream elevation (m)

L = pipeline segment length (km)

L_{eq} = equivalent pipeline segment length (km)

Z = gas compressibility factor ($nondimensional; evaluated\ at\ T_a and\ P_a$)

SG = gas specific gravity ($nondimensional$)

D_{in} = inside diameter of pipe (m)

E = pipeline efficiency *(decimal value less than 1.0)*

λ_{DW} = Darcy-Weisbach friction factor ($nondimensional$)

F = transmission factor *(nondimensional,* $\lambda_{DW} = 4/F^2$*)*

s = elevation correction factor ($nondimensional$)

The Darcy-Weisbach Friction factor is obtained using the Colebrook-White equation defined as:

$$\frac{\mathbf{1}}{\sqrt{\boldsymbol{\lambda_{DW}}}} = -\boldsymbol{2log_{10}}\left(\frac{\boldsymbol{e}}{\mathbf{3.7}\boldsymbol{D_{in}}} + \frac{\mathbf{2.51}}{\boldsymbol{Re}\sqrt{\boldsymbol{\lambda_{DW}}}}\right)$$ **[Eq. 6-25]**

Or in terms of the transmission factor;

$$\boldsymbol{F} = -\boldsymbol{4log_{10}}\left(\frac{\boldsymbol{e}}{\mathbf{3.7}\boldsymbol{D_{in}}} + \frac{\mathbf{1.255}\boldsymbol{F}}{\boldsymbol{Re}}\right)$$ **[Eq. 6-26]**

(For $Re > 4000$)

e = absolute pipe roughness *(mm)*

Re = Reynolds number *(nondimensional)*

The Reynolds number is given by the expression

$$\boldsymbol{Re} = \mathbf{0.5134}\ \frac{\boldsymbol{P_b * SG * Q}}{\boldsymbol{P_b * \mu * D_{in}}}$$ **[Eq. 6-27]**

μ = dynamic viscosity *(Pas)*

A FORTRAN code was developed by the author to model the flow [Eq. 6-20] which is adopted for this study. The CO_2 compressibility factor for flow along the pipe segments were computed via a subroutine containing the PR-EOS algorithm (the PR-EOS formulations are highlighted in section 4.4.1). At any point along the segments that the pipeline flow pressure falls below a set minimum, a warning message “POINT OF RECOMPRESSION” is displayed.

The code reads the pipeline profile through an input file containing values for the segments lengths and corresponding upstream and downstream elevation values. The design of the code to read from an input file gives it the ability to simulate pressure drop of any given profile with slight modification. Given the available pipeline profile data, a total of thirty-one (31) pipeline segments of unequal lengths - a maximum of 28 km and minimum of 7 km are read as input. The code and its input file are as shown in Appendix F.

6.8.3 Techno-Economic Analysis of the Hydraulic Simulation Results

Considering four seasons in a year as earlier established in the preceding sections (see Figure 6-2 to Figure 6-5 above), average CO_2 throughput values of 3.70 MTPY (0.27 mmscfd), 2.41 MTPY(0.18 mmscfd), 2.78 MTPY(0.20 mmscfd) and 3.33 MTPY(0.24 mmscfd) for the hot, early rain, late rain and harmattan seasons respectively were simulated to obtain the pressure distribution along the pipeline profile (a typical output is shown in Appendix G). Owing to the elevation differences in the chosen profile, the 511 km horizontal distance has given rise to an equivalent length of 511.52 km. The behaviours of pressures along the pipeline profile with the average seasonal flows at standard conditions are discussed below.

In actual practice, pipelines are sized and selected for maximum flow capacity or throughput to reduce pressure drop in the system within economic limits. Thus the larger the pipe size for a given flow, the lower the pressure drop. Similarly, for a given pipe size, a reduction in flow from the design capacity gives a lower pressure drop. This phenomenon is illustrated in Figure 6-23 to Figure 6-25 simulated for the 24″- diameter, 14″- diameter and 12¾″- diameter pipelines respectively. The seasonal CO_2 throughputs values are quite close, hence the clustered nature of the plots.

A comparison between the figures revealed that the delivery pressures of the CO_2 throughputs from the 24″- diameter pipeline are relatively higher compared to those from the 14″- diameter and 12¾″- diameter pipelines. Similarly, the delivery pressures of the CO_2 throughputs from the 14″- diameter pipeline are relatively higher compared to that from the 12¾″- diameter pipeline. Obviously, the pressure drop in the 12¾″- diameter pipeline is highest since the smaller the cross - sectional area of flow, the higher the resistance to flow and vice-versa.

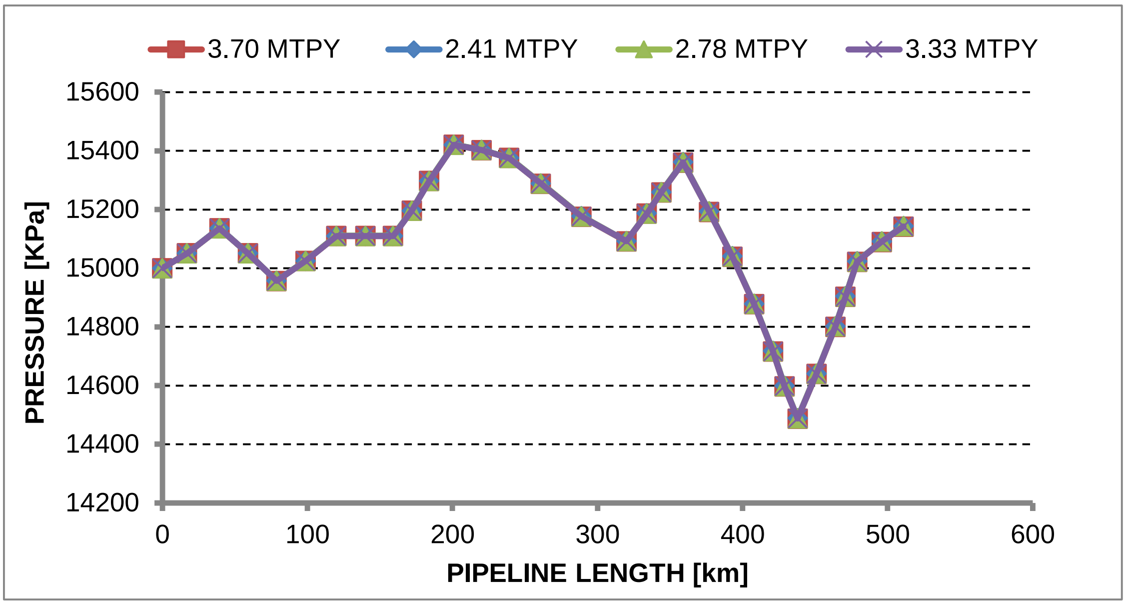

Figure 6-23 Variation of Pressure with Changing CO_2 Throughput at Standard Condition (24″-Pipe Size)

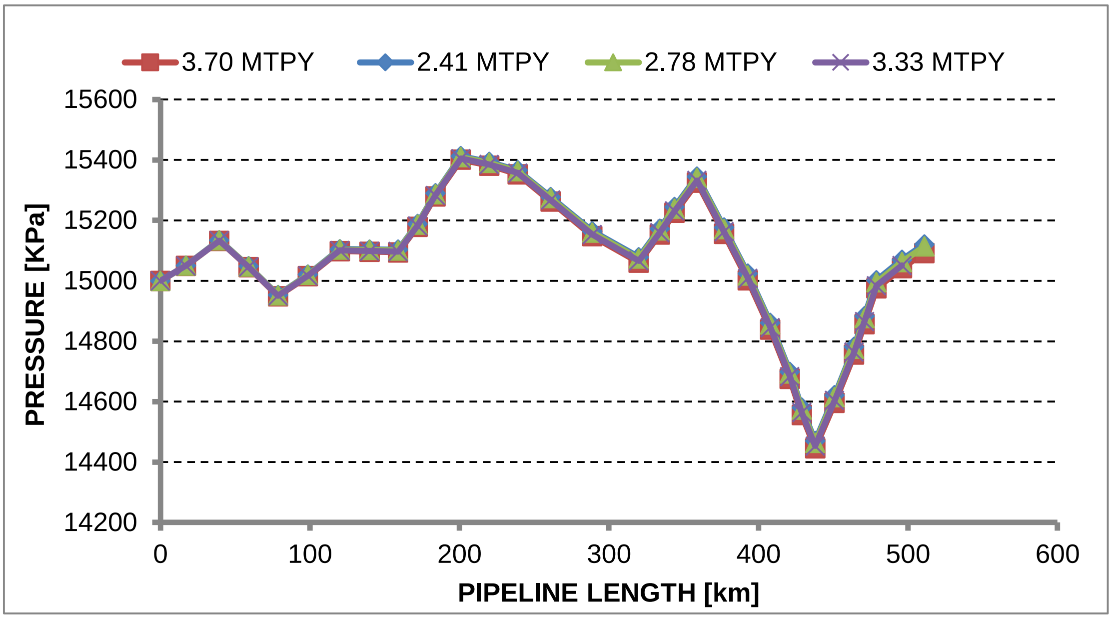

Figure 6-24 Variation of Pressure with Changing CO_2 Throughput at Standard Condition (14″-Pipe Size)

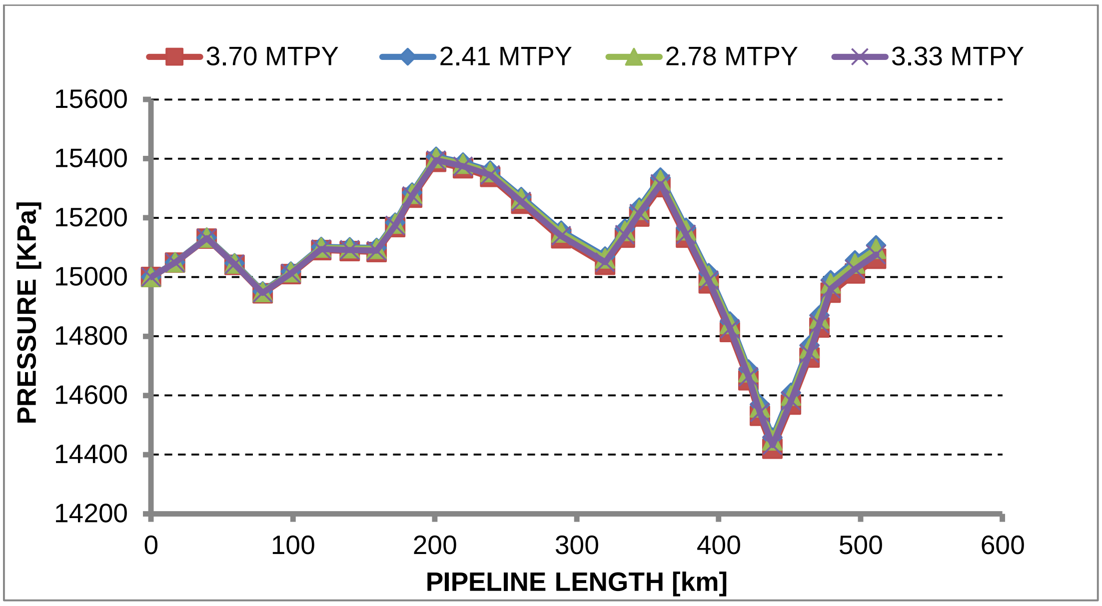

Figure 6-25 Variation of Pressure with Changing CO_2 Throughput at Standard Condition (12¾″-Pipe Size)

Again, considering the figures individually, one observes that the higher the CO_2 throughput, the more the pressure drop and hence the lower the delivery pressure. This can be attributed to the fact that, for a given pipe size, an increase in throughput increases the degree of turbulence or Reynolds number; which invariably increases the pressure loss and hence lower the delivery pressure.

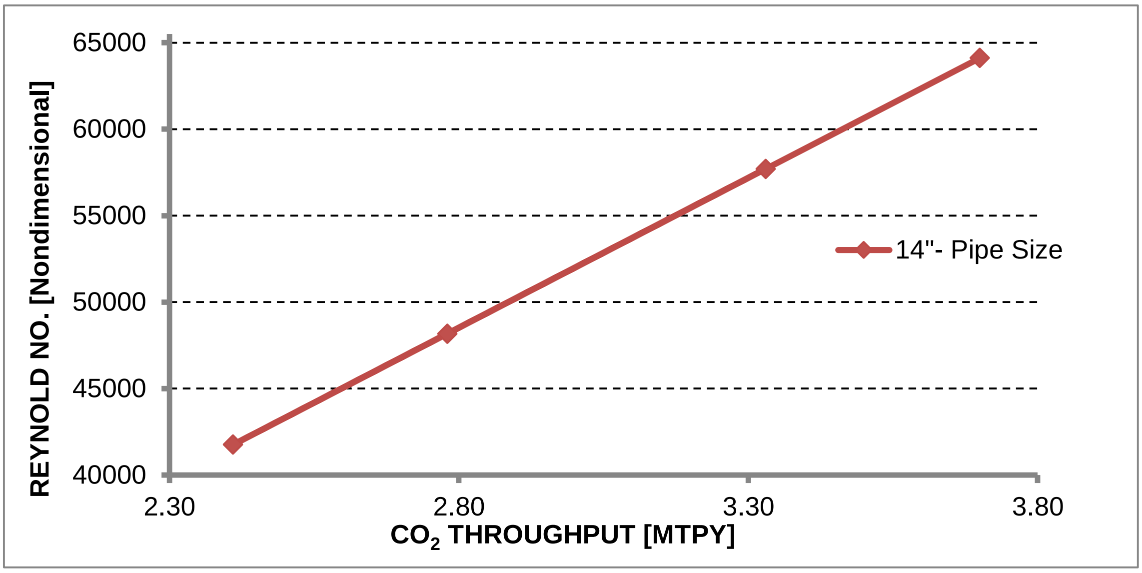

Figure 6-26 Variation of Flow Reynolds Number with CO_2 Throughput

The variation of Reynolds number with the seasonal CO_2 throughput for the14″-diameter pipeline is shown in Figure 6-26 above. These trends on their own merits validate the code used for these hydraulic analyses.

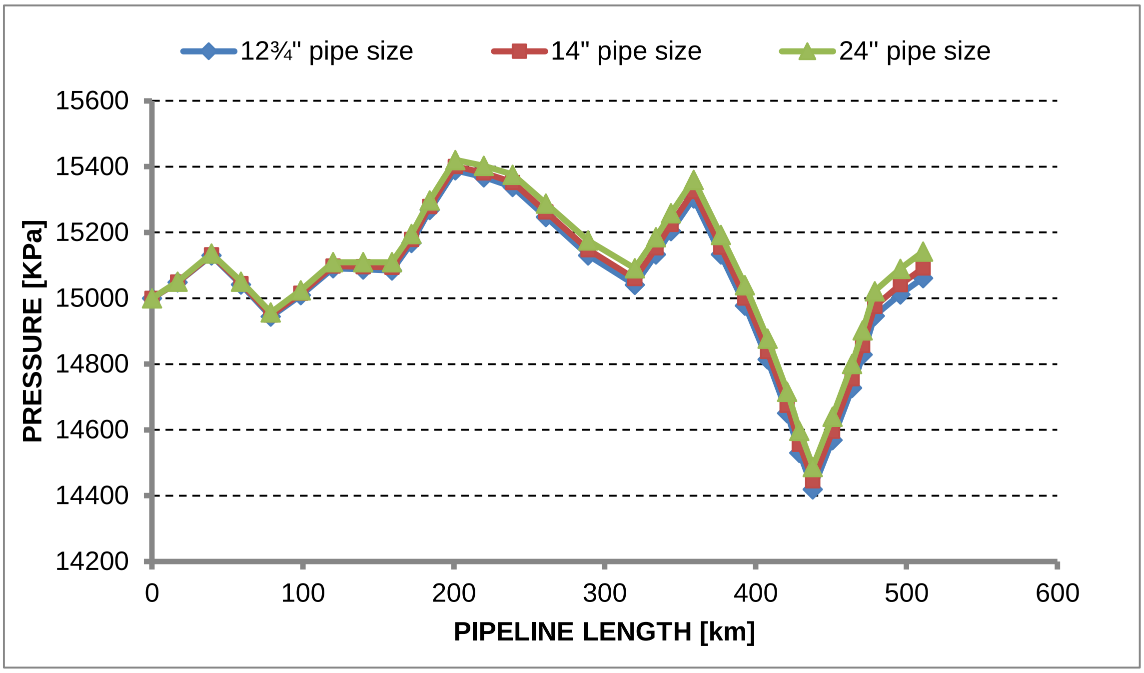

Figure 6-27 Variation of Pressure along the Pipeline Profile for the Chosen Pipe Sizes at Standard Condition (CO_2 Throughput = 3.7MTPY)

Against this back drop, the need for recompression or otherwise will be built upon the maximum seasonal CO_2 throughput (3.70 MTPY) established in this study. From the engineering point of view, the choice of maximum throughput is to base the analysis on the worst case scenario since the highest pressure drop in the pipeline is obtained at the maximum throughput. The satisfaction of the worst case automatically satisfies the other lower flow capacities associated with the pipeline in the current study.

Figure 6-27 above depicts the pressure variation along the pipeline at the maximum CO_2 throughput used for this analysis. Having established the fact that the pressure drop is lowest in the 24″- pipe compared to the 14″ and 12¾″-pipes, the pressure distribution along the pipeline profile is comparatively higher in the 24″- pipe. An immediate eye catching feature of this plot is that the outlet pressure at the receiving terminal is higher than the pipeline inlet pressure.

Thus the overall effect of the profile's elevation difference is the minimising of pressure losses and to a very large extent gain in pressure during the CO_2 transmission. This positive influence on the pressure distribution along the pipeline profile therefore negates any need for an intermediate booster station. To corroborate this finding, the Weyburn CO_2 pipeline which consist 14″ and 12″ pipe sizes, transporting over 1.8MTPY of CO_2 (contains 96% CO_2 + other impurities) along a 330 km pipeline has a delivery pressure of 15200 KPa without an intermediate compressor station [1]. Although the inlet pressure was not mentioned, it is obvious that it would not exceed 15300 KPa which is the maximum allowable operating pressure with ASME-ANSI 900# flange for the line-pipe used in transporting the CO_2.

Another aspect of concern in this plot is the pressure spike above the pipeline maximum allowable operating pressure which brings about issues of mechanical integrity and pipeline safety requiring pressure relief. Assuming this is a preliminary assessment of the flow, it therefore suggest a reduction of the initial inlet pressure of the CO_2 into the pipeline to keep the flow pressure within limits for this particular pipeline profile. This being the case, huge savings in economic terms is achieved since the attendant cost of booster station is eliminated; which means the projected cost of applying the GT prime mover in the CO_2 pipeline will be reduced.

In the light of the above, simulation was carried out to obtain the minimum inlet pressure that can be obtained along this profile to keep the flow within safety limits. The plot shown in Figure 6-28 below depicts that at pipeline inlet pressure of 10500 KPa, the delivery pressure for the pipeline profile is about 10448 KPa. This eliminates the need of combining a compressor and pump for the initial pressure boost as suggested in literature because the increase in capital cost associated with owning a pump and associated driver far outweighs the extra expense incurred in acquiring a rated GT mechanical drive for this compression duty.

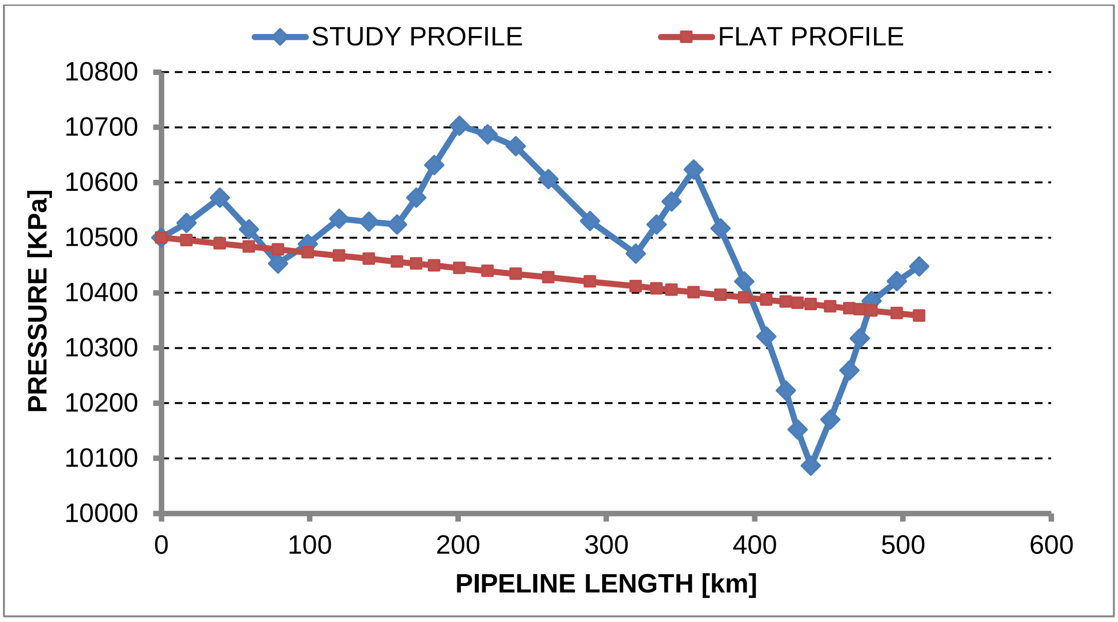

Figure 6-28 Variation of Pressure along the Study Profile and Flat Profile in the 12¾″- Pipe at Standard Condition (CO_2 Throughput = 3.7MTPY)

Perhaps the only advantage of employing a pump for pressure boost will be at the injection site where the scenario may warrant increase in discharge pressure beyond delivery pressure due to build-up of pressure within the storage cavity. Equally noteworthy is that at this inlet pressure, the pressure spike along the pipeline profile is maintained well below the maximum allowable operating pressure.

Assuming the profile is flat, Figure 6-28 also depicts the simulated pressure distribution along the pipeline profile. Using same inlet pressure of 10500 KPa, the delivery pressure is about 10359 KPa. Obviously, the difference in delivery pressures between the profiles is due to the contribution from elevation in the study's profile. In essence, with an inlet pressure of 10.5 MPa, the given flow pressure will not fall below the 10 MPa specified minimum up to triple the current pipeline distance for this given pipe size. Thus the requirement for recompression is once again defeated. Similar conclusion was arrived at in a case study using similar equation on same pipe size over a flat profile but with a CO_2 inlet pressure and throughput of 15.2 MPa and 1.46 MTPY respectively [85]. A delivery pressure of 14.7 MPa over a length of 2562 km was obtained for an average flow temperature of 14°C [85]. This case study indicates the pressure loss during transmission is quite low and is relatively similar to those obtained from the simulation results discussed so far. Therefore, the simulation results as obtained could be said to be justifiable within the set constraints.

6.8.4 Summary

Given the study assumptions and the selection of an economically optimized pipe diameter, the pressure drop along the pipeline is determined by the CO_2 throughputs. The pressure drop is high when the magnitude of the throughput is high and vice-versa. Judging by the results obtained, it may prove difficult for one to accept that during transmission, the high pressure CO_2 pipeline might require a booster station before the point of delivery within 1500 km for a flat terrain if the inlet pressure is maintained at 10.5 MPa, let alone 15M Pa. More obvious is this submission when a real life profile is considered; except for the situation where the pipeline profile climbs uphill for a considerable distance which is uncommon. In addition, the need of combining a compressor and pump for the initial pressure boost becomes uneconomical especially as the difference in pressure is no longer 5 MPa but 0.5 MPa as obtained from the simulation. Equally, the increase in capital cost associated with obtaining a pump and associated driver will not be justified when compared to the expense incurred in acquiring a rated GT mechanical drive to meet the compression duty.

6.9 Validation of the TERA Framework

The entire assessment is built around the TERA framework comprising of the four different modules discussed in chapter 3. The validity of the analysis is hinged upon the validity of the individual modules that make up the TERA framework.

The Pipeline/Compression module as presented is based on established equations for compression and fluid flow analysis. Thus, the results obtained under the stated assumptions are within reasonable accuracy in principle. Furthermore, the employed compression equations are those used for the preliminary design analysis of compression equipment while the flow curves obtained follow established trends. However, the presence of impurity in the captured CO_2 which affects fluid properties during compression and transportation and hence amount of GT power required, is an issue to be considered. Another issue worthy of consideration lies with the state (dense-phase) of the CO_2 been transported. The dense-phase shares the characteristics of both liquid and gas as expounded by literature. This unusual characteristics equally affects properties of density and compressibility of the CO_2 being transported.

The existence of issues of this nature, could reduce the reliability of the module's results. However, the effect of impurity is minimised by the high level of purity expected from the CO_2 capture plants before initial compression to pipeline operating pressure. Hence, for a preliminary assessment, the Pipeline/Compression module gives substantial accuracy. To conclude, it is worth mentioning that the best method to validate this module will be a field study especially as the very nature of CO_2 in the dense-phase is still been researched. This is beyond the scope of this work.

The design point and off-design performance outputs used for this assessment under the Engine Performance Module are based on validated in house gas turbine performance codes. These codes are propriety to Cranfield University and it is no gain saying that they compete favourably with existing commercial gas turbine simulation software like GasTurb. These codes are continuously improved to meet with current challenges in performance analysis of the gas turbine engine.

The result obtained from the Emission Module is quite generic and could be more accurate if the geometry of the individual combustors were captured in the emission simulation code. However, owing to the flexibility in the choice of the CO_2 emission levy during evaluation and the comparative nature of the assessment carried out; the error introduced could be ignored. Hence the reliability of the analysis is not compromised.

The nature of assessment carried out in the Economic Module is predictive and can only be validated over time. However, a vote of confidence in the results obtained can be passed from the fact that the economic performance assessment is based on universally accepted economic evaluation techniques. Although, the economic performance results as obtained cannot be immediately verified, but like every feasibility studies, the results provides the prospective investor a forecast about the business climate.

6.10 Concluding Remarks

The outcome of the case studies indicates that having selected a gas turbine, the profitability of its use will depend on the nearness of the CO_2 throughputs to the amount of CO_2 throughput that initially guided its selection. Also, the economic performance of the GT is more favourable with higher CO_2 throughput ab initio because very high positive NPV and low PBP is obtained. Giving the pipeline sizes under the assumed flow conditions, the need for a booster station across the pipeline profile may not be necessary since the pressure at delivery is still within the set limit. Additional conclusions and recommendation is provided in the next chapter to conclude this study.

7 Conclusion and Recommendation

7.1 Conclusion

The primary objective of this doctoral thesis is to assess CO_2 return pipelines and the close coupling of the compression system with advanced prime mover cycles. This investigation provides a comparative assessment of traditional and novel prime mover options including the design and off-design performance of the engine as well as the economic analysis of the system. The originality being the technical and economic optimisation of GTs based on current and novel cycles for a novel pipeline application in a wide range of operating conditions. To sum up, the following were presented:

a. Modelling and evaluation of CO_2 compression power requirements for GT driven equipment (pump and compressor). The results show that the modelled CO_2 compression power increase as the CO_2 throughput increases. In compression using multi-stage compressors, reducing the number of stages cause an increase in the compressor duty for an average CO_2 throughput due to rise in compressor head or otherwise rise in stage compression ratio. Although the increase in required power means increase in energy cost, it is a trade-off for a compact and relative light weight centrifugal compressor which is advantageous during installation and maintenance. In the same vein, the effect of pressure drop due to intercooling during compression is an increase in compression power requirement. The modelled results also show that, in a combination of compressor and pump to compress the CO_2 to 150 bar, 15.2% power saving is achieved if the CO_2 is introduced into the pump at a pressure and temperature of 100 bar and 25°C respectively.

b. Subroutines implementing variable stators were developed to modify Variflow - an in-house GT simulation code. This modification was achieved by altering the existing compressor maps and main program algorithm of the code. The validation of the modified code was carried out by simulating the design point and off-design performance of a GT engine inspired by an existing industrial GT. The performance trend from the results obtained agreed well with established and published GT behavior that employs variable stators. The code is especially important in simulating GT cycles using working fluids other than air.

c. Bearing in mind that captured CO_2 throughput vary depending on the utilization factor of power generation plants at specific time of the year, two case scenarios were analysed over a 25-year projected CO_2 throughput spread across four seasons in a year for a typical tropical climate. The first case scenario involves the application of GT in providing the initial compression duty to compress the seasonal CO_2 throughputs from atmospheric pressure to 100 bar. The second case scenario involves the determination of suitable compression point (hence GT application) along a real life pipeline profile for the seasonal CO_2 throughputs in three different sizes of pipe.

   Economic model based on NPV method and pipeline hydraulic analysis model based on fundamental gas flow equation were developed in FORTRAN to facilitate the techno-economic and environmental risk analysis of selected GT mechanical drives in the case scenarios. In order to provide the required compression duty, two GTs of which one incorporates a recuperator (9.4 MW capacity) while the other utilizes a dry low emission combustor (33.9 MW capacity) were modelled. The design point and off-design performances of the GT were simulated using Turbomatch and analysed to extract necessary parameters required for the economic performance. When de-rated to the operating ambient condition, their power outputs were 7.6 MW and 25.9 MW respectively

   The economic performance of deploying the GT prime movers in the first case scenario show a very high positive net present value (NPV) in as much as the available CO_2 throughput is maintained as close as possible to the rated throughput that guided the selection of the prime movers at onset. Thus, a good return on investment is expected. Similarly, at rated CO_2 throughput, the associated risk is quite low with a payback period within 3 - 7 years. However, as the CO_2 throughput drops, the risk becomes higher with less return on investment. In fact, when the CO_2 throughput drops to a certain level, the investment becomes highly unattractive and unable to payback itself within the assumed 25 years plant life.

Reduction in discount rate positively influence the investment, which suggests that government policy aimed at interest rate reduction will create a healthy economic climate to prospective investors. The results equally show that cut in expenditure in the form of fuel cost reduction brings tremendous increase in NPV as well as reduction in the payback period assuming the selling price of electricity remain unchanged.

The results of the second case scenario has shown that for a given pipeline, the amount of pressure drop is directly proportional to the magnitude of the CO_2 throughput and hence the pressure distribution along the pipeline. On the other hand, the amount of pressure drop is inversely proportional to the pipe size. Given the assumed pipeline operating pressure of 15 MPa and the minimum 10 MPa threshold for recompression; considering the worst case (i.e. least pipe size and maximum CO_2 throughput) of this study along the given pipeline profile, the outlet pressure obtained at the receiving terminal is higher than the pipeline inlet pressure. In fact, pressure spikes above the pipeline maximum allowable operating pressure were obtained putting to test the pipeline's mechanical integrity and safety. Thus, the effect of the profile's elevation is to minimise the pressure losses and to a very large extent increase the pressure of the CO_2 flow during transmission.

d. Further investigation revealed that maintaining the pipeline inlet pressure at 10.5 MPa along the study's pipeline profile gives a discharge pressure of about 10.45 MPa. It is worthy of note that at this pressure, the flow is not only kept within safety limits but huge savings in energy is achieved since it is no longer necessary to compress to 15 MPa. Simulating the same flow over a flat profile of same distance and pipeline inlet condition; the resulting drop in pressure is about 142 KPa. This implies that the flow pressure will not fall below the 10 MPa specified minimum were it to be transported along thrice the current pipeline distance. Therefore, within the limits of the current study, no booster station is required. This equally eliminates the need of combining a compressor and pump for the initial pressure boost as widely advocated especially as the difference in pressure is no longer 5 MPa but 0.5 MPa. This is because irrespective of the saving in energy, the increase in capital cost associated with obtaining a pump and suitable driver far outweighs the extra expense incurred in acquiring a rated GT mechanical drive to meet the compression duty.

Under the stated assumption for the CO_2 pipeline transmission, it is very unlikely that a booster station will be required especially when a real life profile is considered; except for the situation where the pipeline profile climbs uphill for some considerable distance which is rare. Therefore, this will limit the use of GT mechanical drives to the initial pressure boost to pipeline operating pressure leading to reduced investment cost.

7.2 Recommendation for Further Work

In the course of this analysis, pure CO_2 has been assumed in simulating the compression power requirement. It will be worthwhile to perform similar analysis with CO_2 containing known impurities in order to ascertain the impact on the compression power requirement and selected GT power.

The Variflow code as currently modified and validated can be used to predict the off-design performance of single-shaft GT. However, further improvement will be necessary especially in the handling to enable the simulation and performance analysis of the semi-closed cycle GT that is currently being researched. It is envisaged that this modified Brayton cycle employing oxy-fuel combustion process in a novel application of GT, adds another dimension to the technological option in CO_2 capture from power generation plants. The main essence of which is to take advantage of the high compression ratio of the GT axial compressor to compress the CO_2 to some appreciable level before being introduced into the CO_2 compressor. Thus, the size and power requirement of the CO_2 pipeline compressor will be highly reduced.

Furthermore, the extraction of CO_2 from the GT compressor will naturally results in an inefficient engine. However, its ability to burn cheap fuel like syngas from coal; the reduction in CO_2 compression power requirement and the gains in the efficient control of CO_2 emission are trade-offs that are worth exploring.

Finally, hydraulic analysis of other real life pipeline profiles should be conducted to ascertain whether or not CO_2 recompression will be required in the CO_2 pipeline transport. Such findings will in no small measure empower prospective investors and power generation plant owners to engage government constructively in coming up with appropriate legislation to implement CCS.

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APPENDICES

Appendix A Variflow Design Point Output File

```
VARIFLOW SIMULATION OF EL200 INSPIRED BY SGT200 INDUSTRIAL GAS TURBINE
BY A. ELSULEIMAN, OCT 2014
            D E S I G N   P O I N T   S U M M A R Y

               T              P              W
          1   288.1500       101325.0       29.03000
          2   288.1500       101314.9       29.03000
          3   288.1500       101314.9       29.03000
          4   288.1500       101314.9       29.03000
          5   288.1500       101314.9       29.03000
          6   288.1500       101314.9       29.03000
          7   288.1500       101314.9       29.03000
          8   288.1500       101314.9       29.03000
          9   629.3137       1276567.       29.03000
          10   1326.150      1206356.       26.88995
         11   1326.150       1206356.       26.88995
         12   776.3839       105649.2       26.88995
         13   764.0139       105649.2       29.50265
         14   764.0139       105649.2       29.50265

  INTPLOSS=.000    CPOLY=.908    CBLF=.090    OBLF=.000    CCPLOSS=.055
TPOLY=.911

  WFFI=1                                       LHV= 48.172 MJ/kg
  P1=101325.0 Pa   T1=288.15 K    W1=  29.030 kg/sec
  CPR1= 1.00       CPR2= 1.00     CPR3= 1.00     CPR4=12.60
  TET=1326.15 K    M14=.250
  ADN1=  8.80          ADN2=  8.80        ADN3=  8.80        ADN4=  8.80
(RPM*sec/m)  N=3000.000 RPM
POWER=  7.68 MW   ETATH=33.74%    SP.PWR=0.265 MW*sec/kg    HR=10113.20
BTU/kWhr

CFR  CF(kg/sec)  SFC(kg/MWhr)
CH2  0.00000  0.000    0.000      M2=.450              ADW2=0.0698 m^2
CH4  0.01543  0.408    191.043      A2=0.1748 m^2    A11=0.0208 m^2
C2H6 0.00000  0.000      0.000      M91=.100         ADW4=0.0698 m^2
C3H8 0.00194  0.051     24.021      A91=0.0757 m^2      P10/P12=11.419
 CO  0.00000  0.000      0.000      CCPLF= 8.02         P14/P1=1.0427
CO2  0.00009  0.002      1.115      EGT= 764.01 K
H2   0.00000  0.000      0.000      PHITRB=0.00      W14= 29.503 kg/sec
```

```
H2O  0.00000  0.000      0.000      PSITRB=1.70      A14= 0.4715 m^2
N2   0.00043  0.011      5.323
O2   0.00000  0.000      0.000
TOT  0.01789  0.473    221.503
 SFADN1=   0.9948651      SFADN2=   0.9948651      SFADN3=   0.9948651
SFADN4=  0.9948651
 SFADW1=   0.5884801      SFADW2=   0.5884801      SFADW3=   0.5884801
SFADW4=  0.5884801
```

Appendix B Turbomatch Design Point Input Files

B.1 EL2500RD Gas Turbine Design Point Input File

```
REF_TITLE: Performance simulation modelling of EL2500RD inspired by GE
LM2500RD using the TURBOMATCH scheme
BY A. EL-SULEIMAN
ADAPTED AND MODIFIED FROM THE MODEL by Paul H. Wilkinson
REF_DATE: OCT 2014
```

```
!Turbomatch Programme: DESIGN POINT SIMULATION OF EL2500RD INDUSTRIAL
GAS TURBINE DLE ENGINE

////
DP SI KE VA XP
-1
-1
INTAKE  S1,2      D1-4           R100
COMPRE  S2,3      D5-11          R102     V5    V6
PREMAS  S3,4,22   D12-15
PREMAS  S4,5,23   D16-19
BURNER  S5,6      D20-22         R104
MIXEES  S6,23,7
TURBIN  S7,8      D23-30,102,31           V24
MIXEES  S8,22,9
TURBIN  S9,10     D32-41                  V32 V33
NOZCON  S10,11,1  D41            R110
PERFOR  S1,0,0    D32,43-45,110,100,104,0,0,0,0,0,0,0,0,0
CODEND

DATA ITEMS ////
1 0.0                 ! INTAKE DATA : ALTITUDE
2 0.0                 ! DEV FROM STANDART TEMP
3 0.0                 ! MA-NUMBER
4 0.999               ! PRESSURE RECOVERY
! COMPRESSOR
5 -1.0                ! COMP : Z
6 -1.0                ! RELATIVE ROTATIONAL SPEED
7 23.0                ! PRESSURE RATIO
8 0.89                ! ISENTROPIC EFFICIENCY0.87
9 0.0                 ! ERROR SWITCH
10 3.0                ! MAP-NUMBER
11 0.0                ! STATOR ANGLE RELATIVE TO DP
!PREMAS
```

```
12 0.975                !Cooling bypass: LAMBDA W (BLEED AIR)0.975
13 0.0                  ! DELTA W FLOW LOSS
14 1.0                  ! LAMBDA P PRESSURE RECOVERY
15 0.0                  ! DELTP PRESSURE DROP
!PREMAS
16 0.927                ! Cooling bypass: LAMBDA W (BLEED AIR)
17 0.0                  ! DELTA W FLOW LOSS
18 1.0                  ! LAMBDA P PRESSURE RECOVERY
19 0.0                  ! DELTP PRESSURE DROP
!BURNER
20 0.05                 ! PRESSURE LOSS DP/P
21 0.997                ! COMB. EFF.
22 -1.0                 ! FUEL FLOW
!HP TURBINE
23 0.0                  ! AUXWORK
24 -1.0                  ! DESIGN NON DIM FLOW / MAX
25 0.8                 ! DESIGN NON DIM SPEED
26 0.89                 ! ISENTROPIC EFF
27 -1.0                 ! ROT SPEED OF PT
28 1.0                  ! NUMBER OF COMPRESSOR DRIVEN
29 5.0                  ! MAP NUMBER
30 -1.0                 ! POWER LAW INDEX
31  0.0                 ! NGV ANGLE RELATIVE TO DP
!POWER TURBINE
32 33897494.0           ! AUXWORK
33 -1.0                 ! DESIGN NON DIM FLOW / MAX
34 -1.0                 ! DESIGN NON DIM SPEED
35 0.89                 ! ISENTROPIC EFF
36 1.0                  ! ROT SPEED OF PT
37 0.0                  ! NUMBER OF COMPRESSOR DRIVEN
38 5.0                  ! MAP NUMBER
39 3.0                 ! POWER LAW INDEX
40 -1.0                 ! COMWORK
41  0.0                 ! NGV ANGLE RELATIVE TO DP
!NOZCON
42 -1.0                 ! FIXED CONVERGENT NOZZLE(THROAT AREA)
!PERFOR
43 1.0                  ! PROPELLER EFF
44 0.0                  ! SCALING SWITCH
45 0.0                  ! REQUIRED THRUST at Design point

-1
1 2 91.17               ! item 2 at station 1 (INLET MASS FLOW)
6 6 1550.0              ! Item 6 at station 6 TET OR COT
```

```
-1
-3
```

B.2 EL1200R Design Point Input File

```
REF_TITLE: Performance simulation modelling of EL1200-R inspired by
MAN Turbo THM1304-10R using the TURBOMATCH scheme
         BY A. EL-SULEIMAN
REF_DATE: OCT 2014
```

```
!Turbomatch Programme: DESIGN POINT SIMULATION OF EL1200-R RECUPERATED
INDUSTRIAL GAS TURBINE

////
DP SI KE VA XP
-1
-1
INTAKE  S1,2       D1-4             R100
COMPRE  S2,3       D5-11            R102      V5    V6
PREMAS  S3,4,22    D12-15
HETCOL  S4,5       D16-19
BURNER  S5,6       D20-22           R104
MIXEES  S6,22,7
TURBIN  S7,8       D23-30,102,31              V24
TURBIN  S8,9       D32-41                     V32 V33
HETHOT  S4,9,10    D42-45
NOZCON  S10,11,1   D46              R110
PERFOR  S1,0,0     D32,47-49,110,100,104,0,0,0,0,0,0,0,0,0
CODEND

DATA ITEMS ////
1 0.0                ! INTAKE DATA : ALTITUDE
2 0.0                ! DEV FROM STANDART TEMP
3 0.0                ! MA-NUMBER
4 0.999              ! PRESSURE RECOVERY
! COMPRESSOR
5 -1.0               ! COMP : Z
6 -1.0               ! RELATIVE ROTATIONAL SPEED
7 10.0               ! PRESSURE RATIO
8 0.88               ! ISENTROPIC EFFICIENCY
9 0.0                ! ERROR SWITCH
10 5.0               ! MAP-NUMBER
11 0.0               ! STATOR ANGLE RELATIVE TO DP
!PREMAS
12 0.975             !Cooling bypass: LAMBDA W (BLEED AIR)
```

```
13 0.0                ! DELTA W FLOW LOSS
14 1.0                ! LAMBDA P PRESSURE RECOVERY
15 0.0                ! DELTP PRESSURE DROP
!HETCOL
16 0.0                ! COLD SIDE TOTAL PRES. LOSS/COLD SIDE INLET
TOTAL PRES(DP/Pin)
17 0.8                ! EFFECTIVENESS
18 1.0                ! HEAT EXCHANGER TYPE-RECUPERATOR (SWITCH)
19 0.0                ! MASS FLOW LEAKAGE (COLD SIDE TO HOT SIDE/COLD
SIDE INLET MASS FLOW=DW/DWin)
!BURNER
20 0.05               ! PRESSURE LOSS DP/P
21 0.999              ! COMB. EFF.
22 -1.0               ! FUEL FLOW
!HP TURBINE
23 0.0                ! AUXWORK
24 -1.0               ! DESIGN NON DIM FLOW / MAX
25 0.8               ! DESIGN NON DIM SPEED
26 0.89               ! ISENTROPIC EFF
27 -1.0               ! ROT SPEED OF PT
28 1.0                ! NUMBER OF COMPRESSOR DRIVEN
29 6.0                ! MAP NUMBER
30 -1.0               ! POWER LAW INDEX
31  0.0               ! NGV ANGLE RELATIVE TO DP
!POWER TURBINE
32 9404076.0         ! AUXWORK
33 -1.0               ! DESIGN NON DIM FLOW / MAX
34 -1.0               ! DESIGN NON DIM SPEED
35 0.89               ! ISENTROPIC EFF
36 1.0                ! ROT SPEED OF PT
37 0.0                ! NUMBER OF COMPRESSOR DRIVEN
38 6.0                ! MAP NUMBER
39 3.0               ! POWER LAW INDEX
40 -1.0               ! COMWORK
41  0.0               ! NGV ANGLE RELATIVE TO DP
!HETHOT
42 0.0                ! HOT SIDE TOTAL PRES. LOSS/HOT SIDE INLET TOTAL
PRES(DP/Pin)
43 0.8                ! EFFECTIVENESS
44 1.0                ! HEAT EXCHANGER TYPE-RECUPERATOR (SWITCH)
45 0.0                ! MASS FLOW LEAKAGE (COLD SIDE TO HOT SIDE/COLD
SIDE INLET MASS FLOW=DW/DWin)
!NOZCON
46 -1.0               ! FIXED CONVERGENT NOZZLE(THROAT AREA)
!PERFOR
```

```
47 1.0                ! PROPELLER EFF
48 0.0                ! SCALING SWITCH
49 0.0                ! REQUIRED THRUST at Design point

-1
1 2 45.36             ! item 2 at station 1 (INLET MASS FLOW)
6 6 1200.0            ! Item 6 at station 6 TET OR COT
-1
-3
```

Appendix C Turbomatch Off-Design Performance Input Files

C.1 EL2500RD Off- Design Performance Input File

```
REF_TITLE: Performance simulation modelling of EL2500RD inspired by GE LM2500RD
using the TURBOMATCH scheme
          BY A. EL-SULEIMAN
          ADAPTED AND MODIFIED FROM THE MODEL by Paul H. Wilkinson
REF_DATE: OCT 2014
______________________________________________________________________
!Turbomatch Programme: OFF DESIGN POINT SIMULATION OF EL2500RD INDUSTRIAL GAS
TURBINE DLE ENGINE

////
OD SI KE VA XP
-1
-1
INTAKE  S1,2       D1-4            R100
COMPRE  S2,3       D5-11           R102     V5    V6
PREMAS  S3,4,22    D12-15
PREMAS  S4,5,23    D16-19
BURNER  S5,6       D20-22          R104
MIXEES  S6,23,7
TURBIN  S7,8       D23-30,102,31            V24
MIXEES  S8,22,9
TURBIN  S9,10      D32-41                   V32 V33
NOZCON  S10,11,1   D41             R110
PERFOR  S1,0,0     D32,43-45,110,100,104,0,0,0,0,0,0,0,0,0
CODEND

DATA ITEMS ////
1 0.0                ! INTAKE DATA : ALTITUDE
2 0.0                ! DEV FROM STANDART TEMP
3 0.0                ! MA-NUMBER
4 0.999              ! PRESSURE RECOVERY
! COMPRESSOR
5 -1.0               ! COMP : Z
6 -1.0               ! RELATIVE ROTATIONAL SPEED
7 23.0               ! PRESSURE RATIO
8 0.89               ! ISENTROPIC EFFICIENCY0.87
9 0.0                ! ERROR SWITCH
10 3.0               ! MAP-NUMBER
11 0.0               ! STATOR ANGLE RELATIVE TO DP
!PREMAS
12 0.975             !Cooling bypass: LAMBDA W (BLEED AIR)0.975
13 0.0               ! DELTA W FLOW LOSS
14 1.0               ! LAMBDA P PRESSURE RECOVERY
15 0.0               ! DELTP PRESSURE DROP
!PREMAS
16 0.927             ! Cooling bypass: LAMBDA W (BLEED AIR)
17 0.0               ! DELTA W FLOW LOSS
18 1.0               ! LAMBDA P PRESSURE RECOVERY
19 0.0               ! DELTP PRESSURE DROP
!BURNER
20 0.05              ! PRESSURE LOSS DP/P
21 0.997             ! COMB. EFF.
22 -1.0              ! FUEL FLOW
```

```
!HP TURBINE
23 0.0               ! AUXWORK
24 -1.0               ! DESIGN NON DIM FLOW / MAX
25 0.8              ! DESIGN NON DIM SPEED
26 0.89              ! ISENTROPIC EFF
27 -1.0              ! ROT SPEED OF PT
28 1.0               ! NUMBER OF COMPRESSOR DRIVEN
29 5.0               ! MAP NUMBER
30 -1.0              ! POWER LAW INDEX
31  0.0              ! NGV ANGLE RELATIVE TO DP
!POWER TURBINE
32 33897494.0        ! AUXWORK
33 -1.0              ! DESIGN NON DIM FLOW / MAX
34 -1.0              ! DESIGN NON DIM SPEED
35 0.89              ! ISENTROPIC EFF
36 1.0               ! ROT SPEED OF PT
37 0.0               ! NUMBER OF COMPRESSOR DRIVEN
38 5.0               ! MAP NUMBER
39 3.0              ! POWER LAW INDEX
40 -1.0              ! COMWORK
41  0.0              ! NGV ANGLE RELATIVE TO DP
!NOZCON
42 -1.0              ! FIXED CONVERGENT NOZZLE(THROAT AREA)
!PERFOR
43 1.0               ! PROPELLER EFF
44 0.0               ! SCALING SWITCH
45 0.0               ! REQUIRED THRUST at Design point

-1
1 2 91.17            ! item 2 at station 1 (INLET MASS FLOW)
6 6 1550.0           ! Item 6 at station 6 TET OR COT
-1
1  100.0
2   -10.0              ! OD Calculation; DT = -10, CHANGE IN AMB TEMP WRT ISA
(ISA = 15 DEG C)
-1
6  6  1650.0           ! DT = -10 ; TET = 1650
-1
-1
6  6  1600.0           ! DT = -10 ; TET = 1600
-1
-1
6  6  1550.0           ! DT = -10 ; TET = 1550
-1
-1
6  6  1500.0           ! DT = -10 ; TET = 1500
-1
-1
6  6  1450.0           ! DT = -10 ; TET = 1450
-1
-1
6  6  1400.0           ! DT = -10 ; TET = 1400
-1
-1
6  6  1350.0           ! DT = -10 ; TET = 1350
-1
-1
6  6  1300.0           ! DT = -10 ; TET = 1300
-1
-1
```

```
6  6  1250.0           ! DT = -10 ; TET = 1250
-1
-1
6  6  1200.0           ! DT = -10 ; TET = 1200
-1
-1
6  6  1150.0           ! DT = -10 ; TET = 1150
-1
-1
6  6  1100.0           ! DT = -10 ; TET = 1100
-1
-1
6  6  1050.0           ! DT = -10 ; TET = 1050
-1
1  100.0
2   -5.0              ! OD Calculation; DT = -5, CHANGE IN AMB TEMP WRT ISA (ISA
= 15 DEG C)
-1
6  6  1050.0           ! DT = -5 ; TET = 1050
-1
-1
6  6  1100.0           ! DT = -5 ; TET = 1100
-1
-1
6  6  1150.0           ! DT = -5 ; TET = 1150
-1
-1
6  6  1200.0           ! DT = -5 ; TET = 1200
-1
-1
6  6  1250.0           ! DT = -5 ; TET = 1250
-1
-1
6  6  1300.0           ! DT = -5 ; TET = 1300
-1
-1
6  6  1350.0           ! DT = -5 ; TET = 1350
-1
-1
6  6  1400.0           ! DT = -5 ; TET = 1400
-1
-1
6  6  1450.0           ! DT = -5 ; TET = 1450
-1
-1
6  6  1500.0           ! DT = -5 ; TET = 1500
-1
-1
6  6  1550.0           ! DT = -5 ; TET = 1550
-1
-1
6  6  1600.0           ! DT = -5 ; TET = 1600
-1
-1
6  6  1650.0           ! DT = -5 ; TET = 1650
-1
1  100.0
2   0.0              ! OD Calculation; DT = 0, CHANGE IN AMB TEMP WRT ISA (ISA =
15 DEG C)
-1
```

```
6  6  1650.0           ! DT = 0 ; TET = 1650
-1
-1
6  6  1600.0           ! DT = 0 ; TET = 1600
-1
-1
6  6  1550.0           ! DT = 0 ; TET = 1550
-1
-1
6  6  1500.0           ! DT = 0 ; TET = 1500
-1
-1
6  6  1450.0           ! DT = 0 ; TET = 1450
-1
-1
6  6  1400.0           ! DT = 0 ; TET = 1400
-1
-1
6  6  1350.0           ! DT = 0 ; TET = 1350
-1
-1
6  6  1300.0           ! DT = 0 ; TET = 1300
-1
-1
6  6  1250.0           ! DT = 0 ; TET = 1250
-1
-1
6  6  1200.0           ! DT = 0 ; TET = 1200
-1
-1
6  6  1150.0           ! DT = 0 ; TET = 1150
-1
-1
6  6  1100.0           ! DT = 0 ; TET = 1100
-1
-1
6  6  1050.0           ! DT = 0 ; TET = 1050
-1
1  100.0
2   5.0              ! OD Calculation; DT = 5, CHANGE IN AMB TEMP WRT ISA (ISA =
15 DEG C)
-1
6  6  1150.0           ! DT = 5 ; TET = 1150
-1
-1
6  6  1200.0           ! DT = 5 ; TET = 1200
-1
-1
6  6  1250.0           ! DT = 5 ; TET = 1250
-1
-1
6  6  1300.0           ! DT = 5 ; TET = 1300
-1
-1
6  6  1350.0           ! DT = 5 ; TET = 1350
-1
-1
6  6  1400.0           ! DT = 5 ; TET = 1400
-1
-1
```

```
6  6  1450.0           ! DT = 5 ; TET = 1450
-1
-1
6  6  1500.0           ! DT = 5 ; TET = 1500
-1
-1
6  6  1550.0           ! DT = 5 ; TET = 1550
-1
-1
6  6  1600.0           ! DT = 5 ; TET = 1600
-1
-1
6  6  1650.0           ! DT = 5 ; TET = 1650
-1
1  100.0
2   10.0              ! OD Calculation; DT = 10, CHANGE IN AMB TEMP WRT ISA (ISA
= 15 DEG C)
-1
6  6  1650.0           ! DT = 10 ; TET = 1650
-1
-1
6  6  1600.0           ! DT = 10 ; TET = 1600
-1
-1
6  6  1550.0           ! DT = 10 ; TET = 1550
-1
-1
6  6  1500.0           ! DT = 10 ; TET = 1500
-1
-1
6  6  1450.0           ! DT = 10 ; TET = 1450
-1
-1
6  6  1400.0           ! DT = 10 ; TET = 1400
-1
-1
6  6  1350.0           ! DT = 10 ; TET = 1350
-1
-1
6  6  1300.0           ! DT = 10 ; TET = 1300
-1
-1
6  6  1250.0           ! DT = 10 ; TET = 1250
-1
-1
6  6  1200.0           ! DT = 10 ; TET = 1200
-1
-1
6  6  1150.0           ! DT = 10 ; TET = 1150
-1
-1
6  6  1100.0           ! DT = 10 ; TET = 1100
-1
1  100.0
2   15.0              ! OD Calculation; DT = 15, CHANGE IN AMB TEMP WRT ISA (ISA
= 15 DEG C)
-1
6  6  1150.0           ! DT = 15 ; TET = 1150
-1
-1
```

```
6  6  1200.0           ! DT = 15 ; TET = 1200
-1
-1
6  6  1250.0           ! DT = 15 ; TET = 1250
-1
-1
6  6  1300.0           ! DT = 15 ; TET = 1300
-1
-1
6  6  1350.0           ! DT = 15 ; TET = 1350
-1
-1
6  6  1400.0           ! DT = 15 ; TET = 1400
-1
-1
6  6  1450.0           ! DT = 15 ; TET = 1450
-1
-1
6  6  1500.0           ! DT = 15 ; TET = 1500
-1
-1
6  6  1550.0           ! DT = 15 ; TET = 1550
-1
-1
6  6  1600.0           ! DT = 15 ; TET = 1600
-1
-1
6  6  1650.0           ! DT = 15 ; TET = 1650
-1
1  100.0
2   20.0              ! OD Calculation; DT = 20, CHANGE IN AMB TEMP WRT ISA (ISA
= 15 DEG C)
-1
6  6  1650.0           ! DT = 20 ; TET = 1650
-1
-1
6  6  1600.0           ! DT = 20 ; TET = 1600
-1
-1
6  6  1550.0           ! DT = 20 ; TET = 1550
-1
-1
6  6  1500.0           ! DT = 20 ; TET = 1500
-1
-1
6  6  1450.0           ! DT = 20 ; TET = 1450
-1
-1
6  6  1400.0           ! DT = 20 ; TET = 1400
-1
-1
6  6  1350.0           ! DT = 20 ; TET = 1350
-1
-1
6  6  1300.0           ! DT = 20 ; TET = 1300
-1
-1
6  6  1250.0           ! DT = 20 ; TET = 1250
-1
-1
```

```
6  6  1200.0           ! DT = 20 ; TET = 1200
-1
-1
6  6  1150.0           ! DT = 20 ; TET = 1150
-1
-1
6  6  1100.0           ! DT = 20 ; TET = 1100
-1
1  100.0
2   23.0              ! OD Calculation; DT = 23, CHANGE IN AMB TEMP WRT ISA (ISA
= 15 DEG C)
-1
6  6  1150.0           ! DT = 23 ; TET = 1150
-1
-1
6  6  1200.0           ! DT = 23 ; TET = 1200
-1
-1
6  6  1250.0           ! DT = 23 ; TET = 1250
-1
-1
6  6  1300.0           ! DT = 23 ; TET = 1300
-1
-1
6  6  1350.0           ! DT = 23 ; TET = 1350
-1
-1
6  6  1400.0           ! DT = 23 ; TET = 1400
-1
-1
6  6  1450.0           ! DT = 23 ; TET = 1450
-1
-1
6  6  1500.0           ! DT = 23 ; TET = 1500
-1
-1
6  6  1550.0           ! DT = 23 ; TET = 1550
-1
-1
6  6  1600.0           ! DT = 23 ; TET = 1600
-1
-1
6  6  1650.0           ! DT = 23 ; TET = 1650
-1
1  100.0
2   25.0              ! OD Calculation; DT = 25, CHANGE IN AMB TEMP WRT ISA (ISA
= 15 DEG C)
-1
6  6  1650.0           ! DT = 25 ; TET = 1650
-1
-1
6  6  1600.0           ! DT = 25 ; TET = 1600
-1
-1
6  6  1550.0           ! DT = 25 ; TET = 1550
-1
-1
6  6  1500.0           ! DT = 25 ; TET = 1500
-1
-1
```

```
6  6  1450.0           ! DT = 25 ; TET = 1450
-1
-1
6  6  1400.0           ! DT = 25 ; TET = 1400
-1
-1
6  6  1350.0           ! DT = 25 ; TET = 1350
-1
-1
6  6  1300.0           ! DT = 25 ; TET = 1300
-1
-1
6  6  1250.0           ! DT = 25 ; TET = 1250
-1
-1
6  6  1200.0           ! DT = 25 ; TET = 1200
-1
-1
6  6  1150.0           ! DT = 25 ; TET = 1150
-1
-3
```

C.2 EL1200R Off-Design Performance Input File

```
REF_TITLE: Performance simulation modelling of EL1200-R inspired by MAN Turbo
THM1304-10R using the TURBOMATCH scheme
          BY A. EL-SULEIMAN
REF_DATE: OCT 2014
_______________________________________________________________________
!Turbomatch Programme: OFF DESIGN SIMULATION OF EL1200-R RECUPERATED INDUSTRIAL
GAS TURBINE

////
OD SI KE VA XP
-1
-1
INTAKE  S1,2       D1-4            R100
COMPRE  S2,3       D5-11           R102     V5    V6
PREMAS  S3,4,22    D12-15
HETCOL  S4,5       D16-19
BURNER  S5,6       D20-22          R104
MIXEES  S6,22,7
TURBIN  S7,8       D23-30,102,31            V24
TURBIN  S8,9       D32-41                   V32 V33
HETHOT  S4,9,10    D42-45
NOZCON  S10,11,1   D46             R110
PERFOR  S1,0,0     D32,47-49,110,100,104,0,0,0,0,0,0,0,0,0
CODEND

DATA ITEMS ////
1 0.0                ! INTAKE DATA : ALTITUDE
2 0.0                ! DEV FROM STANDART TEMP
3 0.0                ! MA-NUMBER
4 0.999              ! PRESSURE RECOVERY
! COMPRESSOR
5 -1.0               ! COMP : Z
6 -1.0               ! RELATIVE ROTATIONAL SPEED
7 10.0               ! PRESSURE RATIO
```

```
8 0.88                ! ISENTROPIC EFFICIENCY
9 0.0                 ! ERROR SWITCH
10 5.0                ! MAP-NUMBER
11 0.0                ! STATOR ANGLE RELATIVE TO DP
!PREMAS
12 0.975              !Cooling bypass: LAMBDA W (BLEED AIR)
13 0.0                ! DELTA W FLOW LOSS
14 1.0                ! LAMBDA P PRESSURE RECOVERY
15 0.0                ! DELTP PRESSURE DROP
!HETCOL
16 0.0              !
17 0.8                !
18 1.0                !
19 0.0                !
!BURNER
20 0.05               ! PRESSURE LOSS DP/P
21 0.999              ! COMB. EFF.
22 -1.0               ! FUEL FLOW
!HP TURBINE
23 0.0                ! AUXWORK
24 -1.0                ! DESIGN NON DIM FLOW / MAX
25 0.8               ! DESIGN NON DIM SPEED
26 0.89               ! ISENTROPIC EFF
27 -1.0               ! ROT SPEED OF PT
28 1.0                ! NUMBER OF COMPRESSOR DRIVEN
29 5.0                ! MAP NUMBER
30 -1.0               ! POWER LAW INDEX
31  0.0               ! NGV ANGLE RELATIVE TO DP
!POWER TURBINE
32 9404076.0         ! AUXWORK
33 -1.0               ! DESIGN NON DIM FLOW / MAX
34 -1.0               ! DESIGN NON DIM SPEED
35 0.89               ! ISENTROPIC EFF
36 1.0                ! ROT SPEED OF PT
37 0.0                ! NUMBER OF COMPRESSOR DRIVEN
38 5.0                ! MAP NUMBER
39 3.0               ! POWER LAW INDEX
40 -1.0               ! COMWORK
41  0.0               ! NGV ANGLE RELATIVE TO DP
!HETHOT
42 0.0              !
43 0.8                !
44 1.0                !
45 0.0                !
!NOZCON
46 -1.0               ! FIXED CONVERGENT NOZZLE(THROAT AREA)
!PERFOR
47 1.0                ! PROPELLER EFF
48 0.0                ! SCALING SWITCH
49 0.0                ! REQUIRED THRUST at Design point

-1
1 2 45.36             ! item 2 at station 1 (INLET MASS FLOW)
6 6 1200.0            ! Item 6 at station 6 TET OR COT
-1
1  100.0
2   -10.0               ! OD Calculation; DT = -10, CHANGE IN AMB TEMP WRT ISA
(ISA = 15 DEG C)
-1
6  6  1200.0            ! DT = -10 ; TET = 1200
```

```
-1
-1
6  6  1150.0           ! DT = -10 ; TET = 1150
-1
-1
6  6  1100.0           ! DT = -10 ; TET = 1100
-1
-1
6  6  1050.0           ! DT = -10 ; TET = 1050
-1
-1
6  6  1000.0           ! DT = -10 ; TET = 1000
-1
-1
6  6  950.0           ! DT = -10 ; TET = 950
-1
-1
6  6  900.0           ! DT = -10 ; TET = 900
-1
-1
6  6  850.0           ! DT = -10 ; TET = 850
-1
-1
6  6  800.0           ! DT = -10 ; TET = 800
-1
-1
6  6  750.0           ! DT = -10 ; TET = 750
-1
1  100.0
2   -5.0              ! OD Calculation; DT = -5, CHANGE IN AMB TEMP WRT ISA (ISA
= 15 DEG C)
-1
6  6  750.0           ! DT = -5 ; TET = 750
-1
-1
6  6  800.0           ! DT = -5 ; TET = 800
-1
-1
6  6  850.0           ! DT = -5 ; TET = 850
-1
-1
6  6  900.0           ! DT = -5 ; TET = 900
-1
-1
6  6  950.0           ! DT = -5 ; TET = 950
-1
-1
6  6  1000.0           ! DT = -5 ; TET = 1000
-1
-1
6  6  1050.0           ! DT = -5 ; TET = 1050
-1
-1
6  6  1100.0           ! DT = -5 ; TET = 1100
-1
-1
6  6  1150.0           ! DT = -5 ; TET = 1150
-1
-1
6  6  1200.0           ! DT = -5 ; TET = 1200
```

```
-1
1  100.0
2   0.0              ! OD Calculation; DT = 0, CHANGE IN AMB TEMP WRT ISA (ISA =
15 DEG C)
-1
6  6  1200.0           ! DT = 0 ; TET = 1200
-1
-1
6  6  1150.0           ! DT = 0 ; TET = 1150
-1
-1
6  6  1100.0           ! DT = 0 ; TET = 1100
-1
-1
6  6  1050.0           ! DT = 0 ; TET = 1050
-1
-1
6  6  1000.0           ! DT = 0 ; TET = 1000
-1
-1
6  6  950.0           ! DT = 0 ; TET = 950
-1
-1
6  6  900.0           ! DT = 0 ; TET = 900
-1
-1
6  6  850.0           ! DT = 0 ; TET = 850
-1
-1
6  6  800.0           ! DT = 0 ; TET = 800
-1
1  100.0
2   5.0              ! OD Calculation; DT = 5, CHANGE IN AMB TEMP WRT ISA (ISA =
15 DEG C)
-1
6  6  800.0           ! DT = 5 ; TET = 800
-1
-1
6  6  850.0           ! DT = 5 ; TET = 850
-1
-1
6  6  900.0           ! DT = 5 ; TET = 900
-1
-1
6  6  950.0           ! DT = 5 ; TET = 950
-1
-1
6  6  1000.0           ! DT = 5 ; TET = 1000
-1
-1
6  6  1050.0           ! DT = 5 ; TET = 1050
-1
-1
6  6  1100.0           ! DT = 5 ; TET = 1100
-1
-1
6  6  1150.0           ! DT = 5 ; TET = 1150
-1
-1
6  6  1200.0           ! DT = 5 ; TET = 1200
```

```
-1
1  100.0
2   10.0               ! OD Calculation; DT = 10, CHANGE IN AMB TEMP WRT ISA (ISA
= 15 DEG C)
-1
6  6  1200.0            ! DT = 10 ; TET = 1200
-1
-1
6  6  1150.0            ! DT = 10 ; TET = 1150
-1
-1
6  6  1100.0            ! DT = 10 ; TET = 1100
-1
-1
6  6  1050.0            ! DT = 10 ; TET = 1050
-1
-1
6  6  1000.0            ! DT = 10 ; TET = 1000
-1
-1
6  6  950.0            ! DT = 10 ; TET = 950
-1
-1
6  6  900.0            ! DT = 10 ; TET = 900
-1
-1
6  6  850.0            ! DT = 10 ; TET = 850
-1
-1
6  6  800.0            ! DT = 10 ; TET = 800
-1
1  100.0
2   15.0               ! OD Calculation; DT = 15, CHANGE IN AMB TEMP WRT ISA (ISA
= 15 DEG C)
-1
6  6  800.0            ! DT = 15 ; TET = 800
-1
-1
6  6  850.0            ! DT = 15 ; TET = 850
-1
-1
6  6  900.0            ! DT = 15 ; TET = 900
-1
-1
6  6  950.0            ! DT = 15 ; TET = 950
-1
-1
6  6  1000.0            ! DT = 15 ; TET = 1000
-1
-1
6  6  1050.0            ! DT = 15 ; TET = 1050
-1
-1
6  6  1100.0            ! DT = 15 ; TET = 1100
-1
-1
6  6  1150.0            ! DT = 15 ; TET = 1150
-1
-1
6  6  1200.0            ! DT = 15 ; TET = 1200
```

```
-1
1  100.0
2   20.0              ! OD Calculation; DT = 20, CHANGE IN AMB TEMP WRT ISA (ISA
= 15 DEG C)
-1
6  6  1200.0           ! DT = 20 ; TET = 1200
-1
-1
6  6  1150.0           ! DT = 20 ; TET = 1150
-1
-1
6  6  1100.0           ! DT = 20 ; TET = 1100
-1
-1
6  6  1050.0           ! DT = 20 ; TET = 1050
-1
-1
6  6  1000.0           ! DT = 20 ; TET = 1000
-1
-1
6  6  950.0           ! DT = 20 ; TET = 950
-1
-1
6  6  900.0           ! DT = 20 ; TET = 900
-1
-1
6  6  850.0           ! DT = 20 ; TET = 850
-1
-1
6  6  800.0           ! DT = 20 ; TET = 800
-1
1  100.0
2   23.0              ! OD Calculation; DT = 23, CHANGE IN AMB TEMP WRT ISA (ISA
= 15 DEG C)
-1
6  6  800.0           ! DT = 23 ; TET = 800
-1
-1
6  6  850.0           ! DT = 23 ; TET = 850
-1
-1
6  6  900.0           ! DT = 23 ; TET = 900
-1
-1
6  6  950.0           ! DT = 23 ; TET = 950
-1
-1
6  6  1000.0           ! DT = 23 ; TET = 1000
-1
-1
6  6  1050.0           ! DT = 23 ; TET = 1050
-1
-1
6  6  1100.0           ! DT = 23 ; TET = 1100
-1
-1
6  6  1150.0           ! DT = 23 ; TET = 1150
-1
-1
6  6  1200.0           ! DT = 23 ; TET = 1200
```

```
-1
1  100.0
2   25.0             ! OD Calculation; DT = 25, CHANGE IN AMB TEMP WRT ISA (ISA
= 15 DEG C)
-1
6  6  1200.0          ! DT = 25 ; TET = 1200
-1
-1
6  6  1150.0          ! DT = 25 ; TET = 1150
-1
-1
6  6  1100.0          ! DT = 25 ; TET = 1100
-1
-1
6  6  1050.0          ! DT = 25 ; TET = 1050
-1
-1
6  6  1000.0          ! DT = 25 ; TET = 1000
-1
-1
6  6  950.0          ! DT = 25 ; TET = 950
-1
-1
6  6  900.0          ! DT = 25 ; TET = 900
-1
-1
6  6  850.0          ! DT = 25 ; TET = 850
-1
-1
6  6  800.0          ! DT = 25 ; TET = 800
-1
-3
```

Appendix D Economic Performance Code

D.1 Main Economic Program

```
  PROGRAM ECONOMICMODULE2014

  IMPLICIT NONE

! THIS CODE USED FOR THE ECONOMIC MODULE OF TERA IN APPLICATION TO CO2 PIPELINE
WAS DEVELOPED
!BY A. EL-SULEIMAN AS PART OF THE PhD THESIS IN CRANFIELD UNIVERSITY, UK- 2014.

!ASSUMPTIONS

!CASE 1 IS FOR COAL POWER GENERATION PLANT VALUES
!CASE 2 IS FOR OIL AND NATURAL GAS DUAL FIRED POWER GENERATION PLANT VALUES
!CASE 3 IS FOR CCGT POWER GENERATION PLANT VALUES
!CASE 4 IS FOR GAS TURBINE POWER GENERATION PLANT VALUES

!I = 1 = SZN1, SEASON 1 - Hot Season (March – May)
!I = 2 = SZN2, SEASON 2 - Early Rain Season (June- August)
!I = 3 = SZN3, SEASON 3 - Late Rain Season (September - November)
!I = 4 = SZN4, SEASON 4 - Harmattan Season (December- February)

!K = 1 = GAS TURBINE POWER OUTPUT REQUIRED
!K = 2 = GAS TURBINE FUEL FLOW REQUIRED FOR THE GT POWER OUTPUT

!LF1 = LOAD FACTOR FOR SEASONS 1,3 & 4 = 0.25
!LF2 = LOAD FACTOR FOR SEASON 2 = 0.15
!GT ASSUMED TOTAL LOAD FACTOR IS 0.9


! J = 1,2,3,................,25 REPRESENTS YEARS 2013,2014,.................,2037
! -----------------------------------------------------------------
! TWO GAS TURBINE MECHANICAL DRIVES WERE USED FOR THE COMPRESSION

! -----------------------------------------------------------------

    INTEGER,PARAMETER :: S1 = 12, S2 = 75, S3 = 3
    INTEGER :: I,J,K,IMAX,JMAX,KMAX,ITER,L1,L2, L3, L4, PBP
    REAL, DIMENSION(S1,S2,S3) :: SZN,SZN2,SZN3,SZN4
    REAL, DIMENSION(S1,S2) :: RVN, EXP1, EXP2,EXP3,EXP4
    REAL, DIMENSION(1:25) :: DCF, REV,REV1,REV2,REV3,REV4, EXPEND, OPINCOM,
                     PROFIT, NCF, EXPEND1, EXPEND2,EXPEND3,EXPEND4,EXPEND5
    REAL :: LF1, LF2, PCC1, PCC2, EI1, EI2, EP, CF, ANF, TXRT, LF, TDCF,NPV, NPV2
             ,P1,P2
    REAL :: SUM, SUM1, SUM2, SUM3, SUM4, SUM5, SUM6, VC1, VC2, FC1, FC2,
             ITRT,INVST1,INVST2, ADP, ETX, PL, VSTH
    CHARACTER (LEN=10) :: DUMMY
! -----------------------------------------------------------------
!    FORMATS
200    FORMAT(' ',A55,/)
210 FORMAT(I4,F13.2,F13.2,F13.2,F13.2,F13.2,F13.2,F13.2,F13.2,F13.2)
215 FORMAT(I4,F13.2,F13.2,F13.2,F13.2,F13.2,F13.2)
220 FORMAT(/'NPV(£)= ',F13.2,2X,'YEARLY CAPITAL COST SPREAD(£) = ',F10.2,2X,'PAY
BACK PERIOD = ',I2,/,'ANNUITY FACTOR = ',F10.4,2X,'DISCOUNT FACTOR =
',F10.4,2X,'FUEL PRICE(£/kg) = ',F9.2,2X,'ELECTRICITY PRICE(£) = ',F10.2) !
600 FORMAT(4(1X,F13.2))
```

```
! ------------------------------------------------------------------

    OPEN(UNIT=10,STATUS='REPLACE',FILE='PERFORMANCE_RESULTS.DAT')
    OPEN(2,FILE='COAL_POWER_GENERATION_PLANTNPV.DAT')
       OPEN(4,FILE='OILNGAS_DUAL_FIRED_POWER_GENPLANTNPV.DAT')
       OPEN(6,FILE='CCGT_POWER_GENERATION_PLANTNPV.DAT')
    OPEN(8,FILE='GT_POWER_GENERATION_PLANTNPV.DAT')

! ------------------------------------------------------------------
!      INPUT DATA AND ASSUMPTIONS
! ------------------------------------------------------------------
        IMAX = 4
        JMAX = 25
        P1 = 33.9   !EL2500RD GT NOMINALPOWER
        P2 = 9.4   !EL1200-R GT NOMINAL POWER

        PCC1 = 9615702.69   !EL2500RD GT CAPITAL COST (£) USED FOR CASES 1,2,& 3
        PCC2 = 3551167.17   !EL1200-R GT CAPITAL COST (£) USED FOR CASE 4
        LF1 = 0.25          !SEASONAL LOAD FACTOR X 3 = 0.75
        LF2 = 0.15          !SEASONAL LOAD FACTOR X 1 = 0.15
        PL = 25         !PLANT LIFE IN YEARS
        EP = 0.12       !ELECTRICITY PRICE (£/kW)
        CF = 0.09       !FUEL PRICE (£/kg)
        VC1 = 7.08       !VARIABLE COST (£/MWh) EL2500RD
        VC2 = 10.54     !VARIABLE COST (£/MWh) EL1200-R
        FC1 = 4.80       !FIXED COST(£/KW-YR) EL2500RD
        FC2 = 5.01      !FIXED COST(£/KW-YR) EL1200-R
        ITRT = 0.15      !INTEREST RATE
        TXRT = 20       !PERCENTAGE TAX RATE
        ADP  = 2.5      !ANNUAL DEPRECIATION (%)

        EI1 = 0.49     !EMISSION INDEX (kgCO2/KWh) OF EL2500RD AS PREDICTED USING
                      HEPHAESTUS
        EI2 = 0.56     !EMISSION INDEX (kgCO2/KWh) OF EL1200-R AS PREDICTED USING
                      HEPHAESTUS
        ETX = 50       !EMISSION TAX (£/tonCO2)


     ! DEFINING SOME CONSTANT VALUES

      ANF = (ITRT*(1 + ITRT)**PL)/((1 + ITRT)**PL - 1)     ! ANNUITY FACTOR

      INVST1 = ANF * PCC1         !YEARLY SPREAD OF CAPITAL COST FOR EL2500RD GT
      INVST2 = ANF * PCC2         !YEARLY SPREAD OF CAPITAL COST FOR EL1200-R GT

 !1. READING GT POWER AND FUEL FLOW FROM THE INPUT FILES
      OPEN(UNIT=5,FILE='COALPOWERPLANT.inp')

        !CHECK- READING ALL VARIABLES

        DO K=1,2
         READ(5,'(A10)'),DUMMY

        DO J=1,JMAX

            READ(UNIT=5,FMT= *) (SZN(I,J,K),I=1,IMAX)
        END DO
            END DO
```

```
    ! REVENUE EVALUATION****************************************************
        DO J = 1, JMAX

        SUM = 0
        K = 1
        DO I = 1, IMAX
        IF (I == 2) THEN
        LF = LF2
        ELSE
        LF = LF1
        END IF
    RVN(I,J) = EP * SZN(I,J,K) * LF * 8760 * 1000         !SZN IS THE REQUIRED
POWER OUTPUT
        SUM = SUM + RVN(I,J)
        END DO
    REV(J) = SUM

    ! CALCULATION OF EXPENSES***************************************************
    ! FUEL COST
      SUM1 = 0
        K = 2
        DO I = 1, IMAX
        IF (I == 2) THEN
        LF = LF2
        ELSE
        LF = LF1
        END IF
        EXP1(I,J) = CF * SZN(I,J,K) * LF * 8760 * 3600         !SZN K = 2  IS THE
GT FUEL FLOW IN KG/SEC
        SUM1 = SUM1 + EXP1(I,J)
        END DO
       EXPEND1(J) = SUM1

    ! VARIABLE O & M COST
     SUM2 = 0
        K = 1
        DO I = 1, IMAX
        IF (I == 2) THEN
        LF = LF2
        ELSE
        LF = LF1
        END IF
        EXP2(I,J) = VC1 * SZN(I,J,K) * LF * 8760          !SZN K = 1 IS THE
REQUIRED POWER OUTPUT IN MW
        SUM2 = SUM2 + EXP2(I,J)
        END DO
       EXPEND2(J) = SUM2


   ! FIXED COST

        EXPEND3(J) = FC1 * P1 * 1000

    ! DEPRECIATION COST

     EXPEND4(J) = ADP * PCC1/100

    ! EMISSION COST
        SUM4 = 0
        K = 1
```

```
        DO I = 1, IMAX
        IF (I == 2) THEN
        LF = LF2
        ELSE
        LF = LF1
        END IF

        EXP4(I,J) =  EI1 * ETX * SZN(I,J,K) * LF * 8760
         SUM4 = SUM4 + EXP4(I,J)
        END DO
        EXPEND5(J) = SUM4

    ! TOTAL EXPENITURE
        EXPEND(J)= EXPEND1(J) + EXPEND2(J) + EXPEND3(J) + EXPEND4(J) + EXPEND5(J)
! TOTAL EXPENDITURE

      ! CASH FLOW
        OPINCOM(J) = REV(J) - EXPEND(J) ! OPERATING INCOME

        PROFIT(J) = OPINCOM(J) - INVST1 !PROFIT

        NCF(J) = PROFIT(J)*(1 - TXRT/100) - INVST1  ! NET CASH FLOW
        END DO

    ! DISCOUNTING THE CASH FLOW
        SUM5 = 0
        DO J = 1, JMAX
        DCF(J) = NCF(J)/(1 + ITRT)**J    !DISCOUNTED CASH FLOW
        SUM5 = SUM5 + DCF(J)
        END DO
        TDCF = SUM5                        !SUM OF DCF
        NPV = TDCF - PCC1

        !EVALUATING PAY BACK PERIOD
        SUM6 = 0
        DO J = 1, JMAX
        DCF(J) = NCF(J)/(1 + ITRT)**J
        SUM6 = SUM6 + DCF(J)
        IF ((PCC1-SUM6) <= 0)THEN
        GOTO 1000
        END IF
        !ELSE
        END DO  !SUM5 = SUM5

1000    PBP = J

        PRINT 200, ' E C O N O M I C   O U T P U T   S U M M A R Y '
        PRINT 200, ' COAL POWER GENERATION PLANT '
        WRITE(10,200)' E C O N O M I C   O U T P U T   S U M M A R Y '
         WRITE(10,200)' COAL POWER GENERATION PLANT '
         PRINT*,' YEAR',' REVENUE(£)',' EXPENDITURE(£)',' PROFIT(£)','
                    NCF(£)','      DCF(£)'
           WRITE(10,*)' YEAR',' REVENUE(£)',' EXPENDITURE(£)',' PROFIT(£)','
                           NCF(£)','       DCF(£)'
            DO J=1,25
              PRINT 215, J, REV(J), EXPEND(J), PROFIT(J), NCF(J), DCF(J)
              WRITE(10,210)J, REV(J), EXPEND(J), PROFIT(J), NCF(J), DCF(J)
            END DO
        PRINT 220, NPV,INVST1,PBP,ANF,ITRT,CF,EP
        WRITE(10,220) NPV,INVST1,PBP,ANF,ITRT,CF,EP
```

```
    !2.READING GT POWER AND FUEL FLOW FROM THE INPUT FILES
    OPEN(UNIT=5,FILE='OILNGDUALFIREDPLANT.inp')

    !CHECK- READING ALL VARIABLES

    DO K=1,2
    READ(5,'(A10)'),DUMMY

        DO J=1,JMAX

    READ(UNIT=5,FMT= *) (SZN(I,J,K),I=1,IMAX)
        END DO
    END DO


    ! REVENUE EVALUATION****************************************************

    DO J = 1, JMAX
       SUM = 0
       K = 1
        DO I = 1, IMAX
        IF (I == 2) THEN
        LF = LF2
        ELSE
        LF = LF1
        END IF
    RVN(I,J) = EP * SZN(I,J,K) * LF * 8760 * 1000         !SZN IS THE REQUIRED
POWER OUTPUT
        SUM = SUM + RVN(I,J)
        END DO
    REV(J) = SUM

    ! CALCULATION OF EXPENSES***************************************************
    ! FUEL COST
      SUM1 = 0
        K = 2
        DO I = 1, IMAX
        IF (I == 2) THEN
        LF = LF2
        ELSE
        LF = LF1
        END IF
        EXP1(I,J) = CF * SZN(I,J,K) * LF * 8760 * 3600         !SZN K = 2  IS THE
GT FUEL FLOW IN KG/SEC
        SUM1 = SUM1 + EXP1(I,J)
        END DO
       EXPEND1(J) = SUM1

    ! VARIABLE O & M COST
       SUM2 = 0
        K = 1
        DO I = 1, IMAX
        IF (I == 2) THEN
        LF = LF2
        ELSE
        LF = LF1
        END IF
        EXP2(I,J) = VC1 * SZN(I,J,K) * LF * 8760          !SZN K = 1 IS THE
REQUIRED POWER OUTPUT IN MW
```

```
        SUM2 = SUM2 + EXP2(I,J)
        END DO
       EXPEND2(J) = SUM2


   ! FIXED COST

        EXPEND3(J) = FC1 * P1 * 1000

    ! DEPRECIATION COST

    EXPEND4(J) = ADP * PCC1/100

    ! EMISSION COST
        SUM4 = 0
        K = 1
        DO I = 1, IMAX
        IF (I == 2) THEN
        LF = LF2
        ELSE
        LF = LF1
        END IF

        EXP4(I,J) =  EI1 * ETX * SZN(I,J,K) * LF * 8760
         SUM4 = SUM4 + EXP4(I,J)
        END DO
        EXPEND5(J) = SUM4

    ! TOTAL EXPENITURE
        EXPEND(J)= EXPEND1(J) + EXPEND2(J) + EXPEND3(J) + EXPEND4(J) + EXPEND5(J)
! TOTAL EXPENDITURE

      ! CASH FLOW
        OPINCOM(J) = REV(J) - EXPEND(J) ! OPERATING INCOME

        PROFIT(J) = OPINCOM(J) - INVST1 !PROFIT

        NCF(J) = PROFIT(J)*(1 - TXRT/100) - INVST1  ! NET CASH FLOW
   END DO

    ! DISCOUNTING THE CASH FLOW
        SUM5 = 0
        DO J = 1, JMAX
        DCF(J) = NCF(J)/(1 + ITRT)**J    !DISCOUNTED CASH FLOW
        SUM5 = SUM5 + DCF(J)
        END DO
        TDCF = SUM5                         !SUM OF DCF

        NPV = TDCF - PCC1

        !EVALUATING PAY BACK PERIOD
        SUM6 = 0
        DO J = 1, JMAX
        DCF(J) = NCF(J)/(1 + ITRT)**J
        SUM6 = SUM6 + DCF(J)
        IF ((PCC1-SUM6) <= 0)THEN
        GOTO 2000
        END IF
        !ELSE
        END DO  !SUM5 = SUM5
```

```
2000    PBP = J
        PRINT 200, ' E C O N O M I C   O U T P U T   S U M M A R Y '
        WRITE(10,200)' E C O N O M I C   O U T P U T   S U M M A R Y '
        PRINT 200, ' OIL AND NATURAL GAS DUAL FIRED PLANT '
        WRITE(10,200)' OIL AND NATURAL GAS DUAL FIRED PLANT '
         PRINT*,' YEAR',' REVENUE(£)',' EXPENDITURE(£)',' PROFIT(£)','
                    NCF(£)','      DCF(£)'
           WRITE(10,*)' YEAR',' REVENUE(£)',' EXPENDITURE(£)',' PROFIT(£)','
                           NCF(£)','       DCF(£)'
            DO J=1,25
              PRINT 215, J, REV(J), EXPEND(J), PROFIT(J), NCF(J), DCF(J)
              WRITE(10,210)J, REV(J), EXPEND(J), PROFIT(J), NCF(J), DCF(J)
            END DO
        PRINT 220, NPV,INVST1,PBP,ANF,ITRT,CF,EP
        WRITE(10,220) NPV,INVST1,PBP,ANF,ITRT,CF,EP


 !3. READING GT POWER AND FUEL FLOW FROM THE INPUT FILES
     OPEN(UNIT=5,FILE='CCGTPOWERPLANT.inp')

    !CHECK- READING ALL VARIABLES

    DO K=1,2
    READ(5,'(A10)'),DUMMY

        DO J=1,JMAX

    READ(UNIT=5,FMT= *) (SZN(I,J,K),I=1,IMAX)
        END DO
    END DO


    ! REVENUE EVALUATION****************************************************


    DO J = 1, JMAX
        SUM = 0
         K = 1
        DO I = 1, IMAX
        IF (I == 2) THEN
        LF = LF2
        ELSE
        LF = LF1
        END IF
    RVN(I,J) = EP * SZN(I,J,K) * LF * 8760 * 1000         !SZN IS THE REQUIRED
POWER OUTPUT
        SUM = SUM + RVN(I,J)
        END DO
    REV(J) = SUM

    ! CALCULATION OF EXPENSES****************************************************
    ! FUEL COST
      SUM1 = 0
        K = 2
        DO I = 1, IMAX
        IF (I == 2) THEN
        LF = LF2
        ELSE
        LF = LF1
```

```
        END IF
        EXP1(I,J) = CF * SZN(I,J,K) * LF * 8760 * 3600         !SZN K = 2  IS THE
GT FUEL FLOW IN KG/SEC
        SUM1 = SUM1 + EXP1(I,J)
        END DO
       EXPEND1(J) = SUM1

    ! VARIABLE O & M COST
        SUM2 = 0
        K = 1
        DO I = 1, IMAX
        IF (I == 2) THEN
        LF = LF2
        ELSE
        LF = LF1
        END IF

        EXP2(I,J) = VC1 * SZN(I,J,K) * LF * 8760          !SZN K = 1 IS THE
REQUIRED POWER OUTPUT IN MW
        SUM2 = SUM2 + EXP2(I,J)
        END DO
       EXPEND2(J) = SUM2


   ! FIXED COST

        EXPEND3(J) = FC1 * P1 * 1000

    ! DEPRECIATION COST

    EXPEND4(J) = ADP * PCC1/100

    ! EMISSION COST
        SUM4 = 0
        K = 1
        DO I = 1, IMAX
        IF (I == 2) THEN
        LF = LF2
        ELSE
        LF = LF1
        END IF

        EXP4(I,J) =  EI1 * ETX * SZN(I,J,K) * LF * 8760
         SUM4 = SUM4 + EXP4(I,J)
        END DO
        EXPEND5(J) = SUM4

    ! TOTAL EXPENITURE
        EXPEND(J)= EXPEND1(J) + EXPEND2(J) + EXPEND3(J) + EXPEND4(J) + EXPEND5(J)
! TOTAL EXPENDITURE

      ! CASH FLOW
        OPINCOM(J) = REV(J) - EXPEND(J) ! OPERATING INCOME

        PROFIT(J) = OPINCOM(J) - INVST1 !PROFIT

        NCF(J) = PROFIT(J)*(1 - TXRT/100) - INVST1  ! NET CASH FLOW
   END DO

    ! DISCOUNTING THE CASH FLOW
```

```
        SUM5 = 0
        DO J = 1, JMAX
        DCF(J) = NCF(J)/(1 + ITRT)**J    !DISCOUNTED CASH FLOW
        SUM5 = SUM5 + DCF(J)
        END DO
        TDCF = SUM5                        !SUM OF DCF

        NPV = TDCF - PCC1

         !EVALUATING PAY BACK PERIOD
        SUM6 = 0
        DO J = 1, JMAX
        DCF(J) = NCF(J)/(1 + ITRT)**J
        SUM6 = SUM6 + DCF(J)
        IF ((PCC1-SUM6) <= 0)THEN
        GOTO 3000
        END IF
        !ELSE
        END DO  !SUM5 = SUM5

3000     PBP = J
        PRINT 200, ' E C O N O M I C   O U T P U T   S U M M A R Y '
        WRITE(10,200)' E C O N O M I C   O U T P U T   S U M M A R Y '
        PRINT 200, ' CCGT POWER GENERATION PLANT '
        WRITE(10,200)' CCGT POWER GENERATION PLANT '
         PRINT*,' YEAR',' REVENUE(£)',' EXPENDITURE(£)',' PROFIT(£)','
                    NCF(£)','      DCF(£)'
           WRITE(10,*)' YEAR',' REVENUE(£)',' EXPENDITURE(£)',' PROFIT(£)','
                           NCF(£)','       DCF(£)'
            DO J=1,25
              PRINT 215, J, REV(J), EXPEND(J), PROFIT(J), NCF(J), DCF(J)
              WRITE(10,210)J, REV(J), EXPEND(J), PROFIT(J), NCF(J), DCF(J)
            END DO
        PRINT 220, NPV,INVST1,PBP,ANF,ITRT,CF,EP
        WRITE(10,220) NPV,INVST1,PBP,ANF,ITRT,CF,EP

 !4. READING GT POWER AND FUEL FLOW FROM THE INPUT FILES
     OPEN(UNIT=5,FILE='GASTURBINEPOWERPLANT.inp')

    !CHECK- READING ALL VARIABLES

    DO K=1,2
    READ(5,'(A10)'),DUMMY

        DO J=1,JMAX

    READ(UNIT=5,FMT= *) (SZN(I,J,K),I=1,IMAX)
        END DO
    END DO


    ! REVENUE EVALUATION***************************************************

    DO J = 1, JMAX
       SUM = 0
       K = 1
        DO I = 1, IMAX
        IF (I == 2) THEN
        LF = LF2
        ELSE
```

```
        LF = LF1
        END IF
    RVN(I,J) = EP * SZN(I,J,K) * LF * 8760 * 1000         !SZN IS THE REQUIRED
POWER OUTPUT
        SUM = SUM + RVN(I,J)
        END DO
    REV(J) = SUM

    ! CALCULATION OF EXPENSES***************************************************
    ! FUEL COST
      SUM1 = 0
        K = 2
        DO I = 1, IMAX
        IF (I == 2) THEN
        LF = LF2
        ELSE
        LF = LF1
        END IF
        EXP1(I,J) = CF * SZN(I,J,K) * LF * 8760 * 3600         !SZN K = 2  IS THE
GT FUEL FLOW IN KG/SEC
        SUM1 = SUM1 + EXP1(I,J)
        END DO
       EXPEND1(J) = SUM1

    ! VARIABLE O & M COST
        SUM2 = 0
        K = 1
        DO I = 1, IMAX
        IF (I == 2) THEN
        LF = LF2
        ELSE
        LF = LF1
        END IF
        EXP2(I,J) = VC2 * SZN(I,J,K) * LF * 8760          !SZN K = 1 IS THE
REQUIRED POWER OUTPUT IN MW
        SUM2 = SUM2 + EXP2(I,J)
        END DO
       EXPEND2(J) = SUM2


   ! FIXED COST

        EXPEND3(J) = FC2 * P2 * 1000

    ! DEPRECIATION COST

    EXPEND4(J) = ADP * PCC2/100

    ! EMISSION COST
        SUM4 = 0
        K = 1
        DO I = 1, IMAX
        IF (I == 2) THEN
        LF = LF2
        ELSE
        LF = LF1
        END IF

        EXP4(I,J) =  EI2 * ETX * SZN(I,J,K) * LF * 8760
         SUM4 = SUM4 + EXP4(I,J)
```

```
        END DO
        EXPEND5(J) = SUM4

    ! TOTAL EXPENITURE
        EXPEND(J)= EXPEND1(J) + EXPEND2(J) + EXPEND3(J) + EXPEND4(J) + EXPEND5(J)
! TOTAL EXPENDITURE

      ! CASH FLOW
        OPINCOM(J) = REV(J) - EXPEND(J) ! OPERATING INCOME

        PROFIT(J) = OPINCOM(J) - INVST2 !PROFIT

        NCF(J) = PROFIT(J)*(1 - TXRT/100) - INVST2  ! NET CASH FLOW
   END DO

    ! DISCOUNTING THE CASH FLOW
        SUM5 = 0
        DO J = 1, JMAX
        DCF(J) = NCF(J)/(1 + ITRT)**J    !DISCOUNTED CASH FLOW
        SUM5 = SUM5 + DCF(J)
        END DO
        TDCF = SUM5                        !SUM OF DCF

        NPV = TDCF - PCC2

         !EVALUATING PAY BACK PERIOD
        SUM6 = 0
        DO J = 1, JMAX
        DCF(J) = NCF(J)/(1 + ITRT)**J
        SUM6 = SUM6 + DCF(J)
        IF ((PCC2-SUM6) <= 0)THEN
        GOTO 4000
        END IF
        !ELSE
        END DO  !SUM5 = SUM5

4000     PBP = J
        PRINT 200, ' E C O N O M I C   O U T P U T   S U M M A R Y '
        WRITE(10,200)' E C O N O M I C   O U T P U T   S U M M A R Y '
        PRINT 200, ' GAS TURBINE POWER GENERATION PLANT '
        WRITE(10,200)' GAS TURBINE POWER GENERATION PLANT '
         PRINT*,' YEAR',' REVENUE(£)',' EXPENDITURE(£)',' PROFIT(£)','
              NCF(£)','      DCF(£)'
           WRITE(10,*)' YEAR',' REVENUE(£)',' EXPENDITURE(£)',' PROFIT(£)','
                            NCF(£)','       DCF(£)'
            DO J=1,25
              PRINT 215, J, REV(J), EXPEND(J), PROFIT(J), NCF(J), DCF(J)
              WRITE(10,210)J, REV(J), EXPEND(J), PROFIT(J), NCF(J), DCF(J)
            END DO
        PRINT 220, NPV,INVST2,PBP,ANF,ITRT,CF,EP
        WRITE(10,220) NPV,INVST2,PBP,ANF,ITRT,CF,EP


    END PROGRAM ECONOMICMODULE2014
```

D.2 Typical Input File of the Economic Module

```
   COAL POWER PLANT

GT POWER FOR THE FOUR SEASONS

23.22756    15.09791    17.42067    20.90480

23.10885    15.02075    17.33164    20.79797

21.99545    14.29704    16.49659    19.79591

20.18578    13.12076    15.13933    18.16720

20.02611    13.01697    15.01959    18.02350

19.90550    12.93858    14.92913    17.91495

19.90550    12.93858    14.92913    17.91495

19.88405    12.92463    14.91304    17.89564

19.83841    12.89496    14.87880    17.85456

19.81859    12.88208    14.86394    17.83673

19.81859    12.88208    14.86394    17.83673

19.81859    12.88208    14.86394    17.83673

19.81859    12.88208    14.86394    17.83673

19.81859    12.88208    14.86394    17.83673

19.81859    12.88208    14.86394    17.83673

19.81859    12.88208    14.86394    17.83673

19.81859    12.88208    14.86394    17.83673

19.81859    12.88208    14.86394    17.83673

19.81859    12.88208    14.86394    17.83673

19.81859    12.88208    14.86394    17.83673

19.81859    12.88208    14.86394    17.83673

19.81859    12.88208    14.86394    17.83673

19.81859    12.88208    14.86394    17.83673

19.81859    12.88208    14.86394    17.83673

19.81859    12.88208    14.86394    17.83673
```

GT FUEL FLOW FOR THE FOUR SEASONS

1.467600	1.052200	1.168600	1.348100
1.465400	1.047200	1.163400	1.342100
1.403500	1.013000	1.121900	1.288900
1.314700	0.956700	1.050400	1.210100
1.305100	0.952100	1.047200	1.199200
1.299200	0.947600	1.047200	1.193800
1.299200	0.947600	1.047200	1.193800
1.297200	0.947600	1.042300	1.193800
1.293400	0.946200	1.041300	1.193800
1.291400	0.945100	1.041300	1.193800
1.291400	0.945100	1.041300	1.193800
1.291400	0.945100	1.041300	1.193800
1.291400	0.945100	1.041300	1.193800
1.291400	0.945100	1.041300	1.193800
1.291400	0.945100	1.041300	1.193800
1.291400	0.945100	1.041300	1.193800
1.291400	0.945100	1.041300	1.193800
1.291400	0.945100	1.041300	1.193800
1.291400	0.945100	1.041300	1.193800
1.291400	0.945100	1.041300	1.193800
1.291400	0.945100	1.041300	1.193800
1.291400	0.945100	1.041300	1.193800
1.291400	0.945100	1.041300	1.193800
1.291400	0.945100	1.041300	1.193800
1.291400	0.945100	1.041300	1.193800

Appendix E Typical Output of the Economic Code

E C O N O M I C   O U T P U T   S U M M A R Y

COAL POWER GENERATION PLANT

YEAR	REVENUE(£)	EXPENDITURE(£)	PROFIT(£)	NCF(£)	DCF(£)
1	18556772.00	9653382.00	7415846.50	4445134.00	3865334.00
2	18461940.00	9612910.00	7361486.50	4401646.00	3328276.75
3	17572428.00	9211251.00	6873633.50	4011363.50	2637536.75
4	16126660.00	8572606.00	6066510.50	3365665.00	1924330.00
5	15999102.00	8514003.00	5997555.50	3310501.00	1645904.12
6	15902744.00	8475400.00	5939800.50	3264297.00	1411245.88
7	15902744.00	8475400.00	5939800.50	3264297.00	1227170.38
8	15885605.00	8464362.00	5933699.50	3259416.00	1065509.00
9	15849138.00	8449430.00	5912164.50	3242188.00	921632.31
10	15833308.00	8442747.00	5903017.50	3234870.50	799610.62
11	15833308.00	8442747.00	5903017.50	3234870.50	695313.62
12	15833308.00	8442747.00	5903017.50	3234870.50	604620.56
13	15833308.00	8442747.00	5903017.50	3234870.50	525757.06
14	15833308.00	8442747.00	5903017.50	3234870.50	457180.03
15	15833308.00	8442747.00	5903017.50	3234870.50	397547.88
16	15833308.00	8442747.00	5903017.50	3234870.50	345693.81
17	15833308.00	8442747.00	5903017.50	3234870.50	300603.31
18	15833308.00	8442747.00	5903017.50	3234870.50	261394.19
19	15833308.00	8442747.00	5903017.50	3234870.50	227299.31
20	15833308.00	8442747.00	5903017.50	3234870.50	197651.56
21	15833308.00	8442747.00	5903017.50	3234870.50	171870.94
22	15833308.00	8442747.00	5903017.50	3234870.50	149453.00
23	15833308.00	8442747.00	5903017.50	3234870.50	129959.12

24	15833308.00	8442747.00	5903017.50	3234870.50	113007.94
25	15833308.00	8442747.00	5903017.50	3234870.50	98267.78

NPV(£)= 13886469.00 YEARLY CAPITAL COST SPREAD(£) = 1487543.62 PAY BACK PERIOD = 3

ANNUITY FACTOR = 0.1547 DISCOUNT FACTOR = 0.1500 FUEL PRICE(£/kg) = 0.12 ELECTRICITY PRICE(£) = 0.12

E C O N O M I C   O U T P U T   S U M M A R Y

OIL AND NATURAL GAS DUAL FIRED PLANT

YEAR	REVENUE(£)	EXPENDITURE(£)	PROFIT(£)	NCF(£)	DCF(£)
1	8695422.00	5277094.50	1930783.88	57083.50	49637.83
2	8689431.00	5273361.00	1928526.38	55277.50	41797.73
3	8663972.00	5260700.00	1915728.38	45039.12	29613.96
4	8786194.00	5314965.50	1983684.88	99404.25	56834.71
5	8769392.00	5310544.00	1971304.38	89499.88	44497.26
6	8782578.00	5317798.00	1977236.38	94245.50	40744.93
7	8944026.00	5395177.00	2061305.38	161500.75	60714.12
8	9121522.00	5475777.50	2158201.00	239017.25	78135.17
9	9318165.00	5567754.00	2262867.50	322750.38	91745.81
10	9556299.00	5662533.50	2406222.00	437434.00	108127.02
11	9842310.00	5789817.00	2564949.50	564416.00	121317.41
12	10161383.00	5934676.00	2739163.50	703787.12	131542.88
13	10355977.00	6007931.00	2860502.50	800858.38	130161.91
14	10453566.00	6065572.00	2900450.50	832816.88	117700.93
15	10597240.00	6128320.50	2981376.00	897557.12	110304.85
16	10874790.00	6249234.00	3138012.50	1022866.38	109308.42
17	11083402.00	6340823.00	3255035.50	1116484.88	103750.38

18	11259254.00	6418492.00	3353218.50	1195031.12	96564.66
19	11432952.00	6498905.50	3446503.00	1269658.88	89213.02
20	11481794.00	6518287.00	3475963.50	1293227.12	79016.57
21	11858247.00	6683261.50	3687442.00	1462410.12	77698.88
22	12160592.00	6820369.00	3852679.50	1594600.12	73671.50
23	12435184.00	6935092.00	4012548.50	1722495.12	69200.29
24	12711784.00	7062264.50	4161976.00	1842037.12	64350.28
25	12967383.00	7178537.00	4301302.50	1953498.38	59342.70

NPV(£)= -7580710.00 YEARLY CAPITAL COST SPREAD(£) = 1487543.62 PAY BACK PERIOD = 26

ANNUITY FACTOR = 0.1547 DISCOUNT FACTOR = 0.1500 FUEL PRICE(£/kg) = 0.12 ELECTRICITY PRICE(£) = 0.12

E C O N O M I C O U T P U T S U M M A R Y

CCGT POWER GENERATION PLANT

YEAR	REVENUE(£)	EXPENDITURE(£)	PROFIT(£)	NCF(£)	DCF(£)
1	11797410.00	6656219.50	3653647.00	1435374.12	1248151.50
2	12071232.00	6783266.00	3800422.50	1552794.38	1174135.62
3	12185738.00	6829162.00	3869032.50	1607682.38	1057077.38
4	12612710.00	7018084.00	4107082.50	1798122.38	1028082.38
5	12628616.00	7022270.00	4118802.50	1807498.38	898646.19
6	12652090.00	7033386.00	4131160.50	1817384.88	785705.69
7	12688754.00	7056204.00	4145006.50	1828461.62	687386.56
8	12781377.00	7093824.50	4200009.00	1872463.62	612111.75
9	12855402.00	7129124.00	4238734.50	1903444.12	541077.69
10	13050316.00	7209729.00	4353043.50	1994891.12	493106.66
11	13431044.00	7381996.00	4561504.50	2161660.00	464634.25

12	13868881.00	7580152.50	4801185.00	2353404.50	439868.25
13	13968880.00	7617803.50	4863533.00	2403283.00	390600.78
14	14423426.00	7821001.50	5114881.00	2604361.00	368070.97
15	14960318.00	8052814.00	5419960.50	2848425.00	350055.81
16	15344301.00	8220337.00	5636420.50	3021593.00	322901.97
17	15780118.00	8418664.00	5873910.50	3211585.00	298439.50
18	16181762.00	8596057.00	6098161.50	3390986.00	274009.12
19	16630204.00	8800865.00	6341795.50	3585893.00	251964.03
20	17213440.00	9057382.00	6668514.50	3847268.00	235069.23
21	17469390.00	9164740.00	6817106.50	3966142.00	210723.89
22	17772460.00	9303400.00	6981516.50	4097670.00	189314.86
23	18137800.00	9464276.00	7185980.50	4261241.00	171193.00
24	18453204.00	9602854.00	7362806.50	4402702.00	153805.31
25	18744358.00	9741159.00	7515655.50	4524981.00	137458.31

NPV(£)= 3167886.00 YEARLY CAPITAL COST SPREAD(£) = 1487543.62 PAY BACK PERIOD = 13

ANNUITY FACTOR = 0.1547 DISCOUNT FACTOR = 0.1500 FUEL PRICE(£/kg) = 0.12 ELECTRICITY PRICE(£) = 0.12

E C O N O M I C   O U T P U T   S U M M A R Y

GAS TURBINE POWER GENERATION PLANT

YEAR	REVENUE(£)	EXPENDITURE(£)	PROFIT(£)	NCF(£)	DCF(£)
1	5928081.50	3536133.50	1842584.50	924704.12	804090.56
2	5897863.00	3520713.75	1827785.75	912865.12	690257.19
3	5847811.00	3493910.50	1804537.00	894266.12	587994.56
4	5803305.50	3471442.50	1782499.50	876636.12	501219.56
5	5700221.00	3421987.00	1728870.50	833732.88	414512.62

6	5557013.50	3345567.50	1662082.50	780302.50	337346.34
7	5367618.00	3249375.25	1568879.25	705739.88	265313.81
8	5292015.00	3212265.50	1530386.00	674945.38	220640.88
9	5220184.00	3177199.50	1493621.00	645533.38	183500.91
10	5137563.50	3136113.75	1452086.25	612305.50	151352.58
11	5106397.00	3118724.50	1438309.00	601283.75	129241.89
12	4922094.50	3020970.50	1351760.50	532044.88	99443.02
13	4811953.50	2971444.00	1291146.00	483553.31	78590.95
14	4746804.50	2936688.25	1260752.75	459238.69	64903.61
15	4700766.50	2915109.50	1236293.50	439671.31	54033.20
16	4572182.00	2850482.25	1172336.25	388505.50	41517.57
17	4455721.00	2793097.75	1113259.75	341244.31	31710.44
18	4420815.50	2775094.25	1096357.75	327722.69	26481.68
19	4344582.50	2736684.50	1058534.50	297464.12	20901.42
20	4301636.00	2715947.25	1036325.25	279696.69	17089.55
21	4297896.00	2714746.00	1033786.50	277665.69	14752.57
22	4287935.50	2707422.25	1031149.75	275556.31	12730.87
23	4279458.50	2702712.75	1027382.25	272542.31	10949.24
24	4279458.50	2702712.75	1027382.25	272542.31	9521.08
25	4279458.50	2702712.75	1027382.25	272542.31	8279.20

NPV(£)= 1225207.25 YEARLY CAPITAL COST SPREAD(£) = 549363.50 PAY BACK PERIOD = 7

ANNUITY FACTOR = 0.1547 DISCOUNT FACTOR = 0.1500 FUEL PRICE(£/kg) = 0.12 ELECTRICITY PRICE(£) = 0.12

Appendix F Pipeline Hydraulic Analysis Code

F.1 Main Pipeline Hydraulic Analysis Program

```
PROGRAM PIPELINE_HYDRAULIC_ANALYSIS
IMPLICIT NONE

! THIS CODE IS USED FOR THE PIPELINE/COMPRESSION MODULE FOR TERA IN APPLICATION
!TO CO2 PIPELINE DEVELOPED
!BY A. EL-SULEIMAN AS PART OF THE PhD THESIS IN CRANFIELD UNIVERSITY, UK- 2014.

    !ASSUMPTIONS


! J = 1,2,3,................,25 REPRESENTS YEARS 2013,2014,.................,2037
! ----------------------------------------------------------------
! THE PROFILE IS SHOWN IN FIG 3-2 OF THE MAIN THESIS

! ----------------------------------------------------------------

    INTEGER,PARAMETER :: S1 = 12, S2 = 75, S3 = 3
    INTEGER :: I,J,K,IMAX,JMAX,KMAX,ITER,ITMAX
    INTEGER, DIMENSION(0:S2) :: K1
    REAL, DIMENSION(S1,S2) :: ELV, EXP2,EXP3,EXP4
    REAL, DIMENSION(0:S2) :: LNT, S, LEQV, SLP, POUT,Z1
    REAL :: P,P1, P2,P21,PGUESS,PB,TB, TF,TFAV, EPS, FF, TOL, FFGUESS,
WF,RHO,VIS,PAVE,PMAX,EXP,MAIR,Q1,Q2
    REAL :: REYNO, MCO2, RUNIV, Pcrit, Tcrit, PI, QFL, QFL2, A1, A2, A3, A22,
VEL,TCK, DIN, DOUT, PL, RCO2
    REAL :: LEQVT, SUM, X, Z, PAV, Q3, Q4, A4, SGCO2, PAVE1, X1, A,QFLCO2, T1,
T2, T3,VSP
    CHARACTER (LEN=10) :: DUMMY
! ----------------------------------------------------------------
!    FORMATS
200    FORMAT(' ',A55,/)
215 FORMAT(I7,F18.2,F18.2,F18.2,I10,F13.2,F13.2)
220 FORMAT(/'PIPE INLET PRESSURE(KPa) = ',F8.2)
230 FORMAT(/'PIPE EQUIV LENGTH(KM) = ',F7.2,1X,'REYNOLDS NO = ',F8.2,1X,'FRICTION
FACTOR =',F8.4,/,'FLOW(SCMD) = ',F9.4,1X,'PIPE SIZE(m) =',F8.4)

! ----------------------------------------------------------------

    OPEN(UNIT=6,STATUS='REPLACE',FILE='PRESSURE DROP RESULTS.DAT')

! ----------------------------------------------------------------
!      INPUT DATA AND ASSUMPTIONS
! ----------------------------------------------------------------
        IMAX = 4
        JMAX = 31 ! NUMBER OF SEGMENTS
        ITMAX= 1000
        KMAX = 1000


        OPEN(UNIT=5,FILE='FLAT PROFILE.inp')

        ! READING ALL PROFILE VAR
     READ(5,'(A10)'),DUMMY
        READ(5,*) (LNT(J),J=1,JMAX)
```

```
READ(5,'(A10)'),DUMMY

    DO J=1,JMAX

READ(UNIT=5,FMT= *) (ELV(I,J),I=1,2)
    END DO

   ! DATA ASUMPTIONS

  MCO2 = 44.01 ! MOLECULAR WEIGHT OF CO2
  RUNIV = 8.314 ! UNIVERSAL GAS CONSTANT
  TB = 25+273.15 !BASE TEMPERATURE IN K
  PB = 101.325  ! BASE PRESSURE IN KPA
  Pcrit = 7.38E+06 ! CRITICAL PRESSURE OF CO2 in Pa
  Tcrit = 31.1   !CRITICAL TEMPERATURE OF CO2 IN Degrees Centrigrade

   RCO2 = RUNIV*1E+3/MCO2      ! SPECIFIC GAS CONSTANT
   MAIR = 28.9625  ! MASS OF AIR
   SGCO2  = MCO2/MAIR  !0.8763 !MCO2/MAIR !SPECIFIC GRAVITY OF CO2

   !INPUT
    PAVE = 15000  !INLET PRESSURE IN KPa
   TF = 25        ! AVERAGE FLOW TEMP IN DEGREE CELSIUS
   TFAV = TF+273.15 !
   !RHO = P/(Z*RCO2*T)

   ! CALCULATING INITIAL COMPRESSIBILITY OF THE CO2 AT PIPELINE INLET

   CALL Z_FACTOR(PAVE,TFAV,Z)
   P = PAVE*1E3
   VSP = 8314*Z*TFAV/(MCO2*P)
   QFLCO2 = 3.33         !CO2 THROUGHPUT IN MTPY 3.70,2.41,2.78,3.33
   QFLCO2 = QFLCO2*1E6/365  !CO2 THROUGHPUT IN TPD
   QFLCO2 = QFLCO2*1E3/24  !CO2 THROUGHPUT IN KG/HR
   WF = QFLCO2/3600  !CO2 THROUGHPUT IN KG/SEC MASS FLOW RATE

   QFLCO2 = QFLCO2*VSP  !CO2 THROUGHPUT IN M3/HR
   QFLCO2 = QFLCO2*1E3/(1.179869)  !CO2 THROUGHPUT IN FT3/D
   QFLCO2 = QFLCO2/1440  !CO2 THROUGHPUT IN FT3/MIN

   T1 = 9*TF/5+32   ! TEMPERATURE IN DEGREES FARENHEIGHT
   T1 = T1 + 460
   P = P/(PB*1E3)
   P = P/14.696
   QFLCO2 = QFLCO2*P*520/(14.969*T1) !CO2 THROUGHPUT IN SCFM
   QFLCO2 = QFLCO2*60*24 !CO2 THROUGHPUT IN SCFD
   QFL2 = QFLCO2*1.179869*1E-3 !CO2 THROUGHPUT IN SCMH
   QFL = QFL2*24  !CO2 THROUGHPUT IN SCMD

    PMAX = 15.3 ! IN MPA
    EXP = 2.718281828
    PI = 4.0 * ATAN(1.0)
    RHO = 876.3 ! CO2 DENSITY IN KG/M3 OBTAINED FROM CO2 PIPELINE OPERATORS
    VIS = 9.2605E-5 ! DYNAMIC VISCOSITY OBTAINED FROM EMPIRICAL RELATIONSHIP
    DOUT= 0.3556 ! 0.3556m, 0.3239m or 14" 12and3/4";
    TCK = PMAX*DOUT/(2*482*1*0.72)
    DIN = DOUT-2*TCK
    VEL = 4*WF/(PI*RHO*DIN**2)

    !QFL2 = 327.1964082
```

```
       REYNO = 0.5134*PB*SGCO2*QFL/(TB*VIS*DIN*1000)

       ! FINDING THE FRICTION FACTOR, FF USING COLEBROOK'S-WHITE EQUATION
1000    TOL = 1E-6
        EPS = 0.045
        FFGUESS= 0.0001
       DO ITER = 1,ITMAX
       A22 = EPS/(3.7*DIN*1000)+2.51/(REYNO*SQRT(FFGUESS))
              A2 = -2*LOG10(A22)
        A3 = 1/SQRT(FFGUESS)
        A1 = ABS(A2-A3)
        FFGUESS = (1/A2)**2
       IF (A1<=TOL .OR. ITER == ITMAX) THEN
       FF = FFGUESS
       GOTO 2000
       END IF
       END DO


2000     P1 = 15000  !INLET PIPE PRESSURE IN KPA
          PRINT 220, P1
          WRITE(6, 220) P1
2500     DO J = 1, JMAX
          I = 1
          P2 = P1-500 !GUESSED VALUE FOR POUT
          K = 1

          IF (P2<=10000) THEN
          PRINT *, ' POINT OF RECOMPRESSION'
          END IF
            DO
        X = P1+P2
        PAV = P1*P2/(P1+P2)
        PAVE1 = X-PAV
        PAVE = 2*PAVE1/3   !2/3*(P1+P2-(P1*P2/(P1+P2)))! AVERAGE PRESSURE
        CALL Z_FACTOR(PAVE,TFAV,Z)
        Z1(J) = Z

       ! ELEVATION CORRECTION FACTOR

        A4 = ELV(I+1,J)-ELV(I,J)

       S(J) = 0.0684*SGCO2*A4/((TF+273.15)*Z)! ELEVATION ADJUSTMENT TERM

       ! DEFINING J PARAMETER
       IF (A4 == 0) THEN
       A = 1
       SLP(J) = 1
       ELSE
       SLP(J) = (EXP**S(J)-1)/S(J)
       A =  EXP**S(J-1)
       END IF

       ! EQUIVALENT LENGTH

       LEQV(J) = LNT(J)*SLP(J)*A    !LEQV(J) = LNT(J)*SLP(J)*EXP**S(J-1)


       !CALCULATING P2 FROM THE GENERAL FLOW EQUATION
```

```
5000    Q1 = (QFL*SQRT(FF)*PB*1E4/((DIN*1E3)**2.5*TB*11.4946))**2

        X1 = SGCO2*TFAV*LEQV(J)*1000*Z

        Q4 = Q1*X1
        Q2 = P1**2-Q4
        Q3 = Q2/EXP**S(J)
        P21 = SQRT(Q3)

        A1 = ABS(P2-P21)
        P2 = P21

        IF (A1<=TOL .OR. K == KMAX) EXIT
        K = K+1
            END DO
        POUT(J) = P2
        K1(J) = K
          P1 = P2
     END DO
        SUM = 0
        DO J = 1, JMAX
        SUM = SUM + LEQV(J)
        END DO
        LEQVT = SUM

       PRINT 200, ' HYDRAULIC RESULT   S U M M A R Y '
       WRITE(6,200)' HYDRAULIC RESULT   S U M M A R Y '

       PRINT*,' SEGMENT NO  ',' OUTLET PRESSURE(KPa) ',' EQUIV LENGTH(km) ','
COMP FACTOR',' NO.OF ITER '
       WRITE(6,*)'/ SEGMENT NO  ',' OUTLET PRESSURE(KPa) ',' EQUIV LENGTH(km) ','
COMP FACTOR',' NO.OF ITER /'
       DO J=1,JMAX
       PRINT 215, J,POUT(J),LEQV(J),Z1(J),K1(J)
       WRITE(6, 215) J,POUT(J),LEQV(J),Z1(J),K1(J)
       END DO
       PRINT 230, LEQVT,REYNO,FF,QFL,DOUT
       WRITE(6, 230) LEQVT,REYNO,FF,QFL,DOUT

    END PROGRAM PIPELINE_HYDRAULIC_ANALYSIS
```

F.2 Subroutine for Compressibility Factor Using PR-EOS

```
SUBROUTINE Z_FACTOR(PAVE,TFAV,Z)

IMPLICIT NONE

!THIS CODE IS USED FOR THE PIPELINE/COMPRESSION MODULE FOR TERA IN APPLICATION TO
!CO2 PIPELINE DEVELOPED
!BY A. EL-SULEIMAN AS PART OF THE PhD THESIS IN CRANFIELD UNIVERSITY, UK- 2014.

REAL :: MCO2, RUNIV, Pcrit, TcritC, TcritK,TK, AFAC, EPS, BT, T, P, B, A, H1, H2,
        H3, H4,TFAV,PAVE
REAL ::  PR,  TR, VTH, AOFT, BOFT, A1, A2, A3, B1, B2, DET, C1, C2, RCO2, Z, BT1,
        B21, G1, G2, Z1
!------------------------------------------------------------------------
! SIMULATION CO2 COMPRESSIBILITY FACTOR USING THE PENG-ROBINSON EQUATION OF STATE
```

```
!-----------------------------------------------------------------------

      MCO2 = 44.01 ! MOLECULAR WEIGHT OF CO2
      RUNIV = 8.314 ! UNIVERSAL GAS CONSTANT
      TK = 273.15 !
      Pcrit = 7.38E+06 ! CRITICAL PRESSURE OF CO2 in Pa
      TcritC = 31.1 !CRITICAL TEMPERATURE OF CO2 IN Degrees Centrigrade
      TcritK = TcritC + TK
      AFAC = 0.22394  !ACCENTRIC FACTOR

      RCO2 = RUNIV*1E+3/MCO2      ! SPECIFIC GAS CONSTANT


       P = PAVE*1E3
       T = TFAV
       PR = P / Pcrit
       TR = T / TcritK
       EPS = 0.37464+1.54226*AFAC-0.26992*AFAC**2
       BT1 = 1-SQRT(TR)
       BT = (1+EPS*BT1)**2

       AOFT = 0.45724*BT*RUNIV**2*TcritK**2/Pcrit
       BOFT = 0.0778*RUNIV*TcritK/Pcrit

       A = AOFT*P/(RUNIV**2*T**2)
       B = BOFT*P/(RUNIV*T)

       A1 = -1*(1-B)
       A2 = A-3*B**2-2*B
       A3= -(A*B-B**2-B**3)

       B1 = (A1**2-3*A2)/9
       C1 = B1**3
       B21 = (2*A1**3)-(9*A1*A2)+(27*A3)
       B2= B21/54
       C2 = B2**2

       H1 = C2-C1
       Z1 = H1**0.5

       H2 = ABS(B2)+Z1
       G1 = FLOAT(1)/FLOAT(3)
       H2 = H2**G1

       H3 = B1/H2

       H4 = H2+H3-A1/3

       Z = H4


       RETURN
   END
```

F.3 Pipeline Profile Input File

```
PROFILE SEGMENT LENGTH

17    22.4  19.6  19.7  20    21.3  20    19    13    12    17    19
      19    22    28    31    14    10    15    18    16    15    13
      8     9     13    13    7     8     17    15

SEGMENT ELEVATION

117   111

111   101

101   111

111   122

122   114

114   104

104   104

104   104

104   93.9

93.9  82

82    67.6

67.6  69.7

69.7  72.8

72.8  83

83    96.1

96.1  106

106   94.9

94.9  86.4

86.4  74.5

74.5  94.1

94.1  112

112   131

131   150
```

150	164
164	177
177	159
159	140
140	128
128	114
114	106
106	99.9

Appendix G Typical Pipeline Hydraulic Analysis Output

HYDRAULIC SIMULATION OUTPUT FILE

PIPE INLET PRESSURE(KPa) = 15000.00

HYDRAULIC RESULT S U M M A R Y

/ SEGMENT NO	OUTLET PRESSURE(KPa)	EQUIV LENGTH(km)	COMP FACTOR	NO.OF ITER /
1	15049.65	16.94	0.31	4
2	15133.03	22.12	0.31	4
3	15046.04	19.49	0.31	4
4	14950.61	20.05	0.31	4
5	15016.98	20.16	0.31	4
6	15100.41	20.99	0.31	4
7	15098.67	20.00	0.31	3
8	15097.02	19.00	0.31	3
9	15182.11	12.93	0.31	4
10	15282.81	11.79	0.31	4
11	15404.66	16.64	0.31	4
12	15385.05	18.72	0.31	4
13	15356.85	19.08	0.31	3
14	15267.64	22.20	0.31	4
15	15153.24	28.53	0.31	4
16	15066.04	31.64	0.31	4
17	15159.52	14.07	0.31	4
18	15231.29	9.83	0.31	4
19	15331.80	14.76	0.31	4
20	15162.68	17.96	0.31	4
21	15008.58	16.52	0.31	4
22	14845.57	15.47	0.31	4
23	14683.15	13.43	0.30	4
24	14563.86	8.24	0.30	4
25	14453.16	9.22	0.30	4
26	14604.22	13.06	0.30	4
27	14764.18	12.59	0.30	4
28	14865.51	6.80	0.30	4

29	14983.95	7.83	0.31
4			
30	15050.67	16.66	0.31
4			
31	15101.40	14.81	0.31
4			

PIPE EQUIV LENGTH(KM) = 511.52 REYNOLDS NO = 57706.77 FRICTION FACTOR = 0.0207
FLOW(SCMD) = 6851.5410 PIPE SIZE(m) = 0.3556

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