CRANFIELD UNIVERSITY

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Techno-Economic Study of Gas Turbine in Pipeline Applications

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Abstract

Natural gas being the cleanest fossil fuel today is receiving tremendous rise in demand for both industrial and domestic energy requirements. The availability of natural gas requires it to be transported from the production area through pipeline in most cases to the consumers; this requires compressor station mostly driven by gas turbine. The development of gas pipeline system requires important data such as appropriate pipe sizes, gas rate, required delivery pressure, appropriate compressor and gas turbine sizes. The investment for the pipeline and compressor station is capital intensive and therefore the techno-economic and environmental risk assessment tool to rapidly assess the pipeline becomes imperative.

The objective of this project is to look at advanced pipelines and the close coupling of the compression system with advanced prime mover cycles. The investigation offers a comparative assessment of traditional and novel prime mover options including the design and off-design performance of gas turbine engine and the economic analysis of the system. The originality of the work lies in the technical and economic optimization of gas turbines and fluid movers, based on current and novel cycles for a novel pipeline application in a wide range of climatic conditions.

The techno-economic and environmental risk assessment (TERA) tool created is made up of a number of modules, starting with the pipeline and compressor station modules which compute the necessary flow parameters and compressor performance, as well as the required compressor power. The next is the gas turbine performance simulation module, TURBOMATCH software was used to simulate the performance of the selected gas turbine engines at design and off-design conditions and it computes the thermodynamic conditions in the core of the engine. Receiving information from the performance simulation module, the emission module, which is based on combustion equations, estimates the amount of emission over the period of operation of the gas turbine. The economic module, which is essential to the tool, receives information from all the other modules to establish the life cycle cost and use the net present value (NPV) methodology to assess the plant. It also calculates all associated costs, as well as the cost of transporting natural gas. The economic

module establishes the economic pipe size for any particular throughput. The electric motor drive module is the parallel arm of the methodology, handling all the modules as explained earlier except the gas turbine performance and emission modules. This allows a comparative assessment of gas turbine and electric motor drives to be carried out under any prevailing conditions. This methodology is unique to natural gas pipeline techno-economic assessment and no previous studies have looked at various aspects of the pipeline project before selecting a prime mover or an economic pipe size.

This study further uses a genetic algorithm optimization tool to optimize gas turbine selection and compressor station location along the pipeline, based on total cost objective function. The optimization is limited to a particular pipe size and gas throughput. The use of various pipe sizes as well as varying throughput will be a major area for further studies.

The results from the individual models are presented in chapters 3, 4 and 5. The result of the integrated modules under case study one and two shows that the transportation of 0.5 million cubic meter per day of natural gas over long distance interstate pipeline for both prime movers is uneconomical. The economic pipe size for 3.0 million cubic meter per day of natural gas is 609.6 mm (24”) pipe size for the two prime movers with transportation cost of \$0.063/m^3 and \$0.056/m^3 for gas turbine and electric motors respectively. This is equivalent to \$1.46/GJ and \$1.30/GJ which agrees with the cost of natural gas transportation in literature.

The result of the optimization shows a clear preference for the selection of a 34 MW plant for the pipeline and throughput considered since this presents the minimum cost which is the definition of the genetic algorithm optimizer.

It is worth noting that this techno-economic tool, which is made of many modules, can be used to rapidly assess the profitability or otherwise of a natural gas pipeline project.

Acknowledgement

All praise and thanks is due to Allah, the Lord of the worlds, the most Beneficent the most Merciful and may the abundant peace and blessings of Allah continue to abide by the holy prophet Muhammad, his household, his companion and those following his footsteps till the last day.

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All praise and gratitude, in first and last belongs to Allah whom is the FIRST, the LAST, the MANIFEST and the HIDDEN. May the peace and blessing of Allah be on our Noble Prophet Muhammad, his household and companions.

Dedication

This thesis is dedicated to my wife – **Mrs Hauwa Talatu Abdulkarim** and my children

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Nomenclature

A	pipe cross-sectional area
Amb. T.	ambient temperature
bcf	billion cubic feet
C_{COMPRE}	compressor station cost
C_{EMCAP}	electric drive capital cost
C_{EMkW}	electric drive energy cost
C_{EMOM}	electric drive maintenance cost
C_{GTCAP}	gas turbine capital cost
$C_{GTEMISON}$	gas turbine emission cost
C_{GTFUEL}	gas turbine fuel cost
CO_2	carbon dioxide
C_T	pipe material cost per ton
E_{FUEL}	fuel energy consumed
D	pipe inside diameter
f	friction factor, dimensionless
F	transmission factor
G	gas gravity
GT	gas turbine
i	number of years
L	pipe segment length
m	mass flow
n	polytrophic entropy
N_1	driver rotational speed
NO_x	oxides of nitrogen
P_1	upstream pressure, kPa

P_2	downstream pressure, kPa
P_b	base pressure, kPa
PMC	pipe material cost
Q	gas flow rate, measured at standard conditions, m^3/day
r	discount rate
RPM	revolutions per minute
t	pipe thickness
T_b	base temperature, K (273 +°C)
T_f	average gas flowing temperature, K (273 + °C)
V_A	velocity of flow at point A
V_B	velocity of flow at point B
Z	gas compressibility factor at the flowing temperature, dimensionless

Acronyms

AC	Alternating current
Bcf	Billion cubic feet
bcm	Billion cubic meter
BP	British Petroleum
DC	Direct current
LNG	Liquefied Natural gas
MAOP	Maximum allowable operating pressure
tcf	trillion cubic feet
TMR	TURBOMATCH Result

1 INTRODUCTION

1.1 Background of the Study

The increase in the world’s population has led to a rapid increase in energy demand. At the same time the concern about global warming associated with the release of greenhouses gases into the atmosphere constrains the use of fossil energy to satisfy this increasing demand. Natural gas is considered a clean alternative energy resource to coal and oil. It is expected that global consumption of gas will double by 2030 (Riva et al., 2006). Alternative forms of energy have been proposed, and are increasing their market share. However, even alternative energy sources can create environmental problems and have issues with cost and availability (Rajnauth et al., 2008). Natural gas, which is a naturally abundant resource, is likely to gain preference over liquid and solid fuels because of its less negative impact on the environment when burnt. It is a widely used fuel for heating in industry as well as homes

The efficient and effective movement of natural gas from production regions to consumption regions requires an extensive and elaborate transportation system. The traditional and easiest way of transporting this important energy resource to industries and consumers from the gas treatment plant is by pipeline.

The transmission of natural gas to consumers through pipelines, which may span several kilometres, requires compressor station(s) to ensure the continuous flow of the gas. Presently, one of the methods of providing drive power for this application involves using aero-derivative gas turbines, powered by the very gas being pumped. The gas turbine is usually a single spool with power turbine which drives the pipeline compressor (Wilson and Baumgartner, 1999). Currently, gas turbine is prominent as a prime mover in power generation, in the oil, gas and petrochemical industries. In the past three decades, the advantages of the gas turbine, in terms of flexibility, capacity, and its impact on the environment, has outweighed its disadvantages which are high component costs and maintenance. The challenges offered by the requirement of monitoring these machines are not encountered on other machines, and a full knowledge of different types of gas turbine is necessary in order to understand these fully (Marquez Jr et al., 2000).

Gas turbines, fired by natural gas, are among the dominant prime movers in a world concerned about emissions, energy efficiency and energy reliability. The flexibility of industrial gas turbines not only makes them the driver of choice for electrical power generation, but also for applications related to gas production, gas transmission and gas transportation (John and Rainer, 2000).

The present research looks at technical and economic issues surrounding the use of gas turbine as a prime mover for pipeline compressor vis-à-vis electric motor driven compressor.

1.2 Aims and Objectives

The main aim of this research is develop a decision making tool to guide the selection and operation of prime movers for natural gas pipeline in order to minimize capital and operating cost, and maximize profit. The objectives of this research are:

1. To investigate the performance of aero derivative gas turbines used as prime movers for pipeline compressors under design and off-design conditions,
2. To investigate techno-economically the usage of gas turbine and electric motor as prime movers in natural gas pipeline system,
3. To carry out emission studies of the prime mover options under investigation.
4. To optimize compressor station positioning along a gas pipeline.
5. To optimize power plant size for pipeline compression based on least cost.

1.3 Statement of the problem

The increase in global energy requirements and specifically those of Europe and America, owing to economic growth and the concern for environmental protection, has geared up the need for the usage of a fuel which is environmentally friendly. Natural gas meets this condition far more that oil and coal because it contains less sulphur and with less or no unburnt hydrocarbon. This gives a boost to the demand for this commodity. It becomes imperative to develop further this mode of gas transportation system for moving huge quantities of natural gas from different gas producing nations to those regions where demand is high. To meet this huge supply requires the following conditions:

- A constant supply of clean energy with enough reserves to meet the requirements of peak demand periods.
- Minimum capital costs of transportation systems.
- Minimum operating and integrated logistic system costs.
- High utilization of project facilities with efficient alternatives for increasing capacity.

To meet these needs it is imperative to answer questions such as:

- Which is a more appropriate prime mover option: gas turbine or electric motor under the prevailing conditions?
- How many compressor stations will be required and at what optimum positions along the pipelines?
- What gas turbine engine will best serve as prime mover under varying operational conditions?

All these questions will be addressed in relation to the Techno-Economics of the system.

1.4 Justification

The ability to transport hydrocarbon has always been an important factor in the successful development of an oil or natural gas field, both off-shore and on-shore. Historically, pipelines have been the common means used to transport petroleum products in most countries. In doing this, compressor stations are necessary along the route to keep the gas flowing. Bearing in mind the natural gas inlet pressure into the pipeline and the expected delivery pressure, it is beneficial to optimize the compressor station positions along the pipeline so as to avoid excess gas pressure and also to ensure that the gas pressure does not drop below the delivery pressure at any point. Matching the compression system with the right prime mover and to assessing comparatively the techno-economics of using gas turbine and electric motors as prime movers are also of utmost importance in the development of successful natural gas transportation system. This will eventually lead to the proper selection of gas turbine type as well as the technical and economic advantage of using this prime mover option. The optimization of the pipeline system using a TERA model integrated by GA will minimize cost and consequently reduce the unit cost of the commodity.

1.5 Thesis Structure

The final thesis structure is presented below. It is structured such that all the elements of the research are properly presented to facilitate maximum understanding and comprehension.

Chapter One gives a general overview of the thesis under the headings: background of the study, aims and objectives, statement of the problem, justification for the study and thesis structure.

Chapter Two presents a review of previous related studies. Natural gas reserves, consumption and existing modes of natural gas transmission are extensively reviewed. Pipe flow analysis, compressor types and performance and compressor station operating limits are reviewed. A review of gas turbine as a prime mover for pipeline compressor is presented.

Chapter Three looks at the concept of Techno-economic and Environmental Risk Analysis (TERA) in relation to gas turbine and electric motor. It presents the design and off-design performance modelling with due consideration to daily variable ambient conditions which obviously affects the power output of the gas turbine. It also presents electric motor drive option, including motor performance and transmission loss. The effect of electric transmission distance and consequently power loss is analysed and discussed. The sources of electricity vis-a-vis off-site emission issues as it relates to electric motor are discussed. Gas turbine emissions and its environmental and economic implication are presented.

Chapter Four addresses the capital investment appraisal technique and natural gas pipeline economics.

Chapter Five presents the techno-economic module development and the integration of all the modules to form a TERA methodology. Case study I and II which relates to gas turbine and electric motor respectively are presented and results fully discussed.

Chapter Six addresses the genetic algorithm optimization technique with the results presented and discussed.

Chapter Seven presents conclusion of the study as summary of methods and summary of results. Recommendations for further studies are also presented.

2 NATURAL GAS RESERVES AND CONSUMPTION

This chapter summarizes a number of works in the research areas of gas flow in pipelines, gas turbine as a prime mover of choice, electric motors and a host of other areas of research development that relates to the present work. Figure 2.1 shows the natural gas reserves by country. It is clear that Europe and America, which have the highest consumption of natural gas (Figure 2.2), ironically have almost the lowest reserve of the same commodity. It therefore follows that for this demand to be met, natural gas must be transported from regions with large reserves to Europe and America. A pipeline system is generally used mode of natural gas transportation.

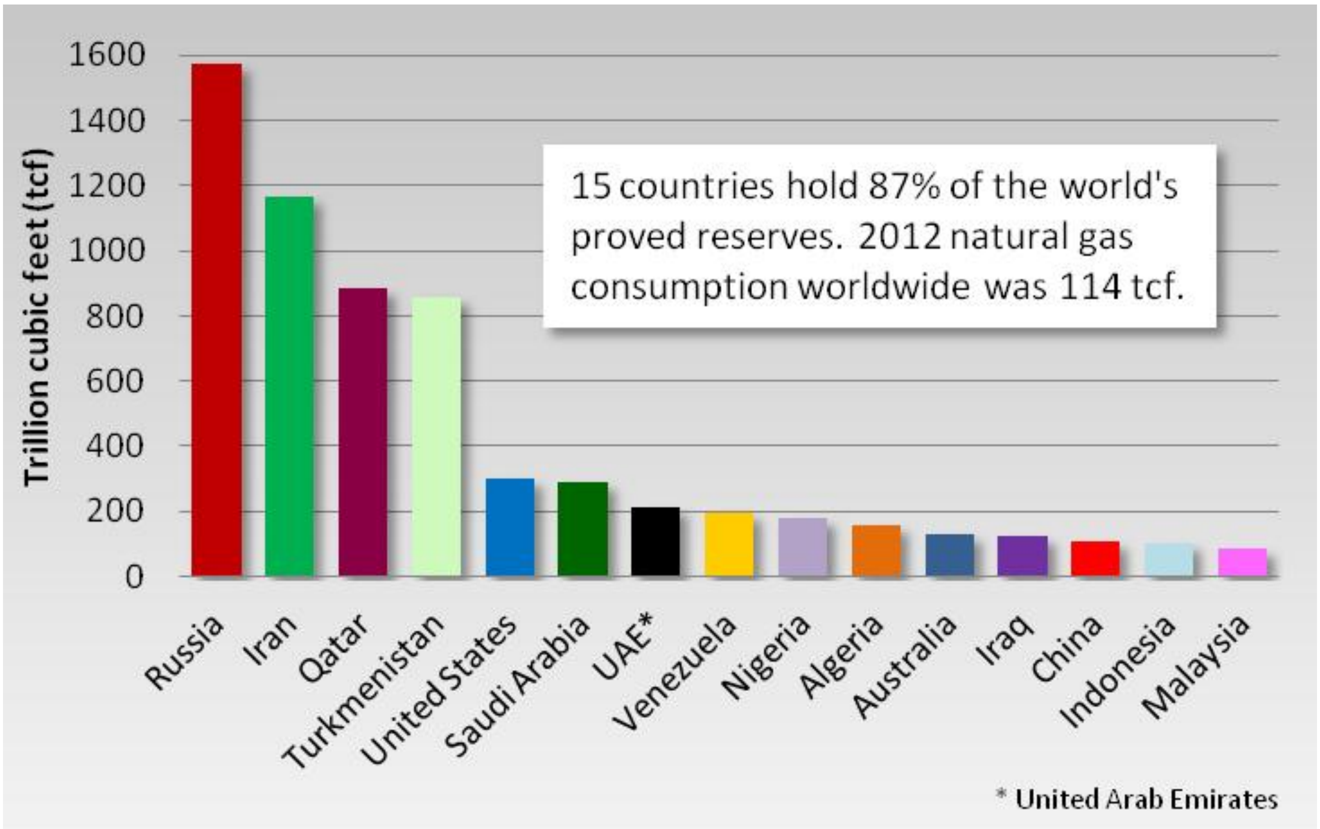

Figure 2-1: World Reserves of Natural Gas (White, 2012)

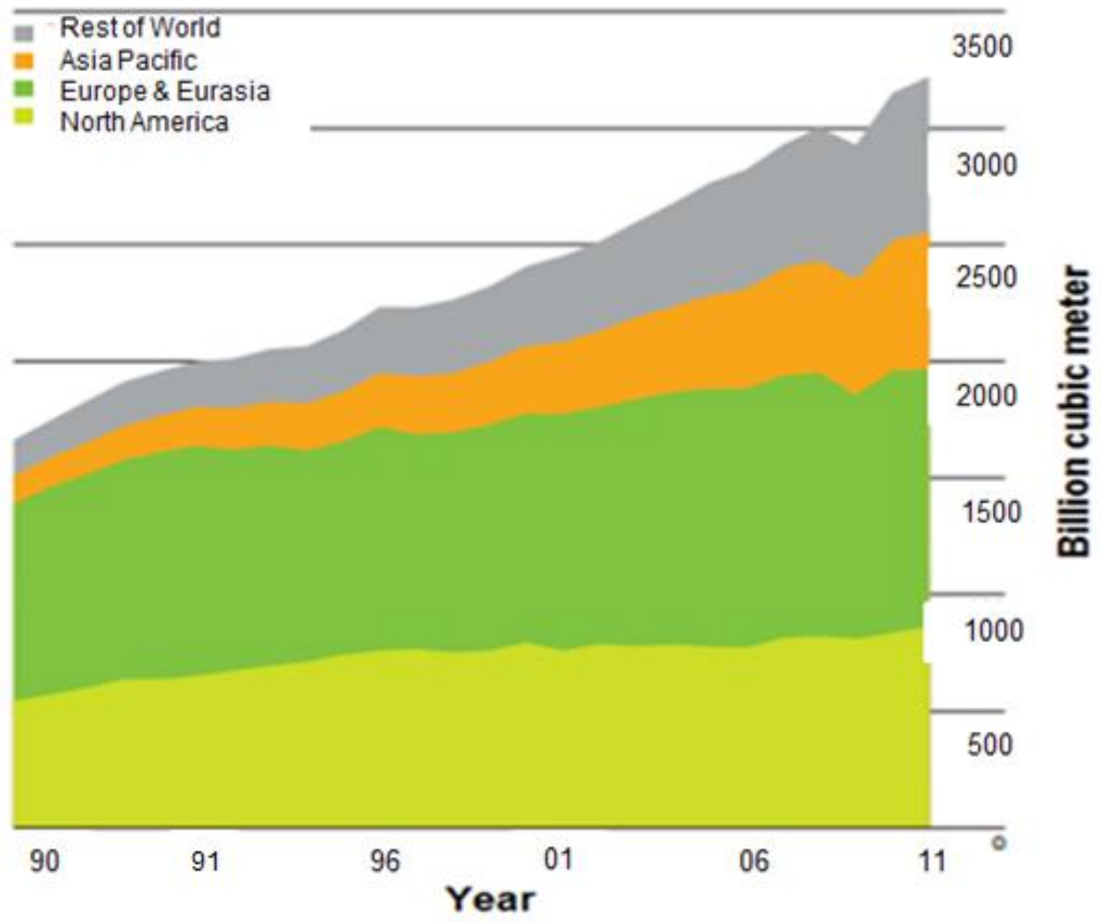

Figure 2-2: Natural gas consumption by region (BP, 2012)

2.1 Natural Gas Transmission/Transportation System

Natural gas is a versatile form of low polluting fuel and it is fast becoming the world's fuel of choice. About 45% of households in North America use natural gas to heat their homes and cook their food. It is also use as a fuel for cars and more importantly it is now employed for power generation owing to its environmental friendliness when burned. The major drawback in the use of natural gas is the transportation and storage owing to its low density. From the production point of natural gas through process plant, there exist several miles between it and the end users. A cost-effective means of transport is essential to bridge the gap between the producer and consumer, although the efficient and effective movement of natural gas from gas treatment plant to consumer regions requires an extensive and elaborate transportation system. The common method for transporting natural gas has been high pressure in underground pipelines, other methods employed include liquefied natural gas, compressed natural gas, gas to solids (hydrate), gas to liquid, gas to wire and other gas to commodity methods (Rajnauth et al., 2008).

Thomas and Dawe (2003) reviewed the available technologies of gas transportation such as Compressed Natural Gas (CNG), Natural Gas Hydrate (NGH), Gas To Liquid (GTL), and Gas To Wire (GTW), Liquefied Natural Gas (LNG) and Pipeline Natural Gas (PNG). (Seungyong, 2001) compared CNG, NGH, GTL, and (GTW) with Liquefied Natural Gas (LNG) and Pipelined Natural Gas (PNG) technologies to derive the most economical one in selected regions such as Mediterranean, Caribbean, Arabian Seas, and Sakhalin Island. The paper concluded that for specific location and economic basis that CNG, NGH and GTW appears to represent the an important approach for producing and transporting the stranded gas instead of gas re-injection. Figure 2-3 shows the gas transportation options between source and customer.

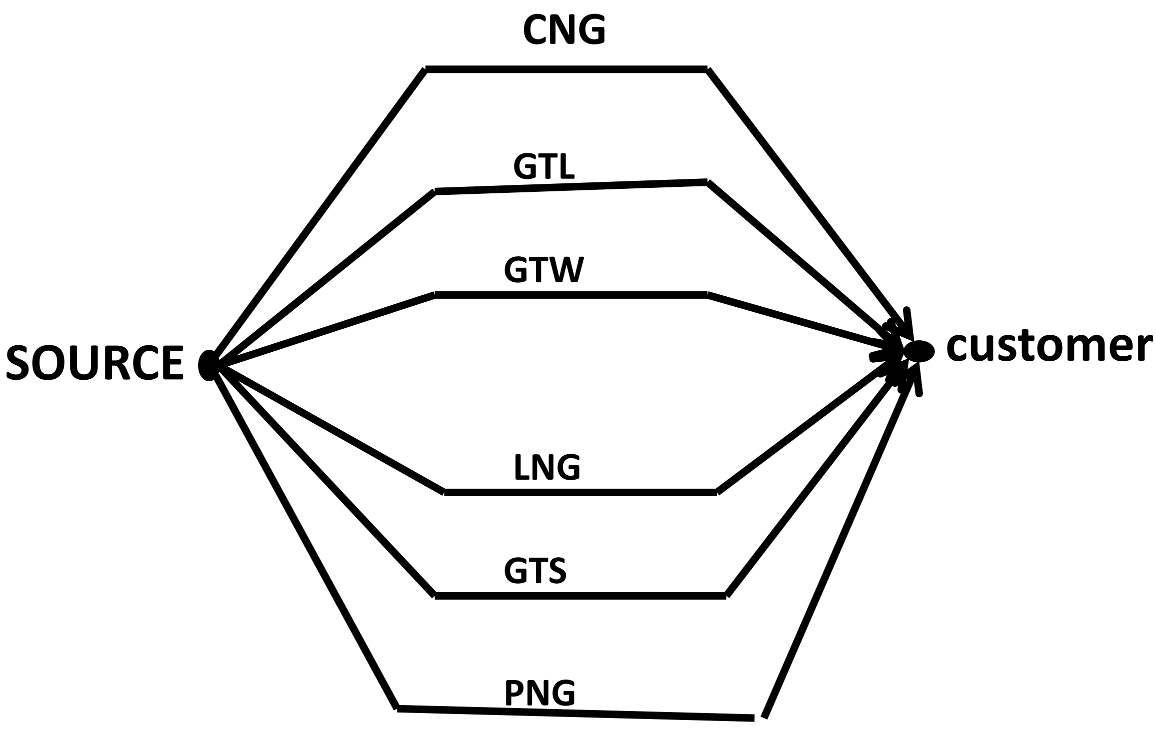

Figure 2-3: Natural gas transportation options

2.1.1 Compressed Natural Gas (CNG)

CNG is natural gas under pressure which remains clear, odourless, and non-corrosive. Pressures in CNG vessels ranges from 130 bar for rich gas to 250 bar for lean gas. CNG can be referred to as a technology for storing and transporting natural gas usually in the marine system. The development of this technology was met with technical difficulties combined with heavy investment required to build its facilities (Dunlop and White, 2003). This retarded the development until the discovery by Cran and Stenning of a new type of pressure containment vessel called a "Coselle", which promised to make CNG transportation attractive. A cost effective solution may be provided for natural gas transportation by compressing it into large diameter Type-4 pressure vessels secured inside of an Intermodal shipping container.

The CNG supply chain includes:

- Gas production
- Compression facilities
- Refrigeration/ ship loading facilities
- Transportation
- Ship receiving facilities
- Storage facilities

Downstream gas distribution facilities (Rajnauth et al., 2008). Figure 2-4 shows a CNG tank.

CNG is technically simple, requiring lower requirements for facilities and infrastructure. The CNG concepts that are found in the literature include:

- Coselle by Cran and Stenning
- VOTRANS by Enerseas (Volume Optimized Transport and Storage)
- GTM by TransCanada (Gas transport Module)
- CRPV Technology (Composite Reinforced Pressure Vessel)
- PNG Technology (Pressurized Natural gas Vessel)

For further information on these CNG technologies, refer to www.coselle.com, www.enersea.com, www.transanscanada.com.

Figure 2-4: CNG Vessel (MARINELOG, 2006)

2.1.2 Gas to Liquid (GTL)

GTL technology is a process which involves the conversion of natural gas or other gaseous hydrocarbons into liquid products such as gasoline or diesel fuel for transport, power and expanded chemical feedstock applications (Subero et al., 2004). Catalytic processes are employed in the conversion of gas to liquid which unlike products refined from crude oil produces clean fuels free from sulphur and aromatic pollutants (Wagner and and Wagensveld, 2002). The process to convert gas to liquid has two stages:

- Natural gas is converted into a synthesis gas (syngas), which is a mixture of carbon monoxide and hydrogen

- Catalytic reactions are necessary to convert syngas into hydrocarbons (Subero et al., 2004).

Figure 2-5 shows a simplified process schematic of GTL.

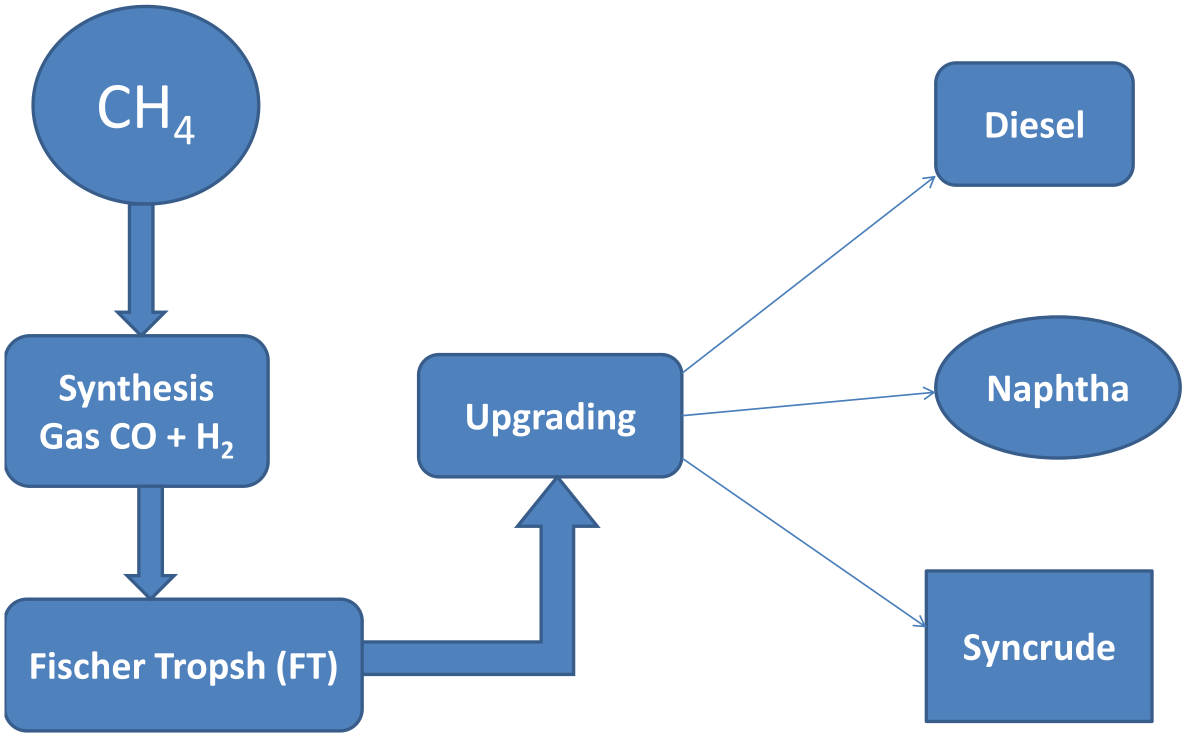

Figure 2-5: GTL Simplified process schematic

2.1.3 Gas to Wire (GTW)

GTW is a mode of transmitting natural gas energy, which involves the conversion of natural gas to high voltage direct current (DC) and transported to the market where it is converted locally for AC use, this is considered to be a high efficiency system with merits from economic and environmental perspective. The replacement of high power lines with DC cables is recommended considering the high cost of high power lines from offshore platform to the shoreline and also the fact that DC requires less core number (Rajnauth et al., 2008). GTW means onsite power generation by produced gas. Figure 2-6 shows gas to wire technology.

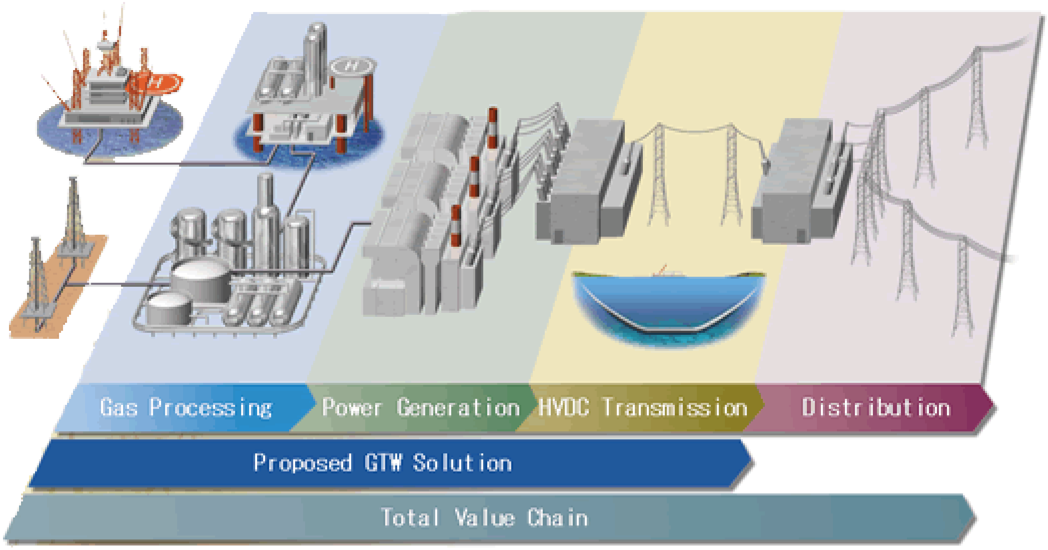

Figure 2-6: GTW Technologies (Watanabe et al., 2006)

GTW is suitable for short distance with minimum gas volume, see Figure 2-7.

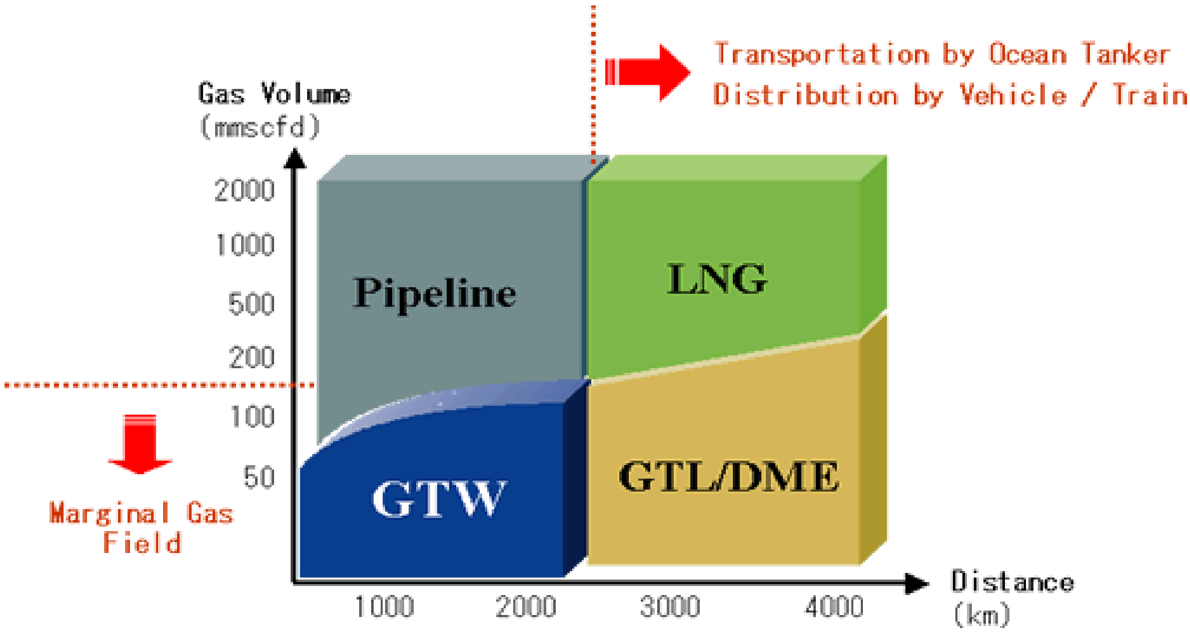

Figure 2-7: Applicable range of GTW (Watanabe et al., 2006)

2.1.4 Liquefied Natural Gas (LNG) Technology

Liquefied natural gas (LNG) is natural gas that has been super cooled to -162^{o}C. Natural gas exists in the liquid state at this temperature and its volume is 1/600 of its volume in the gaseous form. This gives the advantage for transporting it from regions with high reserve to those part of the world with high demand.

The lower NO_x and CO_2 emissions of natural gas compared to other fossil fuels such as oil or coal makes it the world's cleanest fossil fuel and consequently an energy resource of choice. LNG is transported predominantly via LNG ship in which are built vessels of two or three layers with different materials. These vessels consist of an outer hull, an inner hull of stainless steel material and a cargo containment system (Shelley, 2007). Figure 2-8 shows LNG ship.

Figure 2-8: LNG Ship (Sempra LNG, 2010)

2.1.5 Gas to Solid (GTS) – Natural Gas Hydrate

Natural Gas Hydrates are non-flowing crystalline solids that are denser than typical fluid hydrocarbons and are composed of water and natural gas. The gas molecules they contain are effectively compressed and exist within cages of water molecules. Dry hydrate can be produced by two processes viz:

- Primary stage to achieve bulk separation
- Followed by a secondary dewatering unit to complete the process.

In dry state, gas hydrates can be stored for a long time and shipped at atmospheric pressure and a temperature of about -40 with very good insulation.

2.1.6 Natural Gas Pipelines Transportation System

Natural gas transportation system is made up of compressor stations, pipelines, city gate stations and storage facilities. The compressor station serves as the heart of the transmission system as it supplies the energy required to ensure that the gas continues to flow at a prescribed flow rate and pressure. The onshore and/or offshore pipelines have THREE (3) types: trunk or gathering, transmission, transportation and distribution pipelines.

Natural gas pipelines primarily serve as a means of moving gas from the field to consumers. Inter – and intra- state pipelines are used for the transportation of natural gas produced from gas fields either onshore or offshore facilities through gathering systems to commercial, residential, industrial, and utility companies. The pipelines are usually constructed of carbon steel and varying in size from 2 inches (51 mm) to 56 inches (1,400 mm) in diameter, depending on the type of pipeline.

Figure 2-9 shows the transportation efficiency for different gas transportation methods. Pipeline transportation has highest efficiency for pipeline length of less than 1000 km although pipeline transportation is adjudged to be suited for short distances less than 3000 km (Rajnauth et al., 2008).

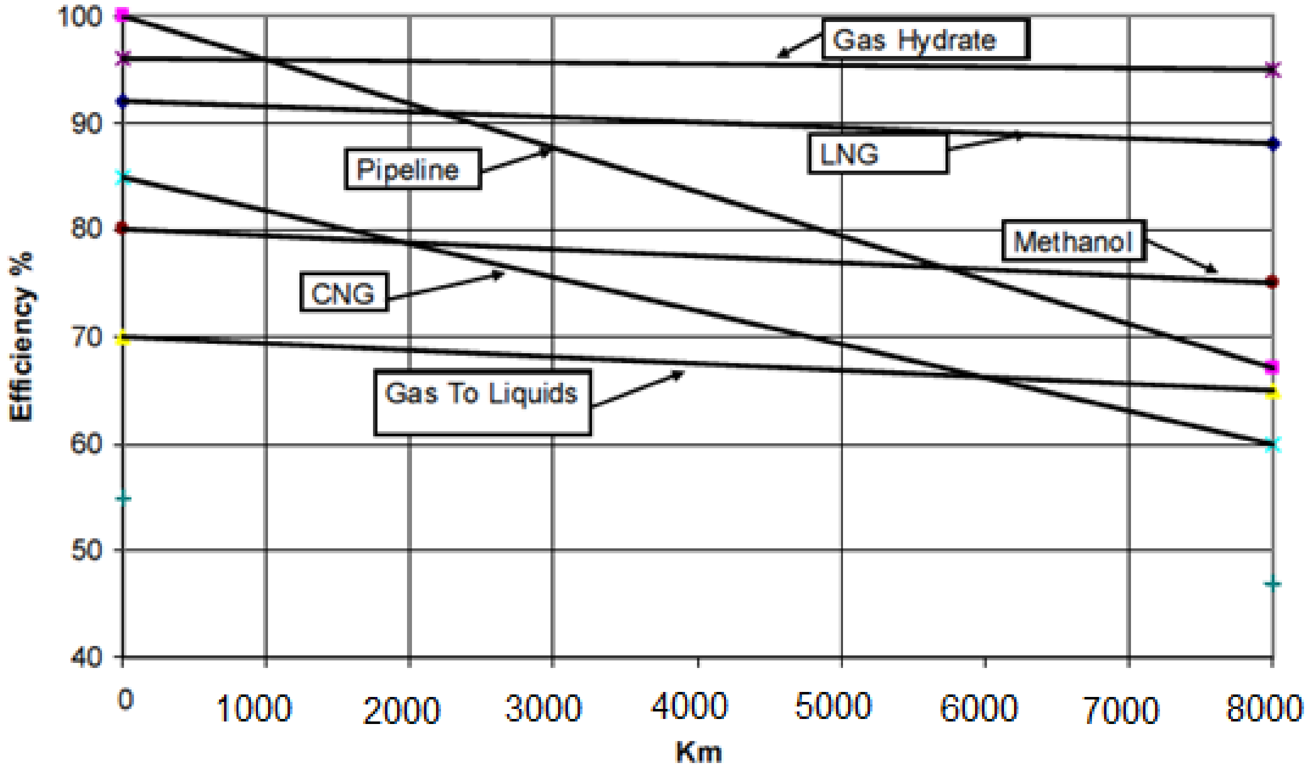

Figure 2-9: Transportation Efficiency for different Gas Transportation Modes (Rajnauth et al., 2008)

A pipeline system refers to a pipeline section which extends from an inlet point (offshore platform or onshore compressor station) to an outlet point (another platform or an onshore receiving station). It can also extend to refer to pipeline from gas treatment plant through gas scrubbers, liquid removal tank, compressor stations, valves, intercooler, after cooler and city gate into the distribution line and then to the consumers. Figure 2-10 shows a pipeline layout.

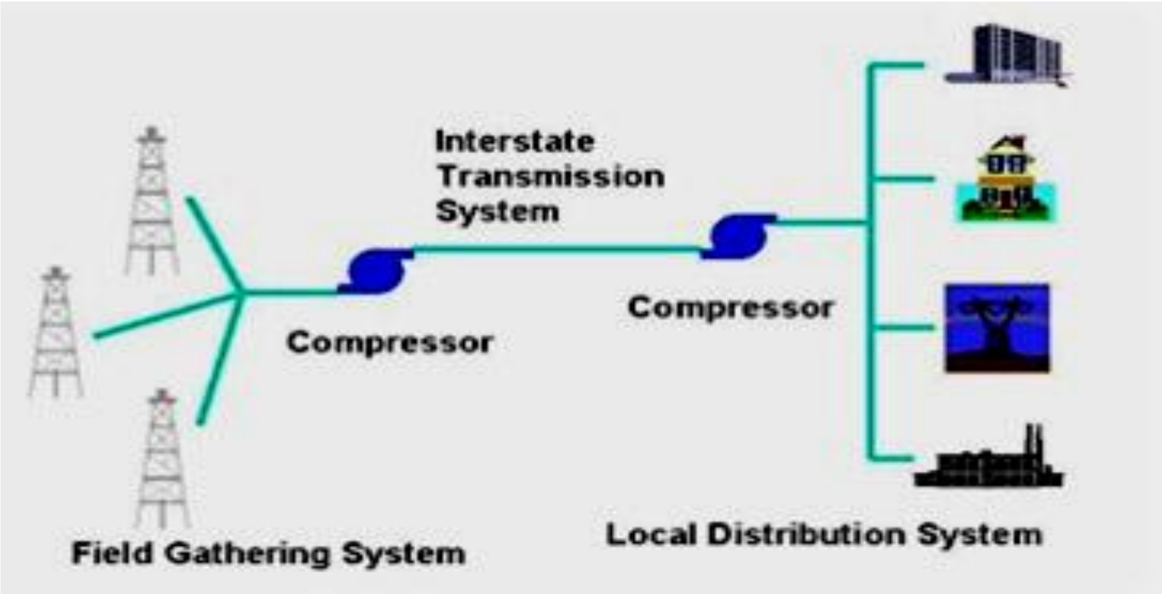

Figure 2-10: Pipeline layout (Watanabe et al., 2006)

The factors that affect the type of gas transportation system used include gas reserve, time frame to monetize the gas, the distance to the market, investment and infrastructure available and gas processing.

(Rajnauth et al., 2008) opined that the amount of natural gas reserve available could determine the gas transportation mode to be adopted. Table 2-1 presents a compilation of the estimated amount of gas reserves needed for each transportation mode.

Table 2-1: Reserves required for gas transportation mode (Rajnauth et al., 2008)

Transportation Mode	Amount of Gas needed for project
Pipeline	Depends on distance
LNG	>1-3 tcf
GTW	10 bcf – 1 tcf
GTL	>500 bcf
NGH	>400 bcf
CNG	>> 300 bcf
GTC	< 1 tcf

Watanabe et al. (2006) suggested a new development concept -Gas to Wire (GTW) of marginal gas field and associated gas with a reserve between 10 bcf and 1 tcf. Bearing in mind the technicalities of gas transportation, the economically attractive gas transportation mode will depend on a number of parameters viz: reserve base, production capacity and distance between the gas source and the consumers. Subero et al. (2004) examines and quantifies how distance, reserves, and gas production rate affect the technology selection and project economics. They concluded, after analyzing the different modes and determining which option delivers the highest net present value, that pipeline and CNG projects are best suited to maximize returns for shorter distances (up to 2000km); and LNG is better suited for longer distance projects.

Considering the importance of gas supply all over the world, one of the challenges pertains to the capacity of the industry to ensure continuous delivery of natural gas while its demand is steadily increasing. It therefore becomes clear and unambiguous that pipelines have become the popular means of transporting natural gas from the wellhead to processing — and from there to the final consumer — since it better guarantees continuous delivery and assures lower maintenance costs. Table 2-2 shows the present stages of the different gas transportation methods.

Table 2-2: Stages of present gas transportation modes (Rajnauth et al., 2008)

Mature	Developing	Future
Pipeline	Gas to Liquid (GTL)	Gas Hydrate
Liquefied Natural Gas (LNG)	Compressed Natural Gas (CNG)	
Gas to Wire	Gas to Products	

Pipeline networks are composed of equipment that operate together to move the gas from location to location. Fig 2-12 shows the main item/equipment that make up gas pipeline system. These include

i. Initial Injection Stations – this is the beginning of the system, where the product is injected into the line. This location usually housed compressors, pumps and storage facilities.
ii. Compressor Station – Compressors are located on the network to keep the gas flowing
iii. Partial Delivery or Intermediate stations – this station is only available on the pipe network if the need be to deliver one of the products before the final delivery station
iv. Block Valve Station- This is the first line of protection for the pipelines. This allow for the isolation of any part of the network for maintenance work.
v. Regulator Station - A valve station along the pipeline for release of pressure.
vi. Final Delivery Station – this is referred as the city-gate, from where the gas is passed into the distribution line for final delivery to consumers.

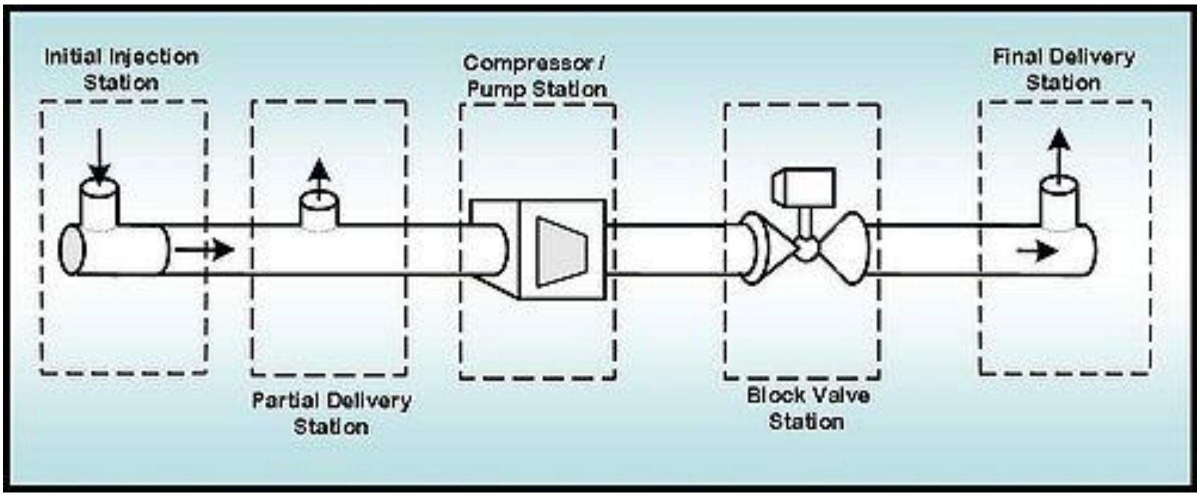

Figure 2-11: A pipeline schematic (Rajnauth et al., 2008)

When gas flows through the network, it suffers energy and pressure losses due to the friction between the gas and the inner walls of gas ducts but also due to the heat transfer between gas and the environment. If demanded gas has to be supplied to delivery points with a specified pressure, the undesired pressure drops along the network must be periodically restored. This task is performed by compression stations installed on the network, but these usually consume over 3% or 5% of total gas transported. The pressure of natural gas pipelines was only 2.5MPa before it rose to 6MPa in the sixties. At present, the pressure of pipeline is about 10MPa (Liyong and Yingbai, 2008).

2.1.7 World Natural Gas Pipelines

There are natural gas pipeline in every continent of the world. The reason of which is to transport natural gas from the production site through treatment/processing plant to the end users. Presented below are some of the existing natural gas pipelines.

- **West African Gas Pipeline**
  This is a three section pipeline of total length of 678 kilometres (421 mi), having 569 kilometres (354 mi) long offshore section and the rest onshore. The pipe diameter is 20 inches (510 mm) offshore and 30 inches (765 mm) for onshore with a capacity of 5 billion cubic meter (bcm) of natural gas per year.
- **Arab Gas Pipeline**
  This is the main natural gas export pipeline from Uzbekistan. It's a 40 inch (1020 mm) pipe with an average delivery of almost 22 bcm of natural gas per year.
- **Central Asia – China gas pipeline**
  This pipeline which connects Central Asia to Xinjiang in the Peoples Republic of China span 1,833 kilometres (1,139 mi) and has a diameter of 1,067 mm (42.0 in). The capacity of this pipeline is about 40 bcm of natural gas per year
- **Trans Thailand–Malaysia Gas Pipeline**
  This Gas Pipeline links the suppliers in Malaysia to consumers in Thailand. The pipeline has a diameter of 34 to 42 inches (860 to 1,070 mm). It is a part of the Trans-ASEAN Gas Pipeline project (BO, 2010).

- **Europipe** **I**

  A 670-kilometre (420 mi) long natural gas pipeline from the North Sea to Continental Europe., has a diameter of 40 inches (1,000 mm) and capacity of about 18 billion cubic meters of natural gas per year (Gilardoni et al., 2008).
- **Langeled pipeline**

  The world's longest underwater pipeline through which Norwegian natural gas is transported to the United Kingdom. The pipe ranging from 1,067 mm (42.0) to 1,118 mm (44.0 in) diameter runs a length of 1,166 kilometres (725 mi) through the North Sea from the Nyhamna terminal in Norway (Acergy, 2006) and has annual capacity of 25.5 bcm (Wojciech, 2007).
- **Trans Austria Gas Pipeline**

  Two parallel pipelines of pipeline is 380 kilometres (240 mi) long transports about 41 billion cubic meters (bcm) of natural gas annually from the Slovak-Austrian border at Baumgartner to Arnoldstein in the south, near the border with Italy
- **Trans Mediterranean Pipeline (TMP)**

  TMP is a natural gas pipeline from Algeria via Tunisia to Sicily and thence to mainland Italy. The pipe diameter ranges from 48 inches (1,220 mm) from Algerian section through Tunisia to 20 inches ((510 mm) at the offshore section across the channel of Sicily and mostly made of two parallel pipelines with delivery capacity of about 30.2 billion cubic meter (bcm) of natural gas per year (Domenico, 2002).
- **TransCanada pipeline(TCP)**

  TCP is a 48 inches (1219 millimetres) diameter pipeline system carrying gas through Alberta, Saskatchewan, Manitoba, Ontario and Quebec. It is the longest pipeline in Canada.

2.2 Pipeline Design

Designing a gas pipeline requires in-depth knowledge of the head loss brought about by fluid friction, and also imperative to considered variables such as length, diameter, slope, the gas being transported, etc. If the distance is great, the changes in pressure are important, it becomes necessary to-approach the problem using concepts of compressible fluid flow. The fact that pipelines are metallic and go underground allows the analysis to assume an isothermal flow at ground temperature.

The pipeline throughput (flow rate) will depend upon the gas properties, pipe diameter and length, initial gas pressure and temperature, and the pressure drop due to friction. Few commonly used model equations will be reviewed. The design of a new gas pipeline transmission system or the expansion of an existing one requires the optimization of its topology, size and operational conditions to minimize investment and maintenance cost. Traditional solutions to this problem use gradient-based techniques. An acceptable convergence for these methods depends on the initial values given by the designer (Montoya-O et al., 2000)

2.3 Gas Flow Analysis

As gas flows through a pipeline, the total energy of the gas at various points consists of energy due to pressure, energy due to velocity, and energy due to position or elevation above an established datum. Bernoulli's equation simply connects these components of the energy of the flowing fluid to form an energy conservation equation. Bernoulli's equation is stated as follows in equation 2.1, considering two points, 1 and 2, as shown in Figure 2-12.

$$Z_A + \frac{P_A}{\gamma} + \frac{V_A^2}{2g} + H_P = Z_B + \frac{P_B}{\gamma} + \frac{V_B^2}{2g} + h_f \qquad (2.1)$$

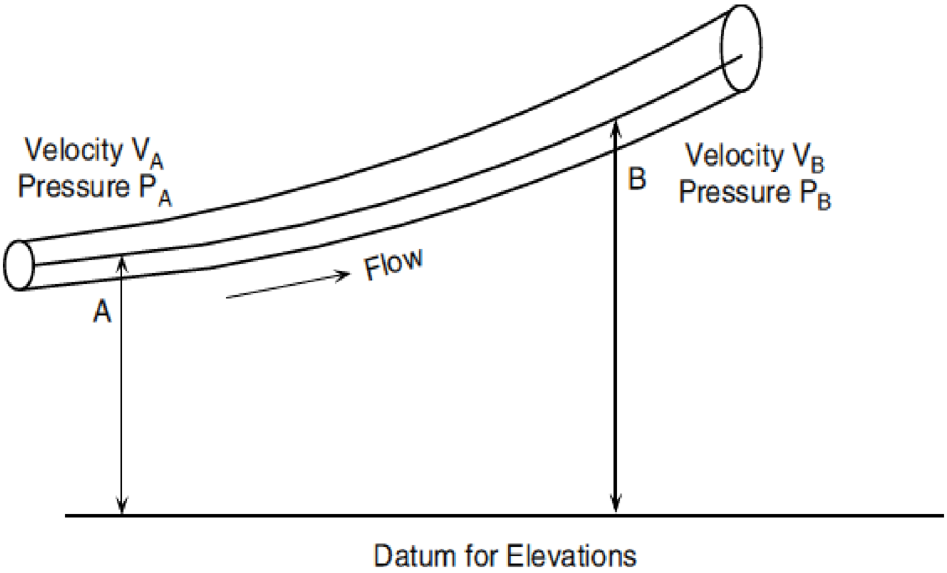

Figure 2-12: Energy of Flowing Fluid

Where H_P is the equivalent head added to the fluid by a compressor at A and h_f represents the total frictional pressure loss between points A and B.

Flow equations have been developed over the years by considering the basic energy equation 2.1 and applying the gas laws to predict the performance of a pipeline transporting gas. These equations are intended to show the relationship between the gas properties, such as gravity and compressibility factor, with the flow rate, pipe diameter and length, and the pressures along the pipeline. Thus, for a given pipe size and length, we can predict the flow rate possible through a pipeline based upon an inlet pressure and an outlet pressure of a pipe segment. Simplifications are sometimes introduced, such as uniform gas temperature and no heat transfer between the gas and the surrounding soil in a buried pipeline, in order to adopt these equations for manual calculations. Although transient situations are experienced in gas flow in pipelines, for practical purposes, the assumption of isothermal flow is good enough, since in long transmission line the gas temperature reaches steady state or constant values, anyway.

2.3.1 Equations of Flow

The analysis of flow and pressure drop in piping systems has been studied by many workers and usually based upon the consideration of steady state conditions. The transient behaviour in a pipe flow, conventionally taking the form of a set of second-order partial differential equations, can be expressed by a set of first-order ordinary differential equations using the new model developed by (Ke and Ti, 2000).

They adopted the electrical analogy method by combining resistance with the theoretically derived models of capacitance and inductance to establish a set of first order ordinary differential equations for transient analysis of isothermal gas flows in pipeline network. Optimum design of a gas transmission pipeline requires methods for predicting pressure drop for a given flow rate or predicting flow rate for a specified pressure drop in conjunction with installed compression power and energy requirement. In other words there is a need for practical methods to relate the flow of gas through a pipeline to the properties of both the pipeline and gas and to the operating conditions, such as pressure and temperature. Isothermal steady-state pressure drop or flow rate calculation methods for single-phase dry gas pipelines are widely used and are the basic relationships in the engineering of gas delivery systems (Beggs, 1984; Aziz and Ouyang, 1995; Smith, 1983).

Several equations are available that relate the gas flow rate with gas properties, pipe diameter and length, and upstream and downstream pressures. Some of these equations include among others

1. General Flow equation
2. Colebrook-White equation
3. Modified Colebrook-White equation
4. AGA equation
5. Weymouth equation etc. (Menon, 2005)

The General Flow equation for the single-phase steady-state isothermal flow in a gas pipeline is the basic equation for relating the pressure drop with flow rate.

Figure 2-13 shows a schematic of a pipe depicting steady flow. The common form of this equation in S.I is given in terms of pipe diameter, gas properties, pressures, temperatures and flow rates below (Mokhatab et al., 2006).

$$Q = 1.1494 \times 10^{-2} \left(\frac{T_b}{P_b}\right) \left[\frac{(P_1^2 - P_2^2)}{fGT_aLZ_a}\right]^{0.5} D^{2.5} \qquad (2.2)$$

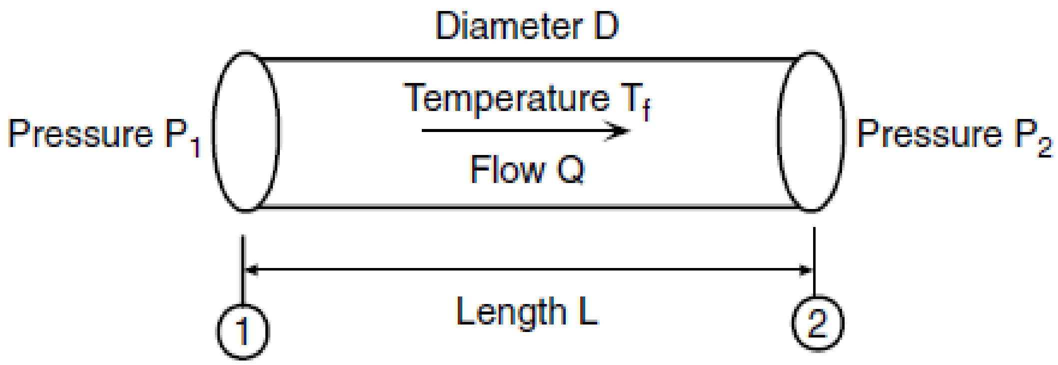

Figure 2-13: Steady flow in gas pipeline

For the pipe segment from section 1 to section 2, the gas temperature T_a is assumed to be constant (isothermal flow).

Upon examining the General Flow Equation 2.2, it is obvious that for a pipe segment of length *L* and diameter *D*, the gas flow rate *Q* (at standard conditions) depends on several factors. *Q* depends on gas properties represented by the gravity *G* and the compressibility factor *Z* . If the gas gravity is increased (heavier gas), the Flow rate will decrease. Similarly, as the compressibility factor *Z* increases, the flow rate will decrease. Also, as the gas flowing temperature T_a *increases*, throughput will decrease. Thus, the hotter the gas, the lower the flow rate will be. Therefore, to increase the flow rate, it helps to keep the gas temperature low. The impact of pipe length and inside diameter is also clear. As the pipe segment length increases for given pressure P_1 and P_2, the flow rate will decrease. On the other hand, the larger the diameter, the larger the flow rate will be. The term $P_1^2 - P_2^2$represents the driving force that causes the flow rate from the upstream end to the downstream end. As the downstream pressure P_2 is reduced, keeping the upstream pressure P_1 constant, the flow rate will increase. It is obvious that when there is no flow rate, P_1 equal to P_2. It is due to friction between the gas and pipe walls that the pressure drop (P_1–P_2) occurs from the upstream point 1 to downstream point 2. The friction factor *f* depends on the internal condition of the pipe as well as the type of flow (laminar or turbulent). Equation (2.2) was developed with the assumption that the gas pipeline is horizontal, this is usually not completely so. However, as long as the slope is not too great, a correction for the static head of fluid (H_c) may be incorporated into equation (2.2) as follows (Schroeder, 2001).

$$Q = 1.1494 \times 10^{-2} \left(\frac{T_b}{P_b}\right) D^{2.5} \left[\frac{(P_1^2 - P_2^2 - H_c)}{fGT_aLZ_a}\right]^{0.5} \qquad (2.3)$$

Where

$$H_c = \frac{0.0684\ g\ (H_2 - H_1)P_a^2}{Z_aT_a} \qquad (2.4)$$

Where H_1 and H_2 are inlet and outlet elevations (m) and g is gravitational constant m/s^2. The average compressibility factor, Z_a, is determined from the average pressure (Pa) and average temperature (T_a), where P_a is calculated from equation 2.5 (Menon, 2005).

$$P_a = \frac{2}{3}\left[(P_1 + P_2) - \left(\frac{P_1P_2}{P_1 + P_2}\right)\right] \qquad (2.5)$$

The average temperature can be determined from equation 2.6 (Menon, 2005; Schroeder, 2001).

$$T_a = \left[\frac{T_1 - T_2}{ln\left(\frac{T_1 - T_s}{T_2 - T_s}\right)}\right] + T_s \qquad (2.6)$$

T_S is the soil temperature while T_1 and T_2 are the upstream and downstream temperatures respectively.

Equation 2.2 can be further expressed in terms of transmission factor, F as expressed by (Menon, 2005).

$$Q = 5.747 \times 10^{-4} F \left(\frac{T_b}{P_b}\right) \left[\frac{(P_1^2 - P_2^2)}{GT_f LZ}\right]^{0.5} D^{2.5} \qquad (2.7)$$

Where the transmission factor and friction factor are related by (Menon, 2005).

$$F = \frac{2}{\sqrt{f}} \qquad (2.8)$$

Enrique et al. (1983) presented a new method of calculating head loss in gas pipelines which gave more coherent solution of the head loss calculation in gas pipelines and a more rapid calculation with the same precision that is especially beneficial in network solutions.

Osiadacz and Chaczykowski. (2001) compared isothermal and non-isothermal pipeline gas flow models. They concluded from their findings that cooling of the gas improves the efficiency of the overall compression process and that there existed a significant difference in the pressure profile along the pipeline between isothermal and non-isothermal process. This difference increases when the quantity of gas increases. This shows that, in the case when gas temperature does not stabilize, the use of an isothermal model leads to significant errors. The problem of choosing the correct model is a function of network structure and network complexity.

Ouyang and Aziz. (1996) opined that many of the equations of flow presented in the literature have been oversimplified by assumptions and approximations and also by inclusion of inaccurate friction factor correlation which consequently leads to nontrivial errors.

2.3.2 Velocity of Gas in a Pipeline

The velocity of gas in a pipeline is not as straight forward as the velocity of incompressible fluid, this represents the speed at which the gas molecules move from one point to another. The gas velocity can be a steady non-uniform flow because it is a function of pressure and hence will vary along the pipeline as pressure varies. The downstream where the pressure is least, experience highest velocity and least velocity is experienced at the upstream where the pressure is highest. For a gas flow from an upstream point 1 to another point 2 with no injection or delivery of gas between these two points, the mass flow rate for a steady flow can be written as

$$M = Q_1\rho_1 = Q_2\rho_1 = Q_b\rho_b \qquad (2.9)$$

where Q_b is the gas flow rate at standard conditions and ρ_b is the corresponding gas density.

Equation 2.9 can be written as

$$Q_1 = Q_b\left(\frac{\rho_b}{\rho_1}\right) \qquad (2.10)$$

Applying the gas law equation, we can write

$$\rho_1 = \frac{P_1}{Z_1RT_1} \qquad (2.11)$$

where P_1 and T_1 are the pressure and temperature at pipe section 1.
Similarly at standard conditions,

$$\rho_b = \frac{P_b}{Z_bRT_b} \qquad (2.12)$$

Putting equation 2.11 and equation 2.12 into equation 2.10, the flow rate can be written as with Z_b approximately equal to 1.00

$$Q_1 = Q_b\left(\frac{P_b}{T_b}\right)\left(\frac{T_1}{P_1}\right)Z_1 \qquad (2.13)$$

The velocity can be obtained from the relation

$$u_1 = \frac{Q_1}{A} \tag{2.14}$$

Therefore,

$$u_1 = \frac{Q_b Z_1}{A}\left(\frac{P_b}{T_b}\right)\left(\frac{T_1}{P_1}\right) \tag{2.15}$$

Where A is the cross-sectional area and other terms are as defined earlier.

2.3.3 Effect of Elevation

Elevation has effect on the flow of gas in a pipeline although the effect is not the same with flow of incompressible fluids. This is because the density or specific weight of a liquid is greater than that of a gas, the gravitational effect when the flow is ascending is obviously greater with liquid. This effect is far less in gas flow. If the elevation difference between the ends of the pipe segment is taking into consideration, then the general flow equation becomes:

$$Q = 5.747 \times 10^{-4} F\left(\frac{T_b}{P_b}\right)\left[\frac{(P_1^2 - e^s P_2^2)}{G T_f L_e Z}\right]^{0.5} D^{2.5} \tag{2.9}$$

Where L_e is given as equation 2.10

$$L_e = \frac{L(e^s - 1)}{s} \tag{2.10}$$

The equivalent length L_e, and the term e^s take into account the elevation difference between the upstream and downstream ends of the pipe. The parameter s which depends upon the gas gravity, gas compressibility factor, the flow temperature and the elevation difference can be expressed as (Menon, 2005).

$$s = 0.0684 G\left(\frac{H_2 - H_1}{T_f Z}\right) \tag{2.11}$$

Where H_1 = upstream elevation, m and H_2 = downstream elevation, m

2.3.4 Weymouth Equation

For gas flow at high pressure, high flow rate, and through a large diameter pipeline, the Weymouth equation is used to calculates the flow rate through a pipeline for given values of gas gravity, compressibility, inlet and outlet pressures, pipe diameter, and length.

The Weymouth equation is as follows:

$$Q = 3.7435 \times 10^{-3} E \left(\frac{T_B}{P_B}\right) \left[\frac{P_1^2 - e^s P_2^2}{G T_f L_e Z}\right]^{0.5} D^{2.667} \qquad (2.12)$$

E is pipeline efficiency, a decimal value less than or equal to 1.0

A brief summary of the application of the flow equations mentioned in section 2.1.3 is given in Table 2-3.

Table 2-3: Summary of Pressure Drop Equations

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

2.4 Compressors

A compressor is a device used to increase the pressure of a compressible fluid. The inlet pressure level can be as low as a vacuum to a high positive pressure while the discharge pressure can range from sub atmospheric levels to very high values. The inlet and outlet pressures are related depending on the type of compression. This can be by trapping a specific volume in a chamber, reducing the volume of the chamber and thereby increasing the pressure of the gas by the ratio of the initial chamber volume to final volume. It can also be by the conversion of kinetic energy into pressure energy. This is achieved through accelerating the fluid to a higher velocity and then decelerating it by changing its direction of flow.

In gas transmission, two basic types of compressors are used: reciprocating and centrifugal compressors, although the major types of compressors will be reviewed. Reciprocating compressors are usually driven by either electric motors or gas engines, whereas centrifugal compressors use gas turbine or electric motors as drivers.

2.4.1 Types of Compressors

There are different types of compressors, the classification of which can be by method of compression i.e. intermittent or continuous and can also be by the action of the compressor such as reciprocating, rotary, centrifugal and axial.

2.4.1.1 Reciprocating Compressor

A reciprocating compressor operates on the principle of trapping in a cylinder, driving a piston into the cylinder to reduce the volume and thereby increasing the pressure according to equation (2.13).

$$P_2 = P_1 \left(\frac{V_1}{V_2}\right)^{\gamma} \qquad (2.13)$$

V_1= Suction volume

V_2= Discharge volume

The same volume of gas is compressed for every repeated cycle. The capacity of a reciprocating compressor is directly proportional to the speed at which the piston moves through the cylinder. This implies that the capacity of a reciprocating compressor is fixed at constant speed.

Reciprocating compressors are suitable for systems requiring high compression ratios (ratio of discharge to suction pressures) per stage with low rates, and handling relatively dry gas. Compressing a gas however, increases its temperature according to equation (2.14) which consequently limit the amount of compression that can be done per stage so as not to exceed the critical temperature of the gas.

$$T_2 = T_1 \left(\frac{V_1}{V_2}\right)^{\gamma-1} \tag{2.14}$$

Where T_1 and T_2 are the suction and discharge temperatures respectively.

The reciprocating air compressor, illustrated in Figure 2.14, is a common design employed today.

The reciprocating compressor normally consists of the following elements.

a. The compressing element i.e. cylinders, pistons, and valves.
b. Connecting rods, piston rods, crankshaft and flywheel for power transmission.
c. A lubricating system for rubbing parts and also a sump for the lubricating oil, and a pump.
d. A control system for maintaining the pressure in the discharge line.
e. An unloading system, which operates in conjunction with the regulator, to reduce or eliminate the load put on the prime mover when starting the unit.

Reciprocating compressors employed in gas production and gathering range in size from about 100 hp (75kW) to around 6700 hp (5000 kW) with a median size from 1300hp(1000 kW) to about 2600 hp (2000kW). The speed range of this type of compressor is between 200 and 1500 rpm (Geitner, 2009).

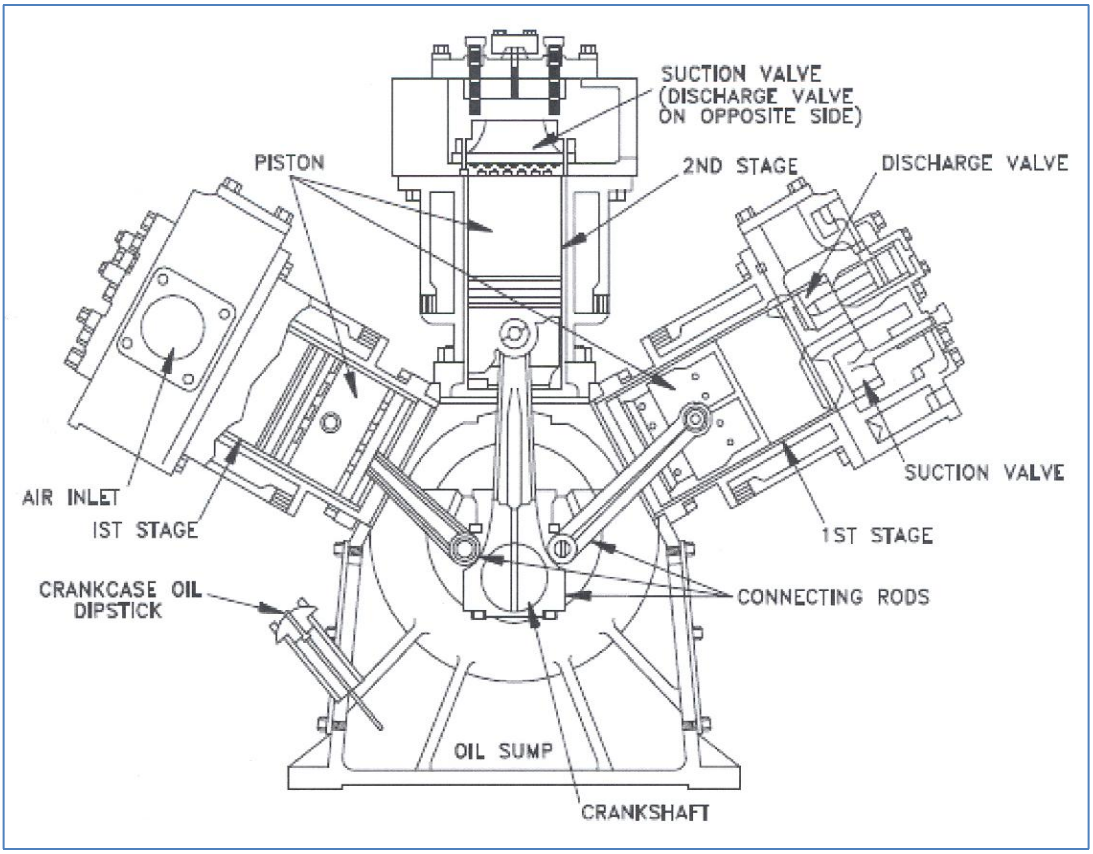

Figure 2-14: Reciprocating Air compressor (Lüdtke, 2004)

2.4.1.2 Rotary Compressor

A rotary compressor operates by confining a volume of gas in a pocket formed by two rotating components. As these components continue their rotation the volume of the pocket is decreased, increasing the pressure of gas, according to equation (2.13). Finally the rotating component line up so as to discharge the gas into a receiver. These types of compressors are the least used in industry of all other compressors, despite many claimed advantages. These advantages include high rangeability in capacity and compression ratio, low rotating speed, and good maintenance performance. Probably, one of the important advantages is in the compression of “dirty” or corrosive gases.

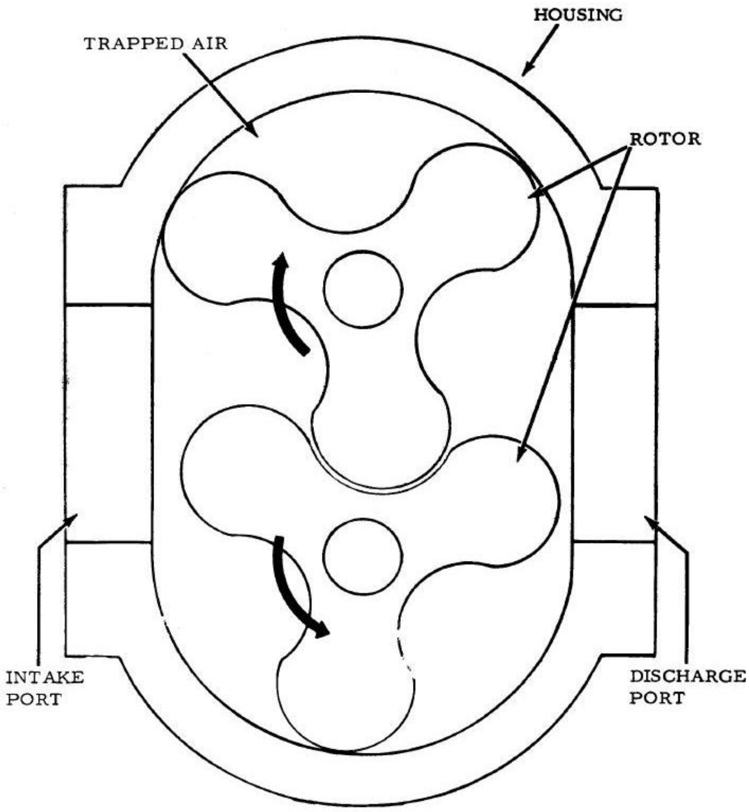

Figure 2-15: Schematic diagram of a rotary compressor

There are five basic types of rotary compressors:

- Helical screw
- Spiral axial
- Straight lobe
- Slide/vane
- Liquid-liner

Each has an almost unique application.

2.4.1.3 Axial Compressor

Axial compressor is so named because the gas flows in the direction along the length of the shaft, just like air flows through a common household fan. Axial compressors are well suited for applications which require high flow rate and high efficiency with low pressure ratios. This compressor type can achieve pressure ratios up to 5.5 while handling up to 350,000 mcf of working fluid with polytrophic efficiencies up to 90% and beyond. Axial compressors are made of series of stages of rotating blade in which fluid is accelerated and stationary blade(vane) which converts this kinetic energy into static pressure. Axial compressors generally have between 3 and 17 stages, depending on the required pressure ratio and rotational speed. Volume capability is controlled by varying the size of the flow path. Off-design operation is achieved through the use of variable geometry (adjustable vanes in the first few stages) or variable speed to meet changing process requirements.

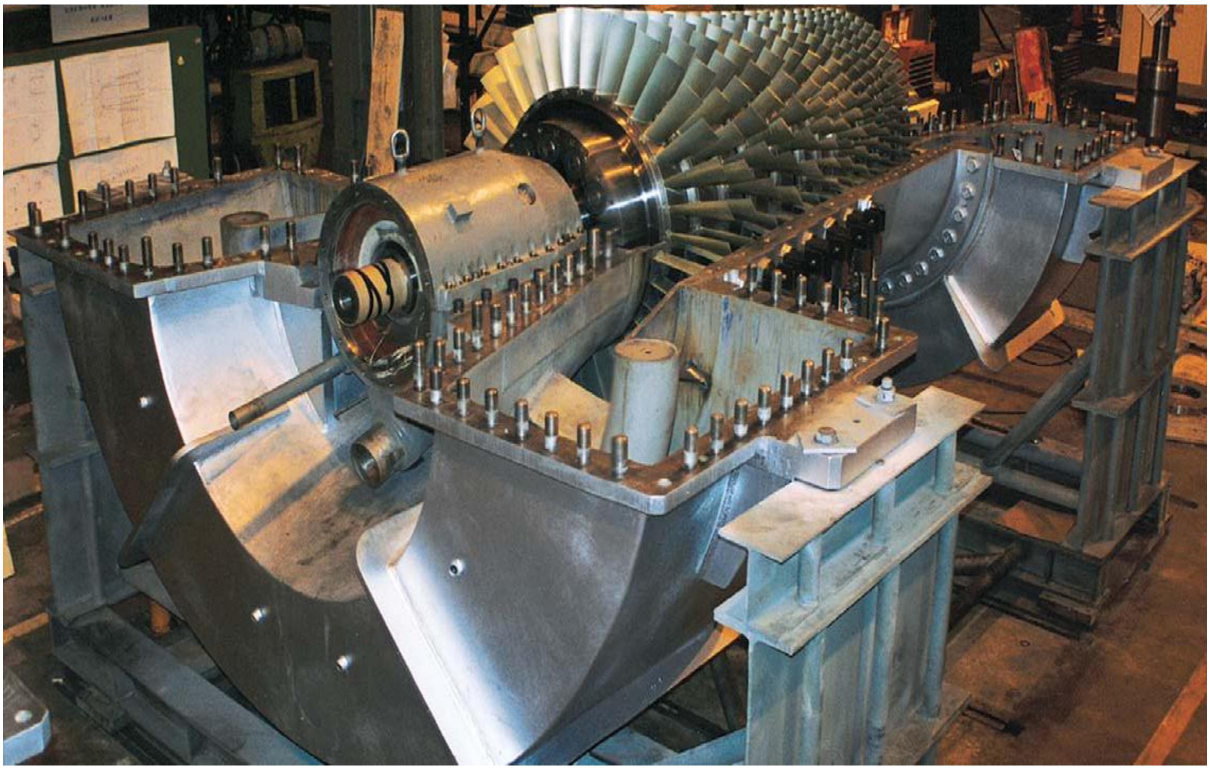

Figure 2-16: Axial compressor (Geitner, 2009)

2.4.1.4 Centrifugal Compressor

Owing to its versatility, reliability and compactness, centrifugal compressor is widely used in the oil and gas industry for gas compression. A centrifugal compressor is a continuous flow machine which increases the pressure of gas by first accelerating the gas and then converting the kinetic energy of the gas to pressure. It has capabilities ranging from below 1MW up to around 50MW and are preferred because of their high efficiency, good economy, low maintenance and high degree of reliability.

Centrifugal compressors are used to raise gas pressure for diverse reasons, viz:

- For the separation of heavy hydrocarbons since this is possible when the hydrocarbon is under pressure. The heavier ones liquefying first.
- To obtain supercool gases for refrigeration. This is owing to the fact that gases under pressure can have their temperature lowered by expansion.
- To cause positive flow through a process by increasing the pressure to overcome process pressure drops due to piping, vessels, heat exchangers, valves, and fittings.
- Once under pressure, the gas temperature can be lowered by expansion of the gas.
- To transport gas product through pipelines (Lapina, 1982).

2.4.1.5 Centrifugal Compressor Operation

The operation of a centrifugal compressor is defined by characteristic curves. The limits of the curve defines the operating window.Operating a compressor within this window is stable. Operating point beyond the window either results in an aerodynamic instability or may result in risk to the mechanical condition of the compressor.

The characteristic curve is unique for a particular speed of the compressor. There are different curves for different speeds of the compressor as shown in Figure 2-18. The top speed is limited by process requirement or the mechanical design of impellers. The bottom speed limit is governed by a phenonmeno n called a critical speed of rotor or the process requirement.

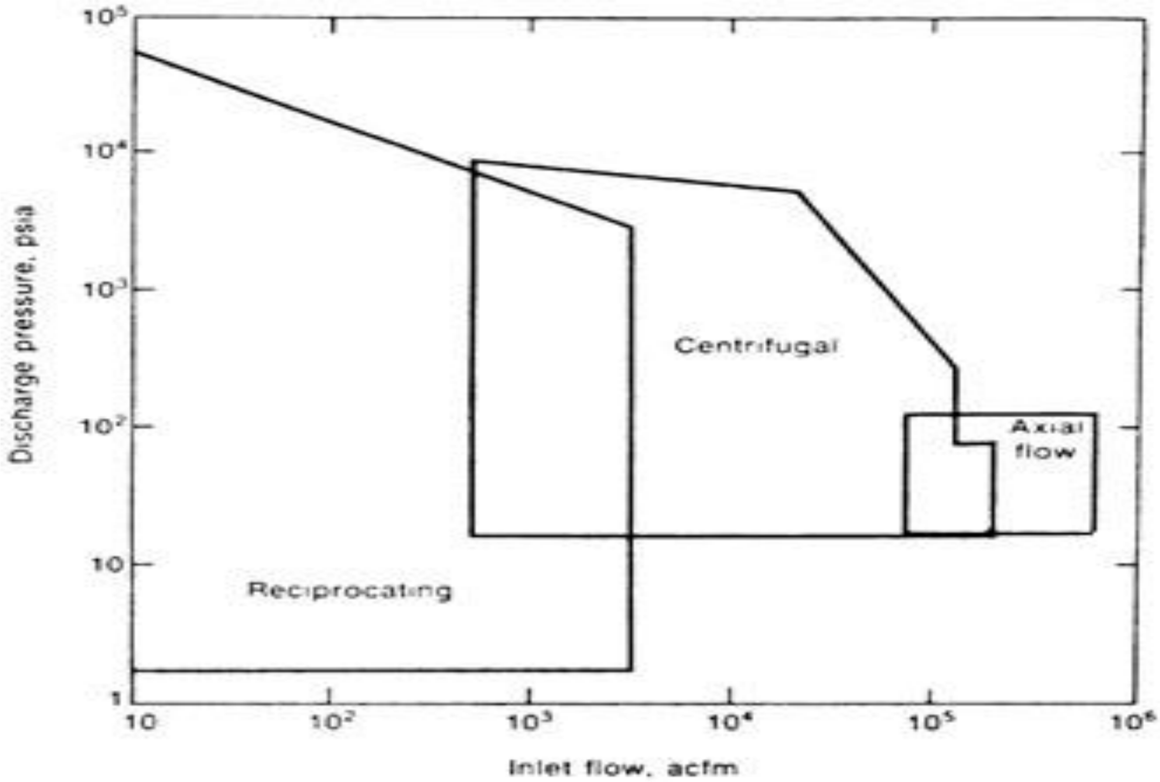

Figure 2-17: Approximate ranges of application for reciprocating, centrifugal, and axial-flow compressors (Dimoplon, 1978)

2.5 Centrifugal Compressor Performance

The present rise in gas prices and increase in the commodity demand has put operating companies under pressure to ensure that their production systems operate at their maximum capacity. The performance and availability of centrifugal compressors is vital to the operation of these facilities. Over the past 20 years, the availability of these machines has been the focus of attention and performance issues have taken a back stage. Even API 617 (1) does not require the vendor to guarantee the compressor efficiency, only the absorbed power with a +/- 4% tolerance (Akhtar, 2006).

The compression that takes place in a reciprocating compressor is considered closer to isentropic or adiabatic process. How ever in case of centrifugal compressors, the rules adopted by the industry include:

For single stage and for compressors handling air, the compression is evaluated considering adiabatic process. Thus only multistage compressors handling gases other than air are evaluated considering polytropic compression.

The parameters that generally describe the performance of a centrifugal compressor are

- Inlet flow/flow coefficient
- Head/head coefficient
- Efficiency and
- Power absorbed

These parameters are affected by other parameters of flow such as gas composition, gas properties and inlet condition into the compressor. In gas transportation through pipelines, it is not uncommon to observe changes in gas composition and or quality owing to flow of gas from different sources. Dos Santos and Lubomirsky. (2006) presented a methodology of evaluation and a sensitivity study using different gas qualities to show the impact on compression overall performance. The proposed methodology assumes that the gas volumes to be delivered to the market are energy based. Different gas compositions have different LHV (Low Heat Value), consequently, gas volumes may differ significantly for the same amount of energy and therefore affect compressor performance, temperature and pressure drop across the pipeline. This study (Dos Santos and Lubomirsky, 2006) assumed a hypothetical pipeline between two compressor stations and designed to operate with three different gas compositions. They concluded that gas composition affects compressor units overall efficiency and finally in their opinion care must be taken in addressing this aspect so as to prevent biased decisions related to equipment selection. One of the parameters to plot as the independent variable on a compressor characteristic is the inlet or actual volume flow. The parameters to be plotted against this include polytrophic head, discharge pressure, power requirement, pressure ratio, pressure rise and discharge temperature. Figure 2-18 shows centrifugal compressor characteristic curves.

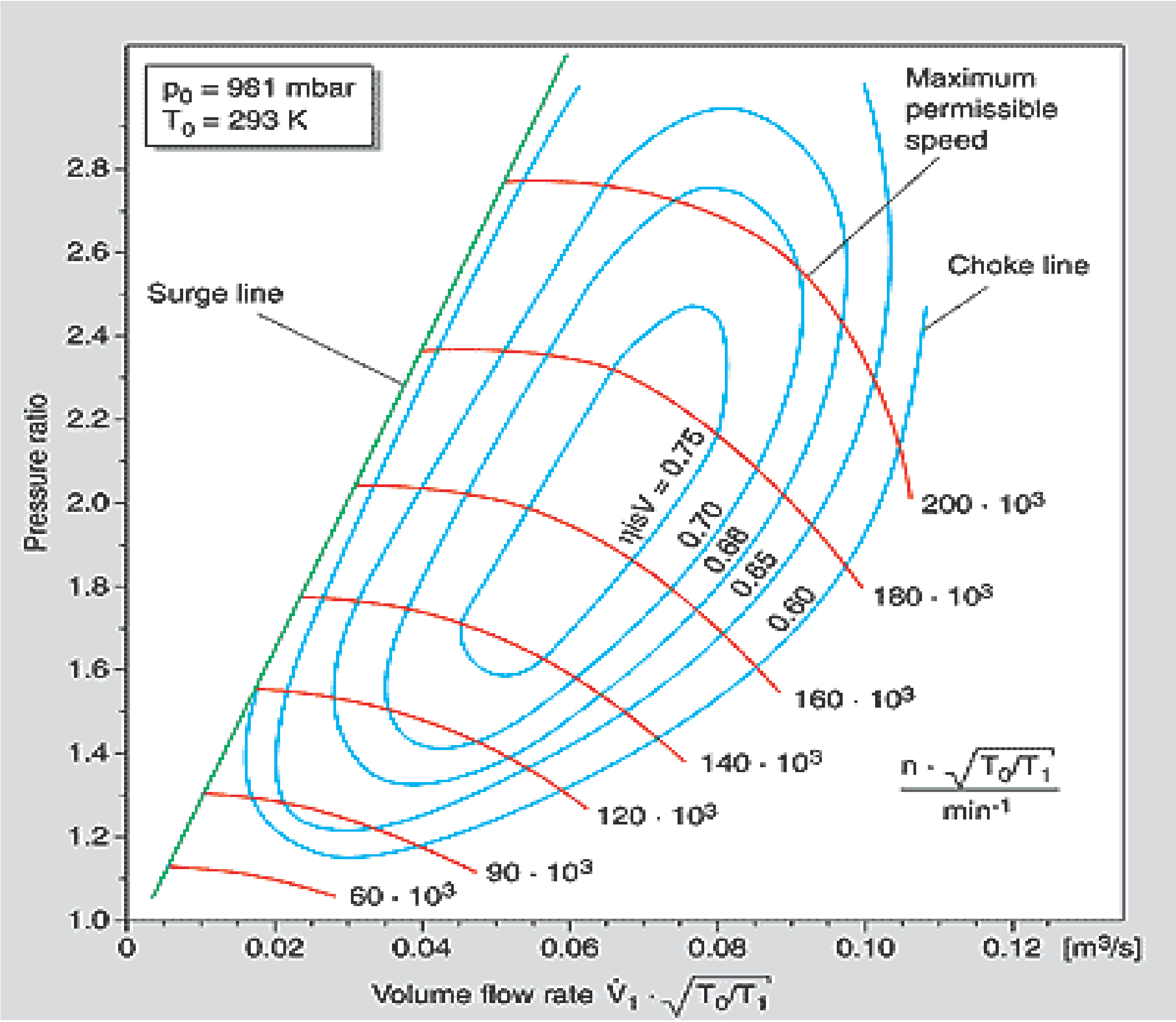

Figure 2-18: The centrifugal characteristic curves

The inlet conditions that can affect the performance of a centrifugal compressor include variation in inlet pressure, temperature and gas properties such as compressibility and molecular weight at inlet.

In ascertaining the thorough performance of a compressor at off-design conditions, it is imperative to field test the compressor under varying conditions. Kruz and Brun. (1998) stressed the importance of a correct and thorough preparation of such tests and presented the concept of test uncertainty. They suggested ways to optimize the performance test which include a cognisant agreement between the responsible parties prior to the test on how to conduct and evaluate the test. In case the results of the test vary significantly from the predicted result. Kruz and Brun, (1998) suggested that an analysis to identify the sources of errors should be performed.

The performance of centrifugal compressor can be displayed in a map showing polytrophic efficiency and polytrophic head as a function of actual inlet flow, with the compressor speed as a parameter (Kurz and Ohanian, 2003a):

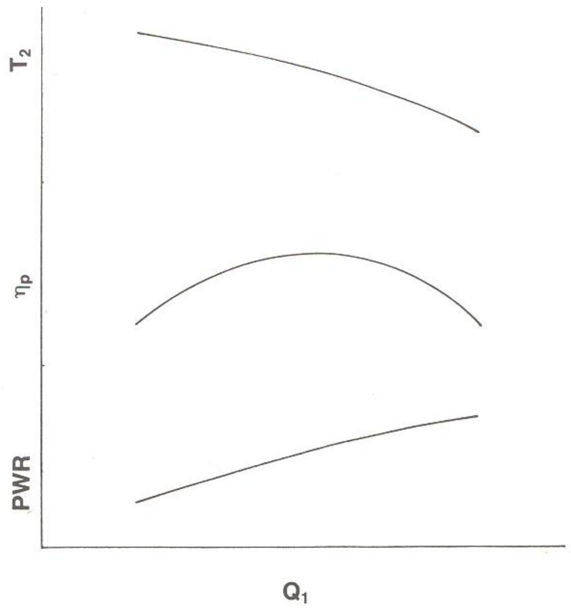

Figure 2-19: Variation of several parameters with inlet flow

While fuel prices were low, the performance of these machines in oil and gas sector at least, has never been a central issue. Operators were content to see that the machines kept running and did not fail. The general perception remained that the loss of compressor performance could be compensated for by an increase in engine power or higher fuel consumption. Since fuel was available in abundance and was low in cost, the economic case for performance improvement of operating machines was weak until now. With the rise in fuel prices, fuel consumption could no longer be ignored and became an issue. The operators also discovered that the production capacity was no longer constant and that this was related to compressor efficiency.

2.5.1 Compressor Operating Limits

Many centrifugal compressors have one or more of the following operating limits:

- ***Minimum Operating Speed*** (MOS) is the minimum speed for acceptable operation. Below this value the compressor may be controlled to stop, or to Idle.
- ***Maximum Allowable Speed*** (MAS) is the maximum operating speed for the compressor. Speeds above this make the compressor unstable because it affects component vibration. Compressors are usually controlled to a lower speed as soon as the speed goes above the maximum allowable speed.
- ***Stonewall or Choke*** defines a usually not detrimental phenomenon which occurs as flow increases, with consequent gas velocity approaching the gas/fluid's sonic speed somewhere within the compressor stage. For low speed equipment, as flows increase, losses increase such that the pressure ratio drops to 1:1.
- ***Surge*** is an undesirable phenomenon which occurs at the point at which the compressor cannot add enough energy to overcome the system resistance. This causes a rapid flow reversal (i.e. surge) and as a result, high vibration, temperature increases, and rapid changes in axial thrust can occur. Some compression systems are designed with anti-surge so as to withstand occasional surging.

  At the instance that the compressor starts, the discharge pressure and the inlet pressure will be equal owing to prior equilibrium. Consequently no head is produced by the compressor (point 1 in Figure 2.20). As the mass of gas in the receiver starts to increases, the pressure in the receiver will start to rise, thereby given some resistance. This causes a slight drop in the flow (point 2 in Figure 2.20), called the stonewall point. As the mass of gas continues to increase, the pressure also increases, providing greater pressure differential from inlet to discharge of the compressor, and at the same time a greater resistance to flow (point 3), a typical operating point. As the mass of gas increases further, a pressure is eventually reached, above which the compressor cannot compress stably(Point 4). This is the surge point, a point of minimum stable flow and highest head. At a flow below this point the compressor becomes aerodynamically unstable. When the velocity becomes too slow, the compressor can no longer[perform stably] and this can result in flow reversal.

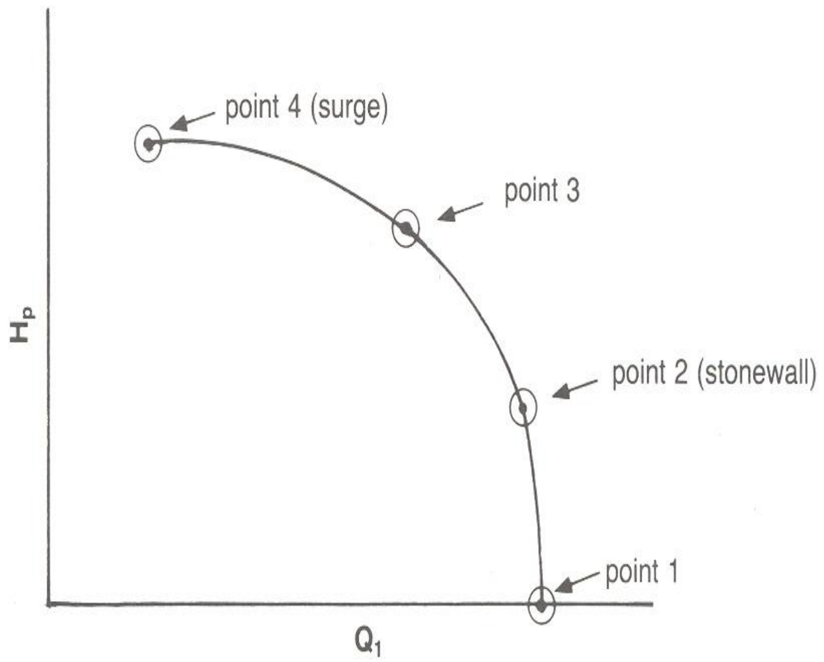

Figure 2-20: The centrifugal compressor characteristic curve

Surge is energetic and can cause damage to thrust bearings, seals, impellers, etc. Because of the damaging effect of surge, surge controls are usually put in place to check this undesirable phenomenon. The stonewall point is the maximum stable compressor flow point. Consequently, it is the minimum head point under stable compressor operation.

Some pipelines are equipped with anti-surge valve arrangements, which are meant to open when the flow is approaching surge point.

2.5.2 Compressor Power

The compressor power (gas power) is the power required by the compressor to compress a certain mass of gas through a specified pressure ratio. In order to correctly establish the gas power, the head in joule developed by a compressor is first calculated. The general form of the thermodynamic head equation for a polytrophic process is

$$H_P = ZRT_1 \frac{n}{n-1}\left(r_P^{\frac{n-1}{n}} - 1\right) \qquad (2.15)$$

The polytrophic head, H_P is proportional to the square of the mechanical tip speed of the impeller (μ).

$$H_P \propto \mu^2 \qquad 2.16$$

$$H_P = \frac{\mu^2 C}{1000} \qquad (2.17)$$

The head coefficient, C, is a variable for any particular impeller. It increases as flow through the impeller decreases and vice versa (Lüdtke, 2004). Polytrophic process is preferred over the adiabatic process because the polytrophic efficiency is independent of the state of the gas as shown in equation (2.18)

$$\eta_p = \frac{\frac{n}{n-1}}{\frac{\gamma}{\gamma-1}} \qquad (2.18)$$

The gas power is thus given by equation (2.19) for a mass m flowing through the machine.

$$PWR = \frac{mH_P}{3600\eta_p} \tag{2.19}$$

The shaft power can be expressed as

Shaft power = Gas power + Mechanical Losses +Leakage Losses+ Coupling losses

2.5.3 Selection of a Centrifugal Compressor

The requirements of a compressor is usually spelt by process demand. The specification lays the expcted flow rate, pressure rise and gas properties. These forms the basis of the type and size of the equipment. Centrifugal compressos have a typical flow rate range from 500m3/hr to 300,000m3/hr and can be multi-stage to give pressure ratio of over 20 (Girdhar, 2008).

Selecting a a compressor to achieve low life cycle costs is very important in compressor selection because the capital and operating costs of centrifugal compressors is relatively high. It is thus essential to specify the entire range of expected performance, modes of operation, off-design duties and possible future reratings. This not only helps in the selection of the compressor and its components but also its control and protection systems.

Though it is technically possible to design compressor components with maximum efficiency for the rated point, the compressor operation is rarely a single point operation over its life cycle. In an effort to optimize the engineering and manufacturing costs, the compressor manufacturers develop a family of frames that cover the range of possible flow rates.

Table 2-4: A typical centrifugal compressor frame data (Girdhar, 2008)

Frame	Nominal Flow Range – m^3/hr	Maximum Number of stages	Nominal Speed RPM	Nominal Polytropic Efficiency	Nominal H/N^2 per stage	Maximum Q/N (m^3/rev)
A	425 – 4250	9	15000	0.75	1.13E-05	0.0042
B	1360 – 15300	9	11500	0.78	2.29E-05	0.0184
C	8500 – 42500	8	8000	0.80	4.57E-05	0.0821
D	25500 – 59500	8	600	0.82	7.62E-05	0.1557
E	51000 – 119000	8	5000	0.83	1.22E-04	0.3398
F	93500 – 212500	8	3000	0.84	3.05E-04	1.4158
G	170000 - 289000	8	2700	0.84	3.66E-04	1.8406

2.6 Compressor Station

The continuous flow of natural gas in a pipeline is facilitated by the help of compressor stations which boost the pressure at predetermined intervals along a pipeline. These stations are generally made up of basic components such as compressor and driver units, scrubber/filters, cooling facilities, emergency shutdown systems, and an on-site computerized flow control-Supervisory Control and Data Acquisition (SCADA) and dispatch systems that maintains the operational integrity of the station (Carter, 1998).

The compressor stations add energy to the gas to overcome frictional losses and maintain the required delivery pressure and flow. Compressor station design has been essential over the years because it is very important in the successful implementation of natural gas pipeline transportation (Mokhatab et al., 2007). The pressure difference between the discharge side of one station and the suction into the city-gate of another station is responsible for the gas flow in the pipelines. An average station may deliver up to 830 mcf of gas per day. Compressor station engines run almost continuously for several months or years (Mokhatab et al., 2007).

Natural gas is received through the suction header in a compressor station and passes through the scrubbers, for the removal of any solids and most liquids from the gas. Owing to its high temperature due to compression, high-pressure gas coolers lower the temperature of the gas before it is discharged into the main pipeline. Cooling reduces the work required for a certain compression and also allows for the transportation of greater volumes, as gas is denser at lower temperatures. Natural gas finally leaves the cooling system into the main pipeline for onward transportation. Fig. 2-22 shows the schematic of a Compressor Station.

Montoya-O et al. (2000) describes the application of a modified Genetic Algorithm (GA) to optimize the design of gas transmission networks operating under steady-stable conditions. Their proposed model integrates an optimization module (Genetic Algorithm) and a module of analysis for gas transmission networks (Hardy-Cross). The G.A. determines the diameters of the pipes and the operation conditions, i.e., pressure and flow, in the network for a minimum investment cost.

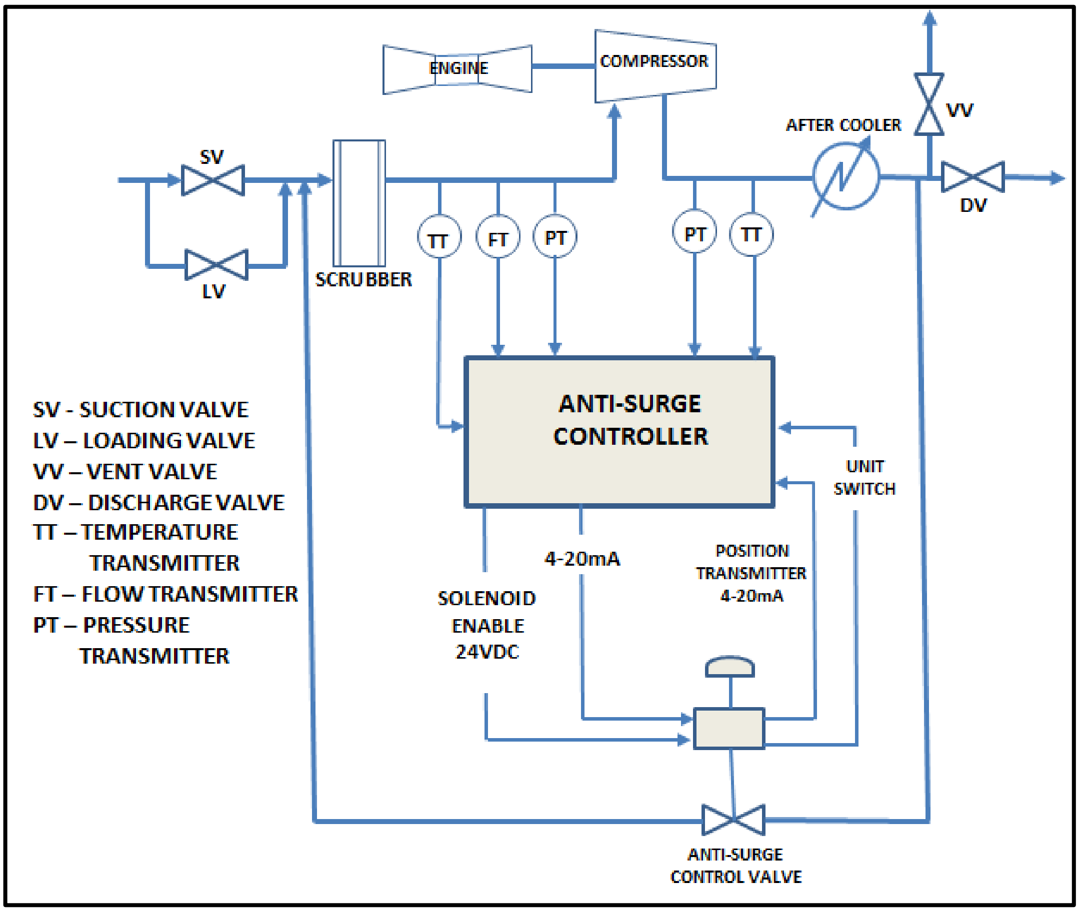

Figure 2-21: The Schematic Diagram of a Compressor Station (Mokhatab et al., 2007)

2.7 Gas Turbine

The present research looks only at gas turbine producing shaft power for driving a pipeline compressor. Among the various means of producing mechanical power, gas turbine is reliable because lack of reciprocating and rubbing members resulting in few balancing problem and also the lubricating oil consumption is exceptionally low (Cohen et al., 2009). In considering the prime mover of choice in the oil and gas industry, there exist different configurations and cycles. A brief review of some of the cycles is presented to serve as a precursor to the main text presenting gas turbine as prime mover for pipeline compressors. The basic components that makes up a gas turbine are the compressor to compress air from ambient pressure to a predetermined pressure, combustor to burn fuel thereby increasing greatly the temperature of the combustion product, turbine to extract power from the hot gases and a nozzle to convert the extracted power to thrust as in aero or power turbine to produce shaft power as in industrial or aero derivative for a single spool GT.

Gas turbine's advantage of producing large amounts of useful power for a relatively small size and weight, having no reciprocating parts – making its mechanical life long and the corresponding maintenance cost relatively low, versatility in the use of fuel, and having its working fluid as atmospheric air make it useful in the aircraft industry, marine, power generation and mechanical drive for pipeline compressor and other industrial equipment.

In the early stage of development of gas turbine, the major disadvantage was its low efficiency when compared to other internal combustion engines and to steam turbine plants. However, owing to the continuous development in GT technology, thermal efficiency has improved from 18% to the present level of above 40% for simple cycle and 55% for combined cycle. With the progress in gas turbine technology development, more fuel-efficient gas turbines with simple cycle efficiencies predicted as high as 45-47% and combined cycle machines in the 60% range are expected (Langston and Opdyke, 1997).

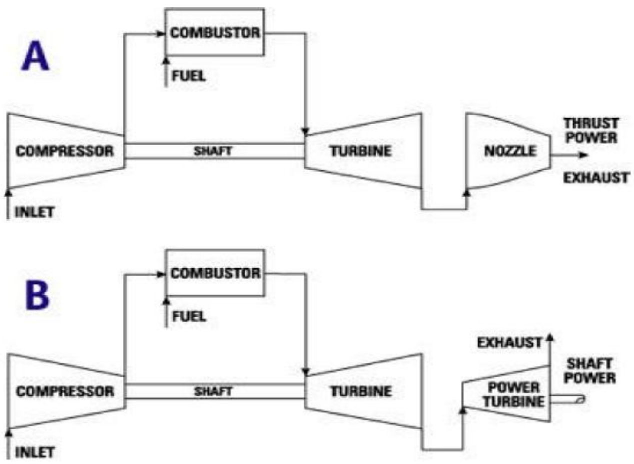

Figure 2-22 Schematic for a) an aircraft jet engine; and b) a land-based gas turbine

3 RESEARCH METHODOLOGY

The main focus of the present research is a techno-economic investigation into viability of the use of gas turbine and electric motors as prime movers for natural gas pipeline compressors. The procedure follows the development of a techno-economic and environmental risk analysis (TERA) architecture which can be employed to assess quickly the economic and technical profitability of a pipeline system. Figure 3-1 show the architecture for TERA for pipeline. The highlight of the two main prime movers used in the compression of natural gas and the development of their modules are presented in this chapter. TURBOMATCH was used to develop the gas turbine performance module and the electric motor module was written in FORTRAN code. Gas turbine emissions, which constitute environmental and human hazard, are presented and the estimated quantity of emissions are evaluated. The heart of the TERA module which is the economic module was also developed by the author in FORTRAN code and receives input from all the other modules. A FORTRAN code called wrapper is used to integrate all the modules to run at the right time. MATLAB genetic algorithm tools was used to optimize the selection of gas turbine based on total cost as objective function. The optimum compressor station location was also carried out using MATLAB GA tools.

3.1 The Concept of TERA

TERA literally stands for **T**echnoeconomic **E**nvironmental **R**isk **A**nalysis. This is a concept developed at Cranfield University as a decision-making framework for the investigation and application of long-term and short-term strategies to improve the performance of gas turbine engines; quantify technical or financial risk; and compare and rank competing schemes on a formal and consistent basis in various applications (Pascovici et al., 2007). Ogaji et al. (2009) outlined a description of TERA for advanced power systems in which it was stated that the concept can consist of three layers of model. The first consist of the suite of performance models of power plants; the second incorporates emissions, environment and economic models; and the third contains two aspects of risk analysis, viz: technological and economic. They concluded that TERA offers increased confidence that research investments and power plant selection decisions are made in a systematic and consistent manner, not unmindful of local conditions. This success developed Cranfield university in-house concept has been applied to various areas of gas turbine applications ranging from marine, civil aviation, to power generation. Ogaji et al. (2009) in their work presented the objective of TERA as a means of ranking and selecting the best schemes, identifying risk so that investment resources can be allocated efficiently.

Pascovici et al. (2007), used this very effective concept of TERA to estimate the cost of maintenance; the direct operating costs; and net present cost of future types of turbofan engines of equivalent thrust bearing. They developed an economic model which is made up of three modules viz:

(1) the lifing module; responsible for estimating the life of the high pressure turbine disk through the analysis of creep and fatigue over a full working cycle of the engine; (2) the economic module; responsible for the estimation of net present cost (NPC) from the knowledge of the direct operating cost (DOC), which is derived as a function of maintenance cost with the cost of taxes on emissions and noise, the cost of fuel, the cost of issurance and the cost of interest paid on the total investment;

(3) the risk module, which uses the Monte Carlo method with a Gaussian distribution to study the impact of variations in some parameters (such as fuel price, interest percentage on total investment, inflation, downtime, maintenance labour cost per man hour and factors used in the emission and noise taxes) on the NPC. For the noise tax, in their work, the minimum and maximum value for the Gaussian of the threshold are 70 and 80 dB and the cost of noise put at 100€/dB and cost of carbon tax at 5 €/kg. They concluded from their findings that for any type of novel cycle engine to be considered economically advantageous, current technology has to be improved and costs of production

have to decrease by around 35%, because they are the main driving factors for the cost of operation of engines.

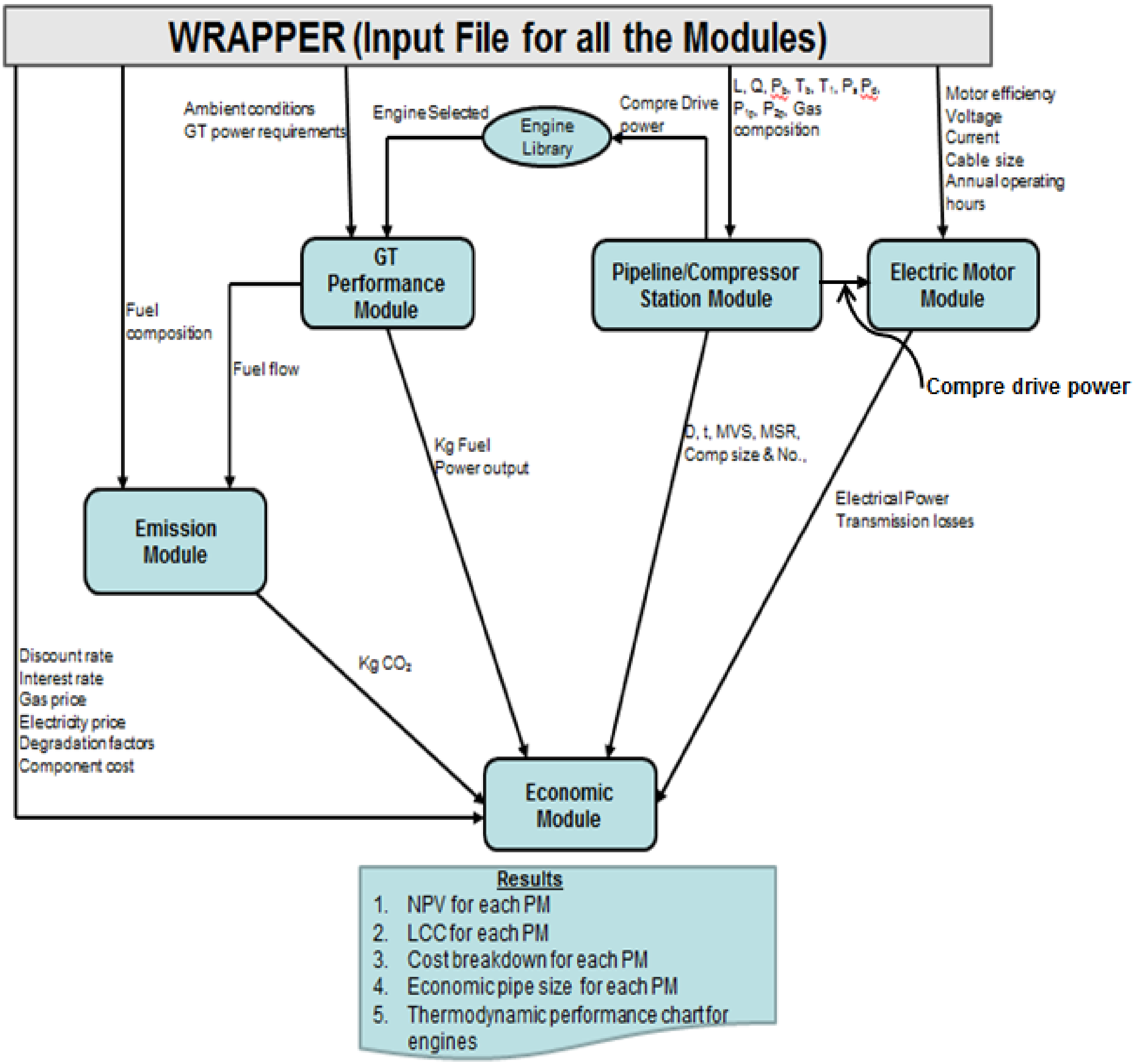

Figure 3-1: TERA for Pipeline Methodology

3.2 PRIME MOVER (PM) OPTIONS FOR PIPELINE COMPRESSORS

Pipeline compressors require a prime mover to perform their function effectively. Prime movers for compressors must supply torque of a specified value, at a certain speed or over a range of speeds. Whenever the PM speed characteristics are not directly usable, they may be modified by a speed-increasing or reducing gear. The only exceptions to the speed-torque criteria are a limited number of reciprocating compressors. These are the integral engines and the direct steam cylinder driven machines. The prime mover's energy source can be either electrical or mechanical. Electrical energy is used by motors, of either induction or synchronous type, while mechanical energy covers a wide range of sources. It may be a fuel, as in internal or external combustion engines, or it may be a gas, such as steam or process gas used in a turbine or expander.

Gas turbine and electric motor are the two main contenders in the prime movers types employed for pipeline compression today. One important consideration in the selection of prime mover is the type of energy available to power the installation. This consideration is closely associated with the economics of the project. For a case requiring the building of long distance of electric line to the site, a rigorous economic analysis will have to be performed to take a decision to use electric motor or not. Another important factor is the availability and reliability of the energy source. Electric motor will not be a viable option for a system characterized by electrical outages. Capital and operating cost plays a very important role in the selection of a prime mover option. In comparing the economics of the prime mover options, annualized and unit cost of the necessary cost components are used. Other parameters that can influence the choice of prime mover include the recommended operating speed of the compressor: the prime mover speed must be compatible with the speed of the driven machine.

The present research will look only into the two commonly used prime movers for pipeline gas compression; these are gas turbine and electric motor.

3.3 Industrial Gas Turbine Performance

Gas turbines are versatile machines used to drive gas compressors in a number of applications, such as in pipelines, on offshore applications and in storage applications. In natural gas pipeline transportation, the preferred means of compressing the gas is by the use of a combination of gas turbine and centrifugal compressor (Kurz and Ohanian, 2003b), and the configuration of gas turbine in this application is the two-shaft gas turbine (Kruz and Brun, 1998). Interest in the knowledge of the field performance of gas turbine in pipeline application is currently increasing because of the need to ascertain return on investment and economic viability of a pipeline project.

Performance parameters such as the efficiency, power, fuel flow, capacity, and head of an installation are of utmost importance as these are the parameters which affect directly or indirectly the techno-economics of a pipeline project.

The gas turbine used for pipeline compressor consists of an air compressor, a combustor, a gas generator turbine and a power turbine. High pressure air is generated by the compressor and passed into the combustion chamber in which fuel is burned. The high pressure, high temperature combustion products expand in the gas generator turbine, the main task of which is to provide power to turn the pipeline compressor. The hot gases leaving the gas generator go into the power turbine for further expansion. The output shaft power as a result of this expansion is responsible for turning the pipeline compressor which is directly connected to the power turbine. The power turbine and the pipeline compressor turn at the same speed, independent of the gas generator turbine.

The gas generator which is controlled by the amount of fuel that is supplied to the combustor has two distinct operating constraints which are the firing temperature (T.E.T.) and the maximum speed. The gas generator speed and the firing temperature increases with increased fuel flow, until one of these two operating constraints reach its limit. This consequently provides the power turbine with gas at a higher pressure, temperature and mass flow, thus allowing the power turbine to produce more power. If the power supplied by the power turbine is greater than the power absorbed by the load, the power turbine together with the driven compressor will accelerate until equilibrium is reached.

An important performance parameter such as the available power at the power turbine output shaft is influenced by a number of factors; these include ambient temperature and pressure, power turbine speed, fuel, inlet and exhaust pressure losses, accessory load and relative humidity. Other gas turbine performance parameters influenced by some of these aforementioned factor are the heat rate and efficiency of the engine. Because gas turbine is an air-breathing engine, the effect of ambient condition on its performance cannot be overlooked. Anything that will affect the density or mass flow of air will definitely affect in no small measure the performance of gas turbine parameters such as power out, efficiency and heat rate.

Figure 3-2 shows the average ambient temperature profile of the region of the natural gas pipeline where the gas turbine is to operate, the maximum ambient temperature being 45 ^{o}C. The effect of these variations was investigated on a Simple cycle Two Shaft (SCTS) engine. A simple-cycle, two-shaft gas turbine which delivers over 40 MW shaft power and at efficiency higher than 40%. The simulation of this engine is presented in the section that discusses design and off-design performance of gas turbine.

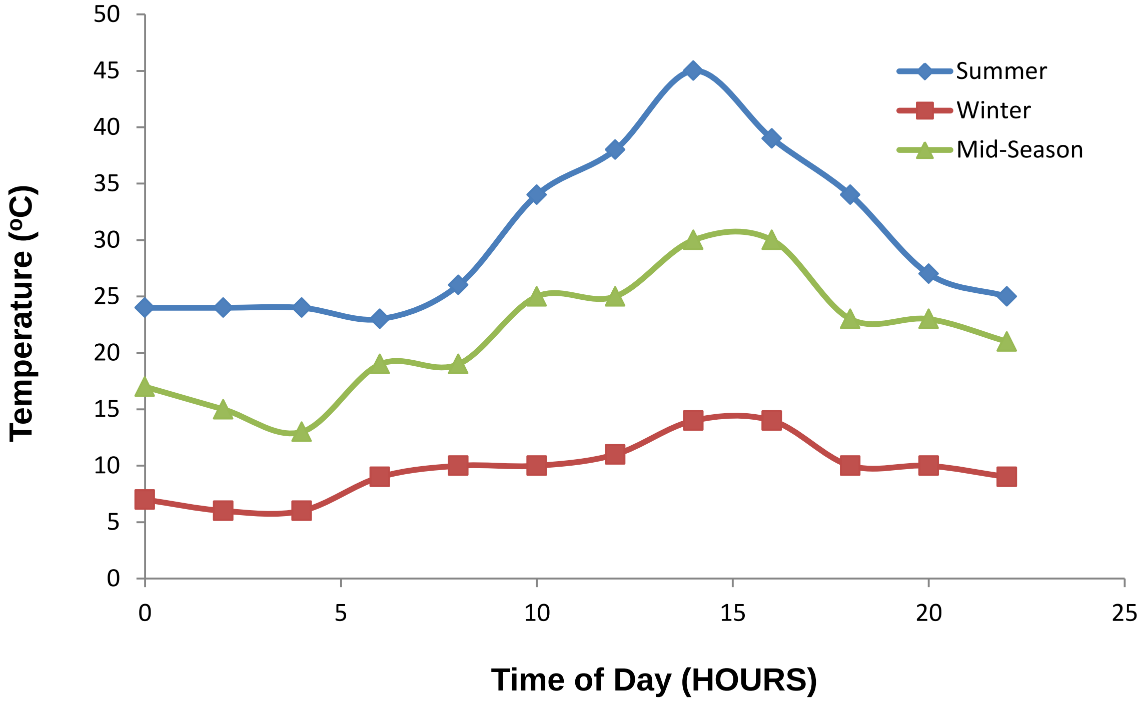

Figure 3-2: Average ambient temperature profile

Ambient pressure has an important effect on the gas turbine output but not on efficiency; this is mostly due to site elevation or simply changes in atmospheric conditions. At low pressure which is a result of high altitude, a reduction in density and consequently a reduction in mass flow ensue, with volume flow rate remaining constant if inlet temperature and firing temperature do not change.. The site considered in this research has its highest elevation of about 177 m. This elevation, which is the maximum in the pipeline profile, suggests that the ambient pressure effect, a function mostly of altitude, will have no noticeable effect on the performance of the gas turbine used for the compression stations. For any operating condition of the gas generator, there is usually an optimum power turbine speed at which the power turbine operates at its highest power and efficiency. The power output and the efficiency of the power turbine will be reduced if the power turbine does not operate at its optimum speed.

3.3.1 Component Characteristic

The full understanding of the design point operation of gas turbine is inadequate as the engine may be required to produce less than its maximum shaft power output in order to meet a demand, and to maintain adequate life for the components or may be operated outside its design conditions. In order to be able to predict the off-design performance of an engine, it is necessary to understand the way the various components behaved. The variation of mass flow, pressure ratio and efficiency with rotational speed of the compressor and turbine is obtained from the compressor and turbine characteristics which are usually produced by the manufacturers.

3.3.2 Design and Off-design Performance Module

In carrying out gas turbine design point simulations, a pressure ratio, component efficiencies and maximum cycle temperature are selected to achieve a required engine performance. The design point simulation determines the thermal efficiency and airflow rate for a given power demand. The modelling and performance simulation of gas turbine engines of simple cycles, but of different configuration and output power, were carried out using TURBOMATCH. Model results of gas turbines of 40.7MW Simple Cycle Two Shaft (SCTS) model and a 33.6 MW Single Spool Simple Cycle model (SSSC) are presented, other performance results are in the appendix.

The design point simulation was done based on the certain parameters which are estimated in order to obtain the desired power output. And the off-design simulation was done with the prevailing ambient temperature profile of the region along which the pipeline compressor stations and consequently the gas turbines are located. The effect of elevation on gas turbine performance is not a major concern in this research because the highest elevation point along the pipeline is 177m. Change in ambient pressure is important in the performance analysis of a gas turbine because this affects the pressure ratio across the power turbine. One very important parameter from the simulation, which obviously affects the economics of the pipeline project is the fuel consumption.

The basic performance parameters of the gas turbines are presented in Table 3-1.

Table 3-1: Performance parameters

	GT Model		
Design Parameter	**40.7 MW SCTS**	**33.6 MW SSSC**	**LM2500+**
Mass Flow (kg/s)	126.6	90.0	69.0
Overall pressure ratio	30.01	25.0	18.8
TET (K)	1540	1575	1505
Thermal Efficiency (%)	40.04	39.60	37.9

3.3.3 40.7 MW Simple Cycle Two-Shaft Gas Turbine

This gas turbine was modelled as a simple cycle engine with the configuration of having a two spool with the LP turbine aerodynamically connected to the power turbine. The model is conceived to have a LP compressor with pressure ratio of 2.45:1 and driven by a LP turbine, HP compressor with pressure ratio of 12.25:1 and driven by a HP turbine. Air leaving the LP compressor is directed into the HP compressor with zero pressure loss and this gives the gas turbine an overall pressure ratio of 30.01. The high and low pressure turbines drive the high and low pressure compressor through concentric drive shafts which rotate independently.

The off-design operating range considered for the simulation is ambient temperature ranging from 10^oC to 50^oC. The effects of varying ambient temperature on some performance parameters are presented in Figure 3-3 to Figure 3-6. For the worst scenario of ambient condition, the gas turbine output power is sufficient for the power demand to compress the natural gas in the modelled natural gas pipeline system. The TURBOMATCH simulation results in a gas turbine with thermal efficiency of 40.04%, with an overall pressure ratio of 30.01:1. The fuel flow of the gas turbine at design point is 2.3587 kg/s. The effect of the ambient temperature and turbine entry temperature (TET) on the power output is shown in Figure 3-3. The output power increases with TET and reduces with increase ambient temperature. From a materials point of view, the TET cannot be increased ad infinitum so as to avoid the early failure of major components and consequently reduced life of the gas turbine. Figure 3-4 shows increase in thermal efficiency with TET at varying ambient temperature.

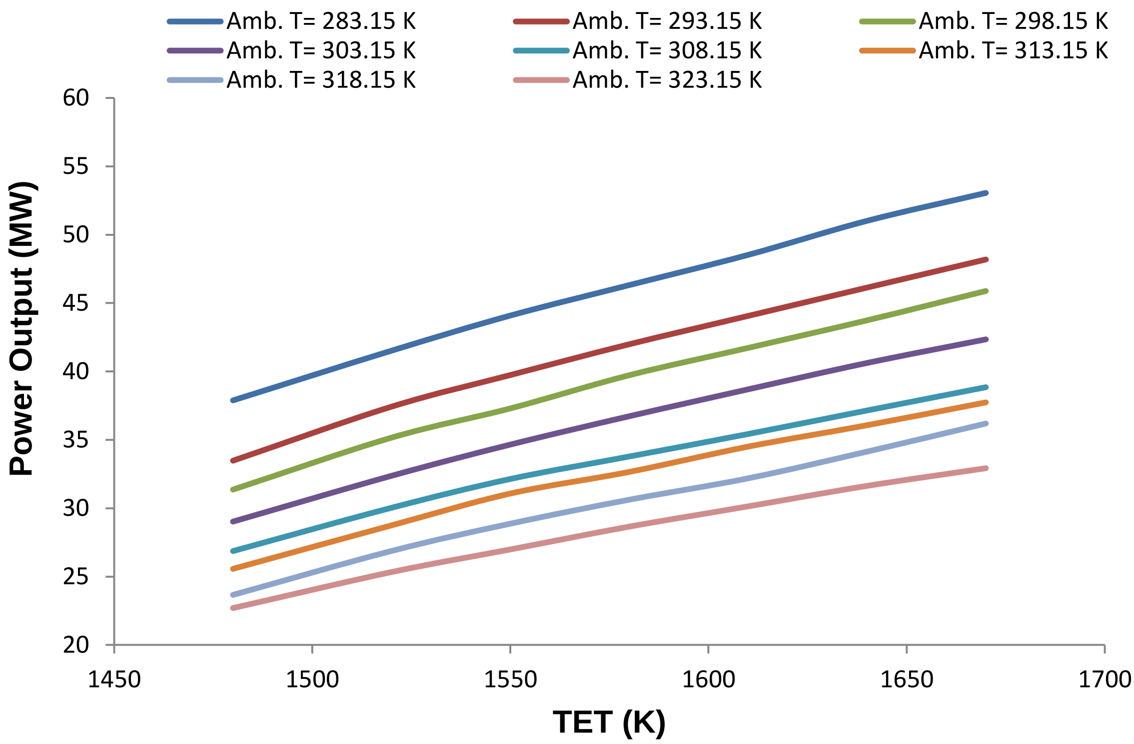

Figure 3-3: Power output against TET for 40.7 MW SCTS

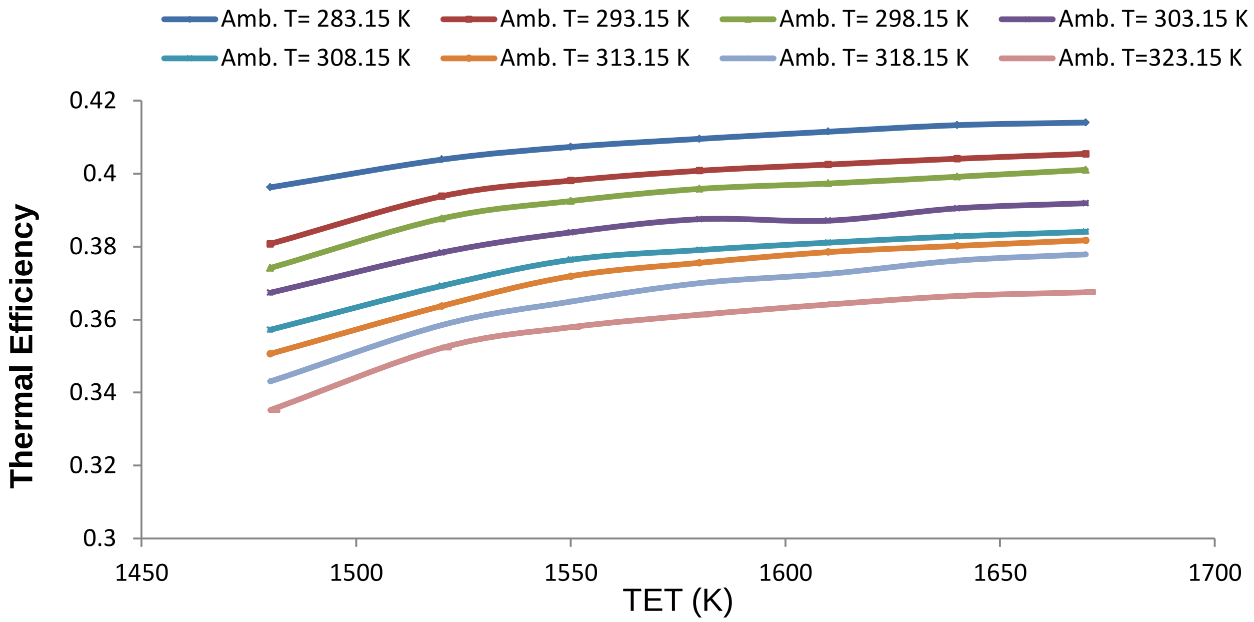

Figure 3-4: Thermal efficiency against TET (K) for 40.7 MW SCTS

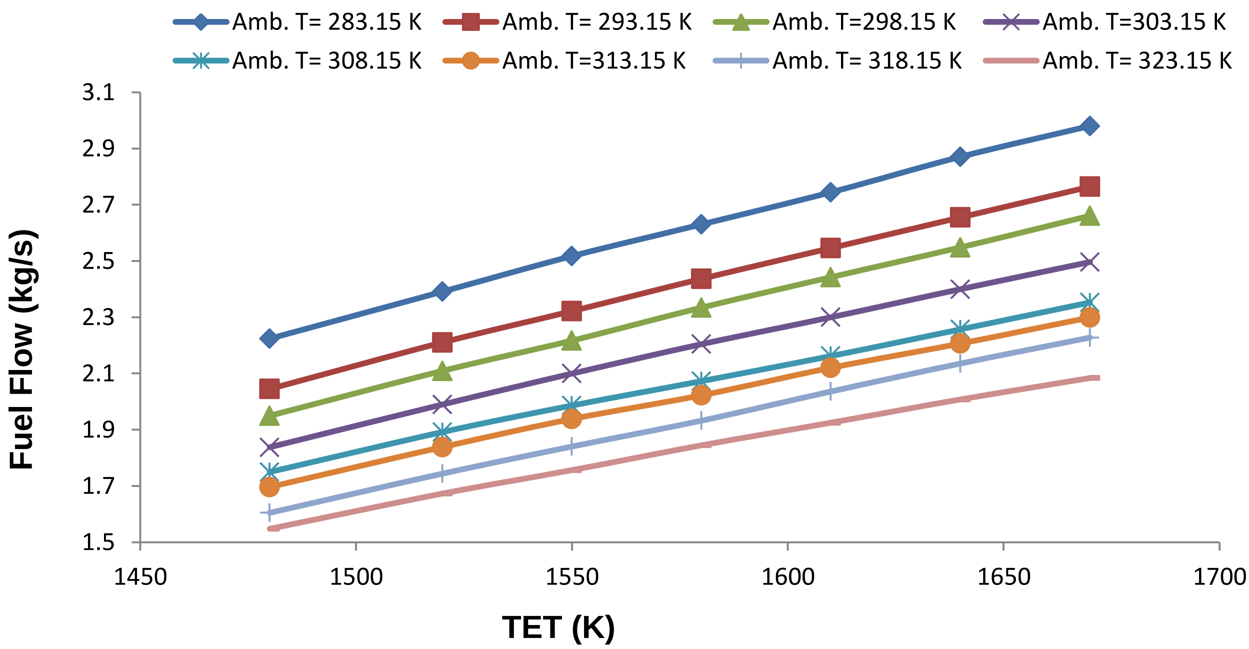

Figure 3-5: Fuel flow against TET for 40.7 MW SCTS

Figure 3-5 shows the change in fuel flow against TET different ambient temperatures. At off-design condition having higher ambient temperature than the design point, the fuel flow increases and this is a major parameter in the establishment of the life cycle cost of the plant and the general natural gas pipeline system. The analysis of this important data is discussed in a later chapter in this thesis.

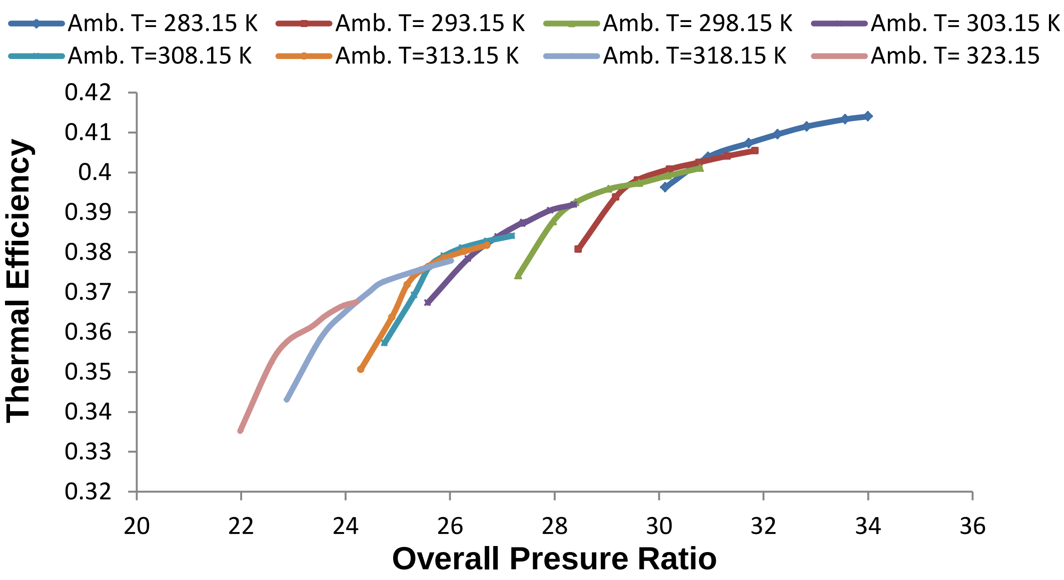

Figure 3-6: Thermal efficiency versus pressure ratio for 40.7 MW SCTS

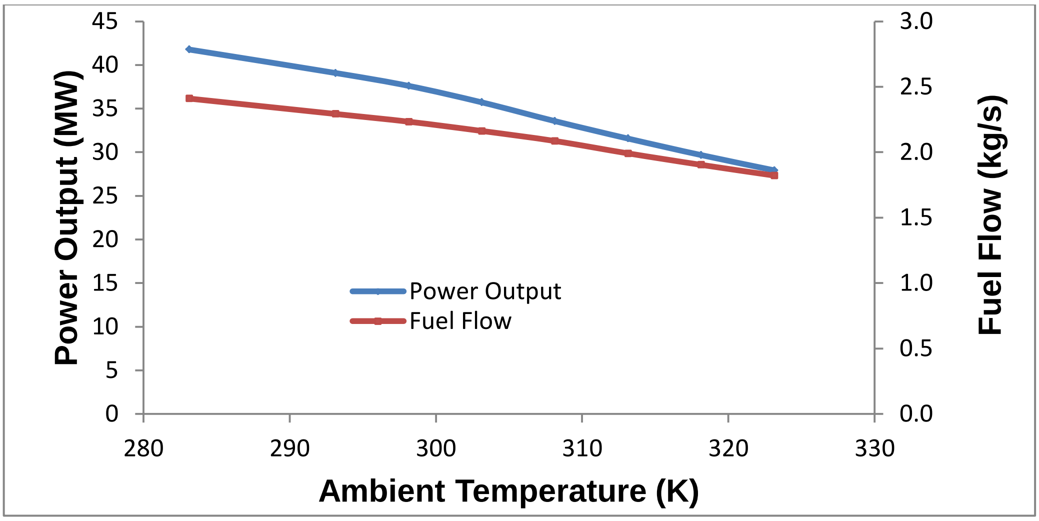

Figure 3-7: Variation of power output and fuel flow with ambient temperature at constant TET for 40.7 MW SCTS

At constant TET, the power output and fuel flow variation with ambient temperature is shown in Figure 3-7. As the ambient temperature increases, the power output decreases with consequent reduction in fuel flow. A 6.8% drop in output power which is equivalent to 2.7 MW occurs with a 3.5% rise in ambient temperature, and consequent 5.1% reductions in fuel flow.

3.3.4 33.6 MW Single Spool Simple Cycle

This gas turbine was modelled as a simple cycle with the configuration of single spool and the gas generator which drives the compressor aerodynamically connected to the a single turbine. The compressor pressure ratio at design point is 25.0:1. The power turbine has output shaft power of 33.6 MW at design point. The off-design operating range considered for the simulation using TURBOMATCH is ambient temperature ranging from 10^{o}C to 45^{o}C, this corresponds to the ambient condition along the natural gas pipeline route. The results of the off-design simulation showing the effect of ambient temperature on the performance parameters of the gas turbine are presented in Figure 3-8 to Figure 3-11. The effect of ambient temperature on the power output and fuel flow at constant TET is presented in Figure 3-12.

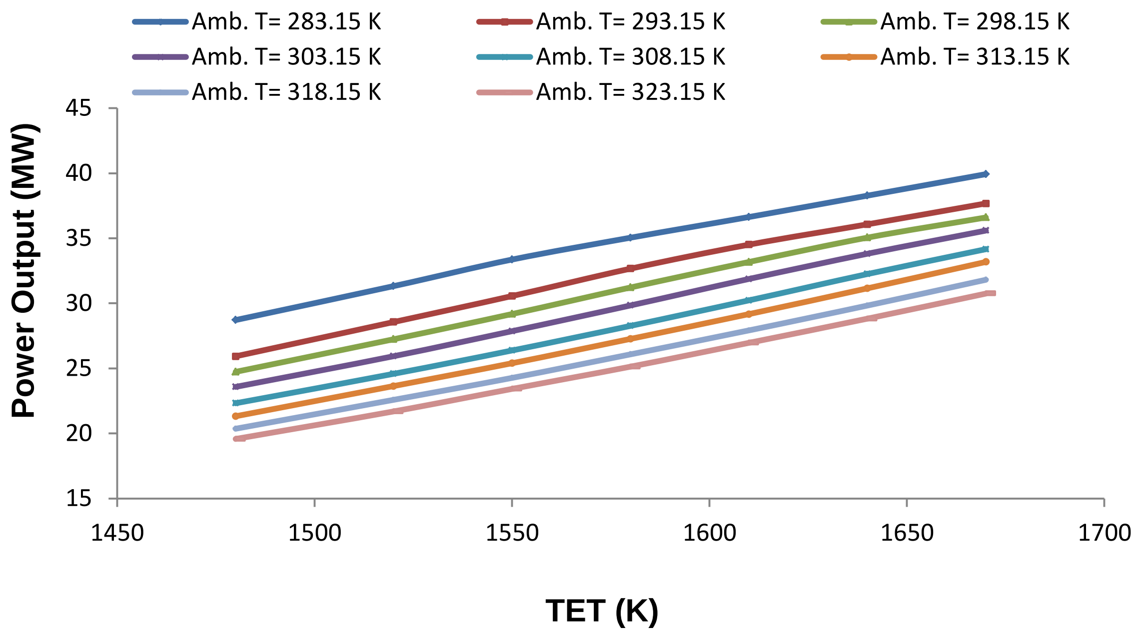

Figure 3-8: Power output against TET for 33.6 MW SSSC

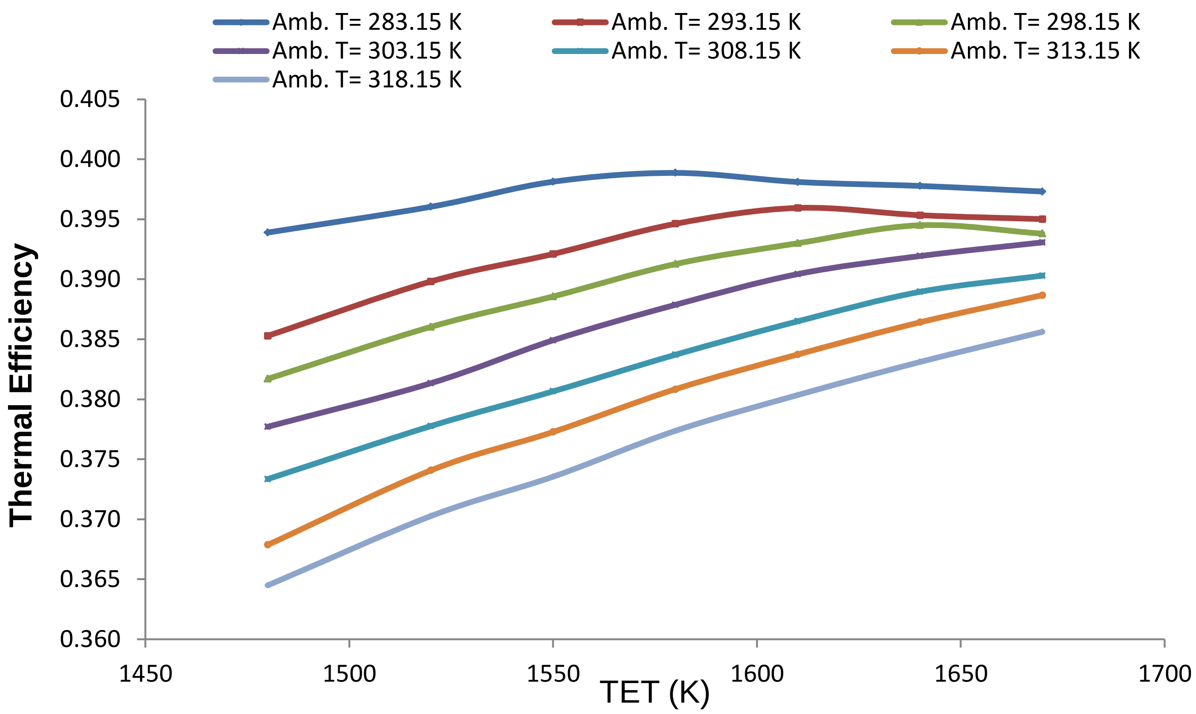

Figure 3-9: Thermal efficiency against TET (K) for 33.6 MW SSSC

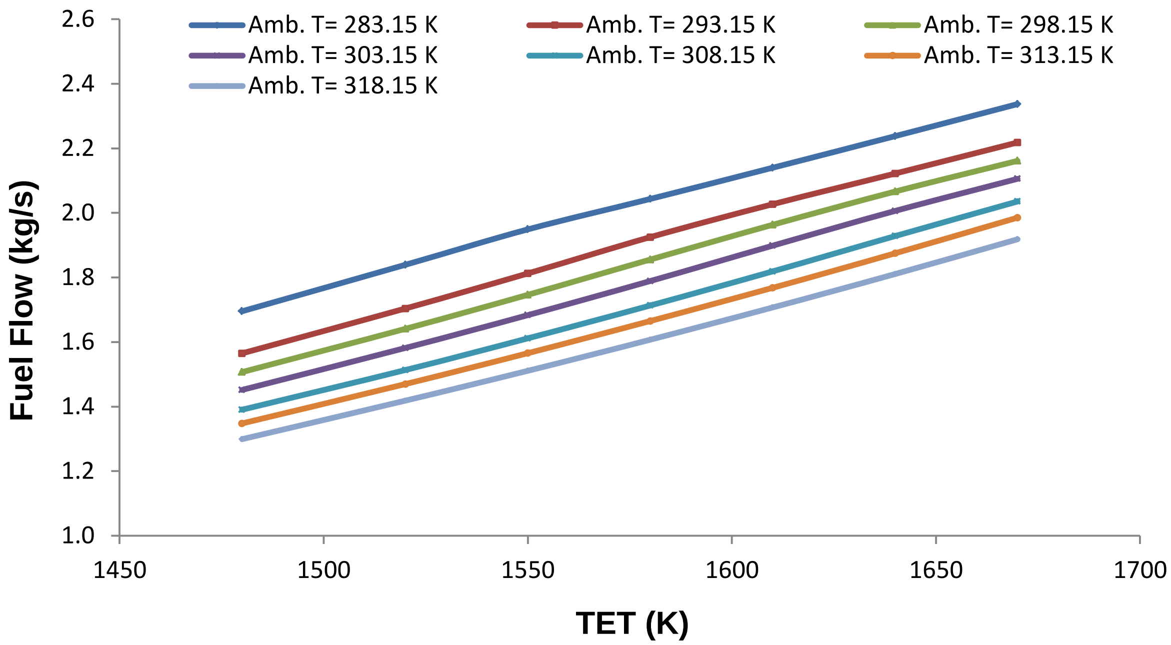

Figure 3-10: Fuel flow against TET for 33.6 MW SSSC

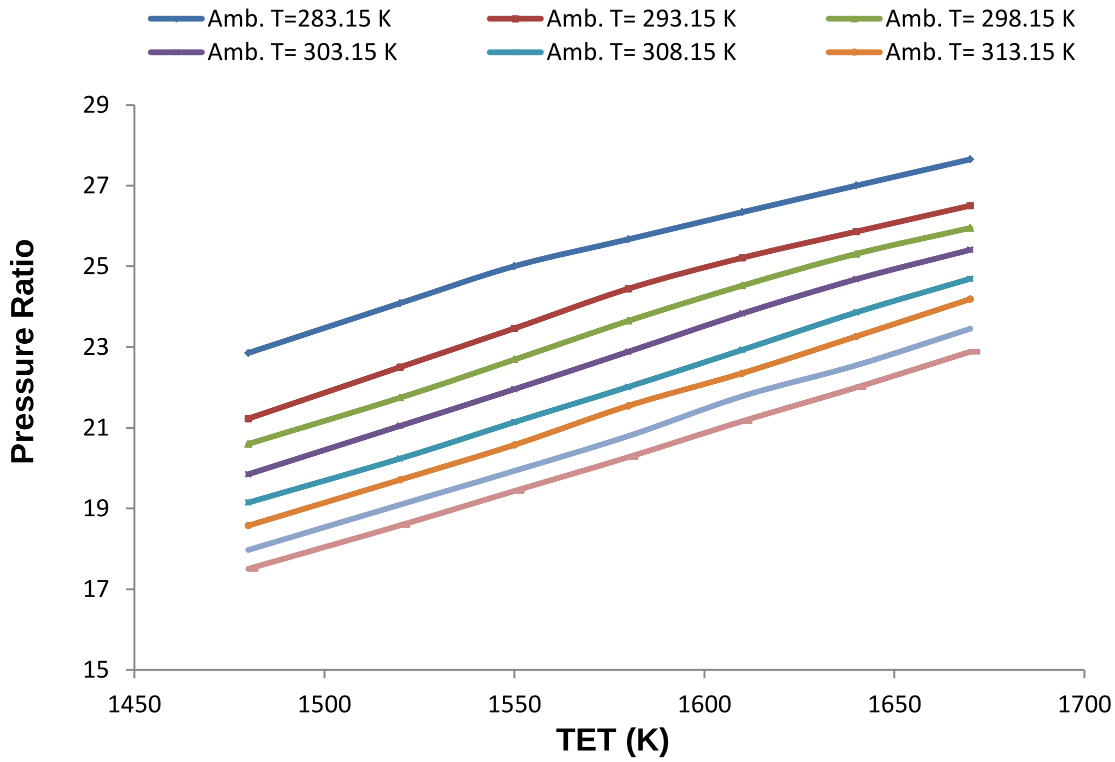

Figure 3-11: Effect of TET on pressure ratio for 33.6 MW SSSC

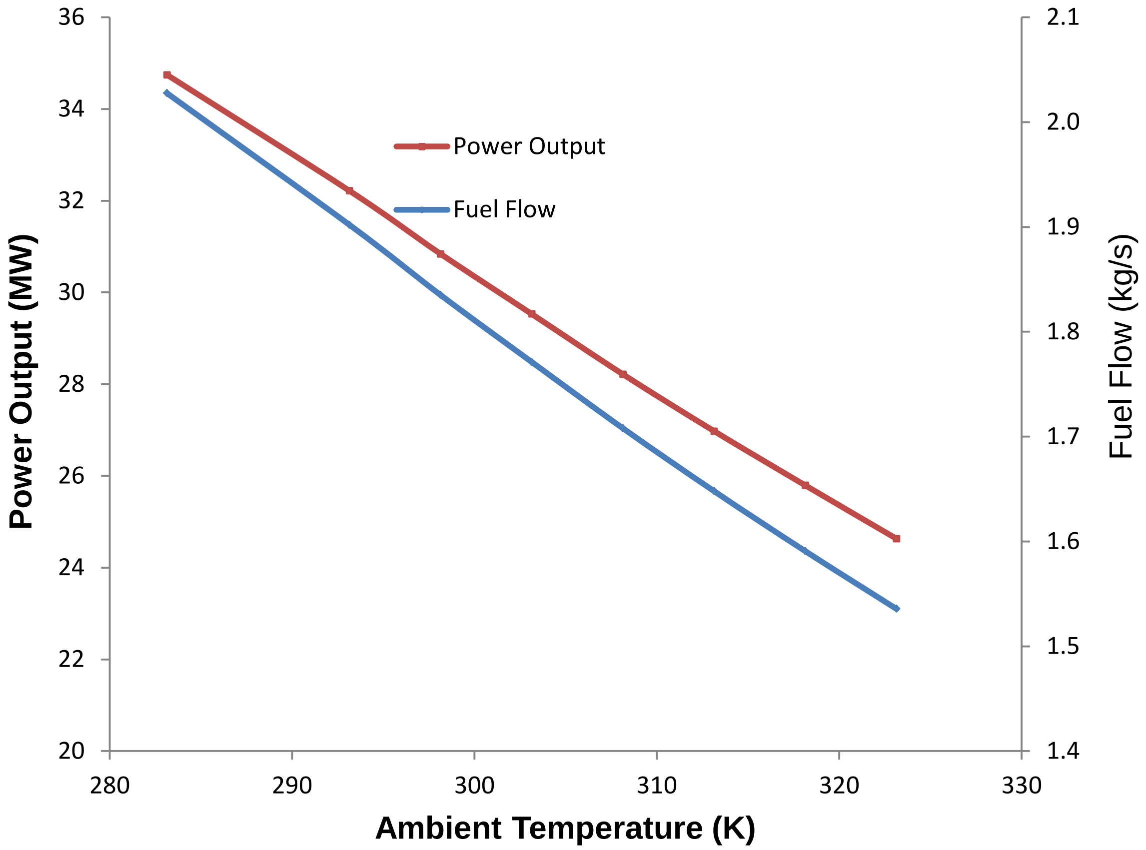

Figure 3-12: Effect of ambient temperature on power output and fuel flow at constant TET for 33.6 MW SSSC

With air mass flow of 90 kg/s and a TET of 1575 K, the thermal efficiency of the gas

turbine was found to be 39.60%.

3.3.5 Gas Turbine Emissions

Since its advent, gas turbine has been the focus of intensified and concerted research and development efforts concentrating on efficiency boosting, both in industry and academia. This interest was pursued vigorously, both within the industrial sector and the academia, with increase in TET being an important focus. Material limitations were thought to be the primary constraints on efficiency until environmental concerns over exhaust emissions and air pollution became an important issue, in view of the prevailing environmental protection regulations and directives. The introduction of stringent environmental regulations led to gas turbine practitioners changing their focus towards the design of gas turbine with reduced emissions to meet these regulations.

Emissions are among the key challenges in the development of gas turbine to meet environmental regulations (Nasir et al., 2012). Emissions from gas turbine include oxides of carbon (CO_x), oxides of nitrogen (NO_x), unburned hydrocarbons (UHC), and smoke. These materials have negative impact on the environment. Generally, high combustion temperature is associated with high efficiency and high pollution, two contradictory requirements that need to be balanced (Abdul-Wahab et al., 2005). An important characteristic of a simple cycle gas turbine combustor is the amount of nitrogen oxide (NOx) emitted in the exhaust gas. The types of pollutant emitted are primarily determined by the type of fuel used, as the sulphur content of a fuel determines the emissions of sulphur oxides. Generally, SO_x emissions are greater when heavy oils are fired in the turbine. In principle the products of complete combustion of fossil-fuel are water (H_2O) and carbon dioxide (CO_2). CO_2 is a greenhouse gas because of its contribution to the depletion of the ozone layer and consequently global warming. It is therefore considered a pollutant. UHC, particulate matter (PM), and sulphur dioxide (SO_2) are becoming important because of the limits imposed by the air quality regulations, although the emission of oxide of sulphur is being addressed by fuel treatment, while the emissions of NO_x and CO are generally considered significant. Table 3-2 shows the By-product of combustion and its effect on the environment.

Table 3-2: By-product of combustion (Abdul-Wahab et al., 2005)

By-product	Source	Effect
CO	From incomplete combustion	A toxic exhaust gas, continuously monitored
C_nH_m	Also called unburned hydrocarbon (UHC)	Strong-smelling, potentially carcinogenic substances
VOC	Without methane and ethane, the UHC is called volatile organic compounds (VOC)	VOC contributes to formation of ground-level atmospheric zone
Smoke	Soot particles	From incomplete complete combustion; assist in the formation of carcinogenic substances
NO_x	NO and NO_2 generated from air at very high temperature	Negative effects on plants; part of infamous acid rain (HNO_3) and destroys the ozone layer of the earth
SO_x	SO and SO_2, unavoidable oxidation products when the fuel contains sulphur	Major components of acid rain (H_2SO_3 and H_2SO_4)
PM	Particulate matter	Soot and other particulates from incomplete combustion
PM 10	Particulate matter smaller than 10 microns	Natural gas combustion produces a very small amount
PM 2.5	Particulate matter smaller than 10 microns	Found mostly in liquid fuel combustion

The main emission concern of this research is the emission of oxides of carbon and nitrogen (CO_x) (NO_x). This is primarily because the gas turbine under study is natural gas fired gas turbine and the predominant pollutant generated by this type of gas turbine is $CO_{x.}$ This is a gas with very high contributory effect to global warming and legislation and regulation on emissions are starting to affect the decision making process for building new natural gas compression stations. Nearly all the fuel carbon is converted into CO_2 during the combustion process. The amount of CO_2 emitted is a function of amount of fuel used, the carbon content of the fuel and the efficiency of the combustion system. The carbon content of natural gas is 14.4 metric tonnes carbon per terajoule. Ambient temperature has a direct effect on the power output of a gas turbine which is the sole controller of the amount of fuel consumed. This trend implies that the amount of CO_2 emitted is invariably a function of the ambient temperature.

3.3.6 NO_x and CO_x Formation in Gas Turbines

Nitrogen oxide (NO_x) emissions, a mixture of mostly nitric oxide (NO) and nitrogen dioxide (NO_2) in variable composition are the primary pollutants generated by gas turbine. Although gas turbines are described as NO_x emitters, most of the NO_x is emitted in the form of NO which is subsequently oxidised in the atmosphere to produce NO_2. The formation of NO_x is based on three mechanisms: thermal NO_x, prompt NO_x, and fuel-bound NO_x. The predominant NO_x formation associated with gas turbine is the thermal NO_x (Zachary, 2001). At high combustion temperature similar to gas turbine operations, thermal dissociation and subsequent oxidation of atmospheric nitrogen takes place, and this is the only significant source of gas turbine NO_x emissions. Assuming constant ambient conditions (temperature, pressure and humidity), the rate of formation of thermal NO_x is highly dependent on combustion temperature, air-to-fuel ratio and residence time. Combustor design and percentage load have significant effect on the aforementioned factors on which the formation of thermal NO_x is dependent. The higher the load, the higher the temperature and the higher the emission of thermal NO_x. The rate of formation of thermal NO_x increases with increased temperature. CO_x is a major product of combustion of hydrocarbons fuels especially natural gas. Carbon monoxide (CO) is a poisonous gas emitted by gas turbine as a result of incomplete combustion which may be attributed to insufficient residence time at high temperature or incomplete mixing, which inhibits the final step in fuel carbon oxidation. CO is constantly monitored and for health and safety reasons, it is usually regulated to levels below 50 ppm (Energy Nexus Group, 2002). CO is oxidized to CO_2 under a favourable temperature and the availability of oxygen. The oxidation rate is slow at low gas turbine combustion temperature; this makes the formation of NO_x and CO a conflicting case. Figure 3-13 shows NO_x and CO emission versus combustion temperature. The emission of thermal NOx increases with increased combustion temperature while the emission of CO decreases with increased combustion temperature.

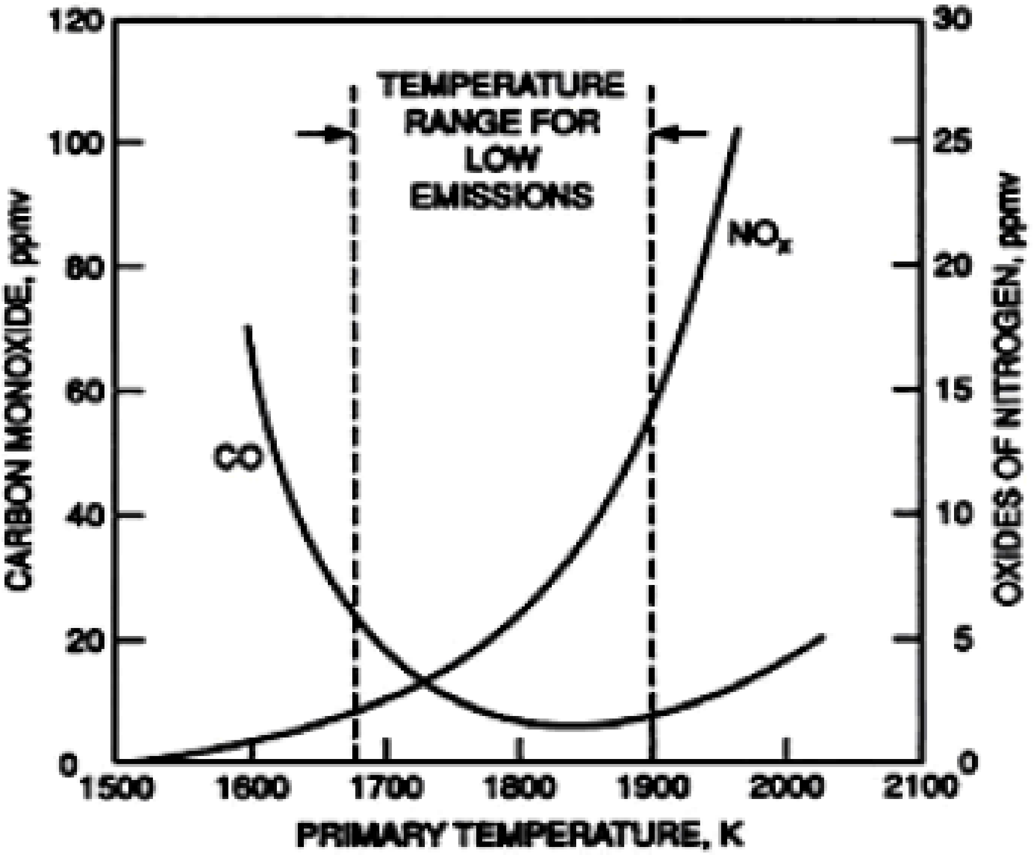

Figure 3-13: NOx and CO versus combustion temperature (Rokke et al., 2003)

3.3.7 Factors Affecting Formation of Emission

The amount of exhaust emission in a gas turbine is influenced by a number of factors which affect the combustion reactions; these include ambient conditions, operating conditions, combustion and post-combustion conditions. Ambient conditions affect the air charge entering the gas turbine. As the ambient temperature decreases, the mass flow increase with consequent increase in fuel flow and power output. Also with reduced ambient temperature or increased relative humidity, the peak combustion temperature decreases, and this inhibits NO_x emission.

The results of the fuel flow obtained from the off-design performance simulation of the investigated gas turbines were used to estimate the annual CO_2 emissions, as well as the annual emission cost. The total effect of CO_2 emission on the life cycle cost of the entire natural gas pipeline system is presented and discussed in a later chapter in this thesis.

3.3.8 CO_2 Emission Calculations

The calculation of the amount of CO_2 emitted is based on the genuine assumption that complete combustion of the fuel takes place in the presence of excess air. The general equation for complete combustion of a hydrocarbon fuel of alkane group of which natural gas predominantly methane is one can be written as:

$$2C_nH_{2n+2} + [3n+1]O_2 \rightarrow 2nCO_2 + [2n+2]H_2O + Heat \qquad (3.1)$$

And for generally for all hydrocarbon fuel the combustion equation can be represented as:

$$C_xH_y + nO_2 \rightarrow n_1CO_2 + n_2H_2O + Heat \qquad (3.2)$$

For a molar balance of equation 3.2, we have

$$n_1 = x$$

$$n_2 = 0.5y$$

$$n = n_1 + 0.5\, n_2$$

For the combustion of methane, the equation for its complete combustion is given as:

$$CH_4 + 2O_2 \rightarrow 2H_2O + CO_2 \qquad (3.3)$$

I mole of methane reacting with 2 moles of oxygen will produce 1 mole of carbon dioxide.

Carbon has 12.0107 g/mol and hydrogen has 1.00794 g/mol, therefore, 1 mole of methane weighs:

$$(12.0107\ \times 1) + (4\ \times 1.00794) = 16.04\frac{g}{mol} \qquad (3.4)$$

1 mole of CO_2 weighs:

$$(12.0107 \times 1) + (15.9994 \times 2) = 44.01\frac{g}{mol} \qquad (3.5)$$

Having established the molecular weights of fuel and the CO_2 produced per mole of fuel, the mass of fuel consumed by the gas turbine over the off-design condition and operating period are obtained from the gas turbine simulation (TURBOMATCH). This fuel flow results is used to calculate the amount of CO_2 emitted over the period and conditions of operations of the gas turbine. The amount of CO_2 emitted goes into the economic module where the cost implication is computed on annual basis and over the entire life of the project. Figure 3-14 and Figure 3-15 shows the effect of the ambient temperature on the CO_2 emission, as well as the cost incurred for 40.7MW SCTS and 33.6MW SSSC respectively.

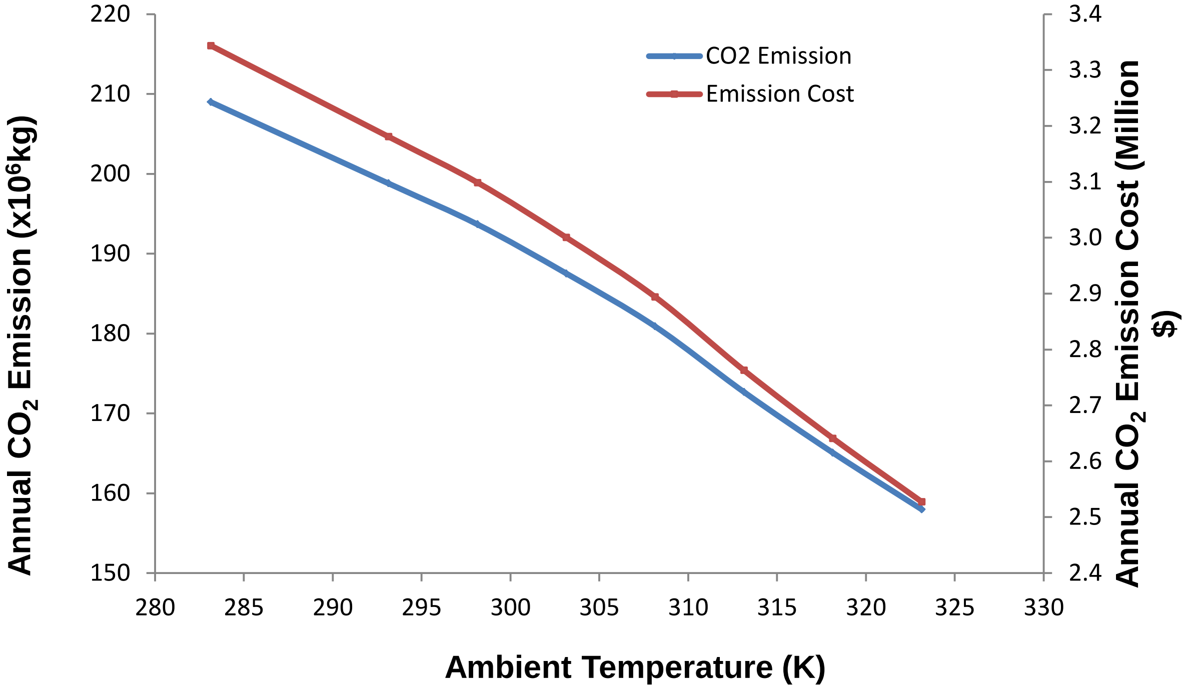

Figure 3-14: Effect of Ambient Temperature on the CO_2 Emission and Emission Cost for 40.7 MW SCTS

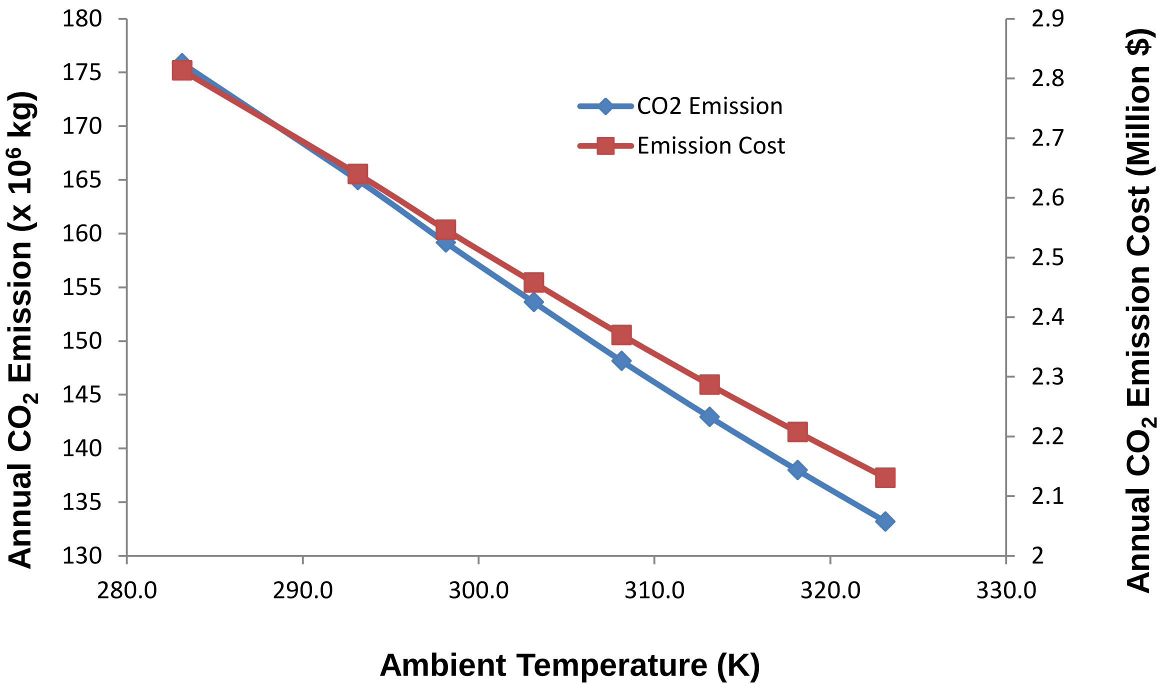

Figure 3-15: Effect of Ambient Temperature on the CO2 Emission and Emission Cost for 33.6 MW SSSC

3.3.9 Some Environmental and Health Impact of NO_x and CO_x Emissions

The emission of NO_x and CO_x causes a wide range of health and environmental impacts because of their various compounds and derivatives which includes nitrogen dioxide (NO_2), nitric acid (HNO_3), nitrous oxide (N_2O), nitrates (NO_3), and nitric oxide (NO), CO, CO_2 etc. In the presence of heat and sunlight NO_x can react with volatile organic compounds (VOCs) to form ground level ozone which has adverse health effect on human lungs and can cause damage to vegetation and reduce crop yield. It can also react with sulphur dioxide to form acid rain, which falls to earth as rain, fog, snow or dry particles. Acid rain causes damage to ecosystems by making lakes and streams acidic and unsuitable for fish. Its cause burns on human skin and also destroys vegetation. Excessive presence of nitrogen in bodies of water upsets the chemical balance of nutrients used by aquatic plants and animals.

CO is hazardous to human health and its oxidation produces CO_2. CO_2 and N_2O are greenhouse gases. These accumulate in the atmosphere with other greenhouse gases to cause global warming, a gradual rise in temperature of the earth which will lead to increased risk to human health, a rise in the sea level and other adverse changes to plant and animal habitat. NO_x can also react with common organic chemicals to form a wide variety of toxic products such as nitrate radical, nitoarenes and a host of others. Some of these can cause biological mutations. The presence of nitrate particles and nitrogen oxide in certain percentages can block transmission of light and thereby reduce visibility.

3.4 Electric Motor Drive Option

Other than gas turbine, another viable prime mover option for natural gas compressor is the electric motor. In the last decade, electric motor driven compression has become more common in the natural gas industry. Many of the components of an electric motor drive system have undergone technological changes to meet the needs of gas compressor applications. The evolving of variable frequency drives (VFD), variable speed drives (VSD), and motors with advance bearing technologies has provided pipeline compression a more efficient drive system, with larger and more flexible operating envelopes. There is better energy conversion efficiency in electric motor than in gas turbine, with over 95% of the electrical energy coming in being converted into mechanical energy going out. This high electric motor efficiency can further be improved with VSD which modulate motor output by varying the speed of the motor itself, rather than through the use of control valves. Figure 3-16 shows Electric Motor parts.

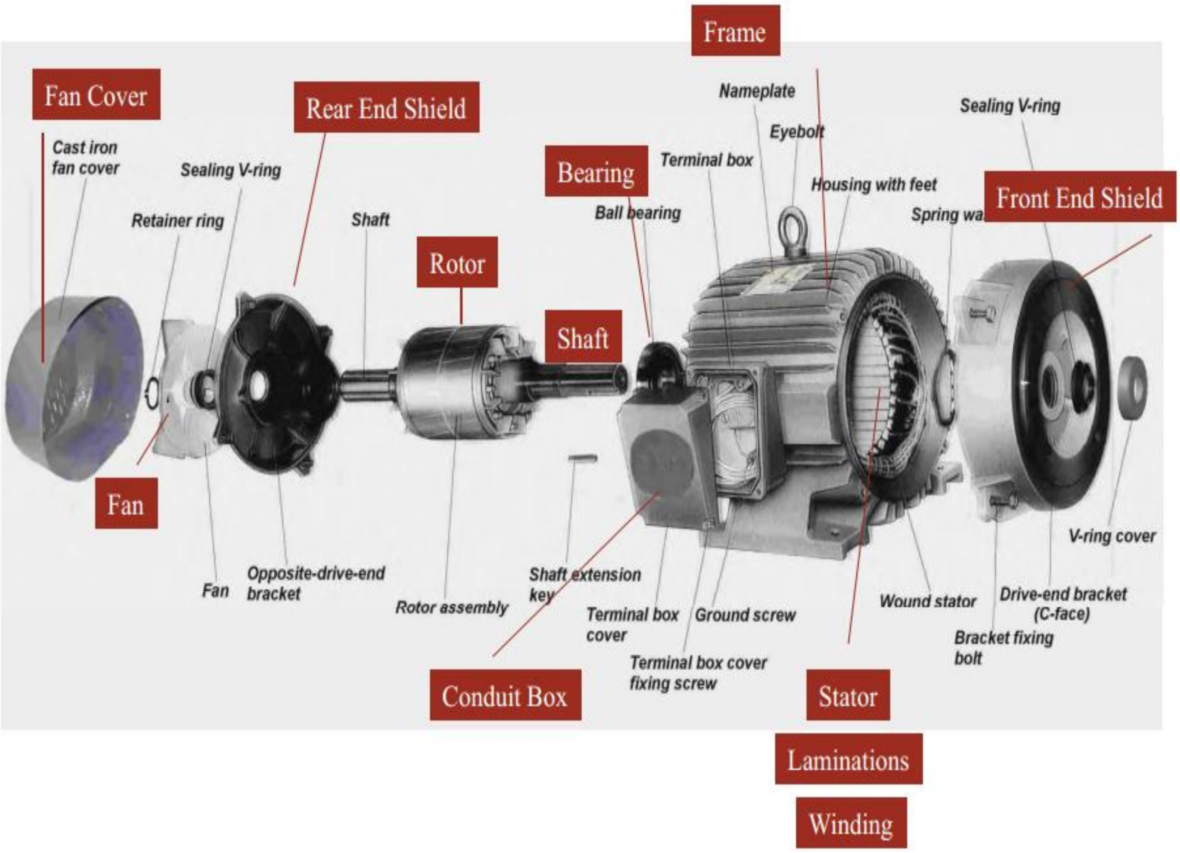

Figure 3-16: Electric motor parts (Duhaime, 2012)

3.4.1 Types of Electric Motor

Electric motors can be classified as either alternating current (AC) or direct current (DC) motor, depending on the type of current on which its operation is based. In pipeline compression, the commonly employed type of electric motor is AC and induction or asynchronous motors. Figure 3-17 shows the classification of electric motors.

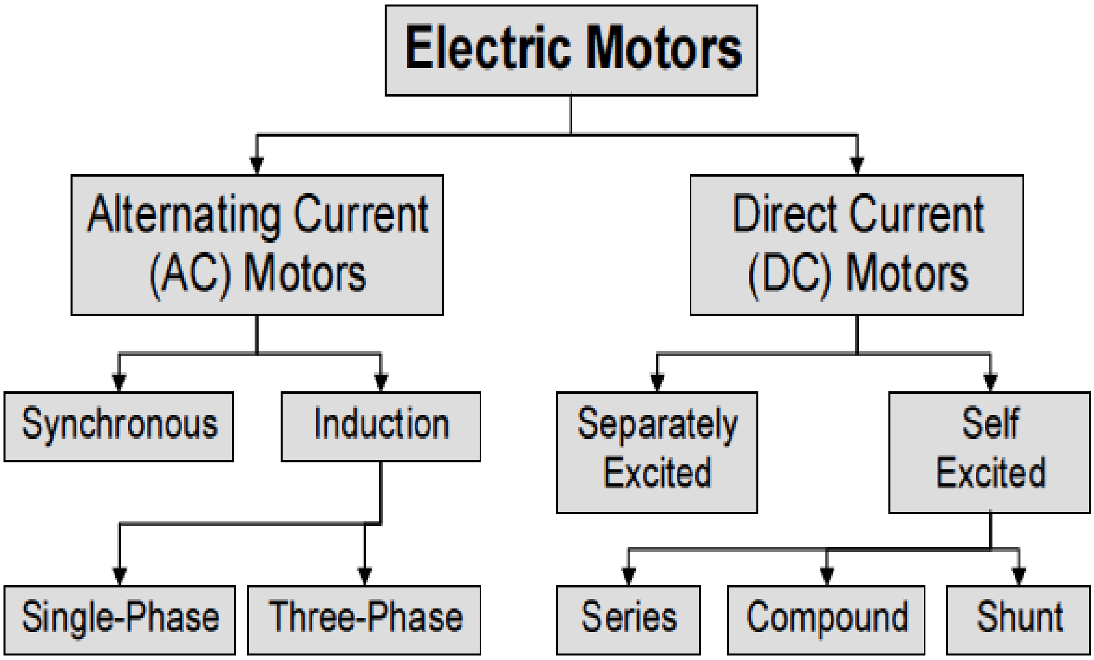

Figure 3-17: Classification of Electric Motors

3.4.1.1 Induction (AC) Motor

Induction motors are the common motors used for various equipment in industry. Their popularity is due to their simple design, ruggedness, low cost and easy maintenance and can be directly connected to an AC power source. The induction motor works by inducing current in the rotor through the small air gap between the stator and the rotor. The stator current generates a rotating magnetic field in the air gap between the stator and rotor. The interaction of the induced rotor current with the rotating magnetic field generates a torque on the rotor. The synchronous speed which is the rate of rotation of the rotating magnetic field created by the stator is given by the equation

$$n_s = \frac{60 \times f}{p} \qquad (3.6)$$

Where f is the frequency of the AC supply current in Hz and p is the number of magnetic pole pairs per phase.

A major characteristic of the induction motor is the presence of slip (S) which is the difference between the rotating speed of the magnetic field (synchronous speed) and the rotating speed of the rotor. This determines the motor's torque and can be calculated from

$$\%S = \frac{n_s - n_r}{n_s} \qquad (3.7)$$

where n_s is stator electrical speed and n_r is rotor mechanical speed The base speed of the induction motor is based on the number of magnetic poles of the machine. The speed of the rotating magnetic field or synchronous speed is determined by the number of poles or the windings in the stator. If there is no load on the motor shaft, the rotor will turn at a speed that slightly lags the synchronous speed, which defines the slip of the induction motor.

As the motor is loaded, the difference between the synchronous speed and the motor speed increases, this in turn increases the percentage of slip. When slip increases, a higher current is induced in the rotor bars, the interaction of the two currents become stronger and a higher torque is provided to the motor load. Table show typical operating speed for induction motors for no-load and full-load cases.

The induction motor speed can be controlled to suit natural gas compressors requiring variable speed operation. The speed control can be achieved by varying input voltage, varying input frequency, changing the winding pole number or by varying input frequency and voltage together. Compressor variable speed requirement can also be met by a constant speed motor, utilizing a variable speed gearbox. The rotor speed, which is the speed delivered to the pipeline compressor if it is a direct coupling or to a gear arrangement if this exists, can be obtained from equation 3.8 with the knowledge of the percentage slip and synchronous speed. Table 3-3 shows typical operating speed for induction motors.

$$n_r = (1 - s)n_s \tag{3.8}$$

Table 3-3: Typical operating speed for induction motors for no-load and full-load case (Nored et al., 2009)

No. of Poles	Synchronous speed with no slip rev/sec (RPM) for 60 Hz supply	Actual speed with full load and slip rev/sec (RPM) for 60 Hz supply
2	60 (3600)	58.8-59.4 (3528-3564)
4	30 (1800)	29.4-29.7 (1764-1782)
6	20 (1200)	19.4-19.8 (1164-1188)
8	15 (900)	14.7-14.8 (882-888)

3.4.1.2 Synchronous (AC) Motor

A synchronous motor is an AC motor which runs at constant speed fixed by frequency of the ssystem i.e. it rotates in synchronism with the stator's magnetic field (no slip). The rotor of a synchronous motor requires direct current (DC) for excitation to provide the rotor magnetic field. It has low starting torque and is therefore suited for applications that start with a low load. Synchronous motors cannot be run directly from the AC power line as synchronous motor controller is required for rotor control. The rotor magnetic field is controlled by controlling the field current. Once a synchronous motor is operating at rated speed, the angular displacement between the stator and the rotor magnetic field will change with the load. The speed of a synchronous motor can also be varied, just like that of the induction motor.

3.4.2 Drive Train Configuration

Working within the operational speed of the compressor is of utmost importance in gas compression systems, because centrifugal compressors operates efficiently in terms of capacity control by varying speed. Capacity control in centrifugal compressor without speed control involves suction or discharge throttling or recycling gas. These capacity control options are significantly less efficient than changing the rotational speed of the centrifugal compressor. This makes it important for electric drive for natural gas compression system to be equipped with an adjustable speed system which is typically accomplished through a variable frequency drive (VFD) controlling the motor, or a variable speed hydraulic drive (VSHD) with a fixed speed motor. The primary issue specific to gas compression which guides the selection of drive train is the operational speed of the compressor. Cost, complexity and other issues patterning to electric motor needs to be considered before deciding what drive train is selected. The four commonly used type of drive train are presented in the next sub-topics.

3.4.2.1 Direct Drive Train (With and Without VFD)

This is the preferred drive train option because of its simplicity and cost. This involves driving a compressor with a motor operating at the same speed. This eliminates the need for a gearbox. The motor speed must be controllable over the operating speed range of the compressor. Figure 3-18 shows the general direct drive train.

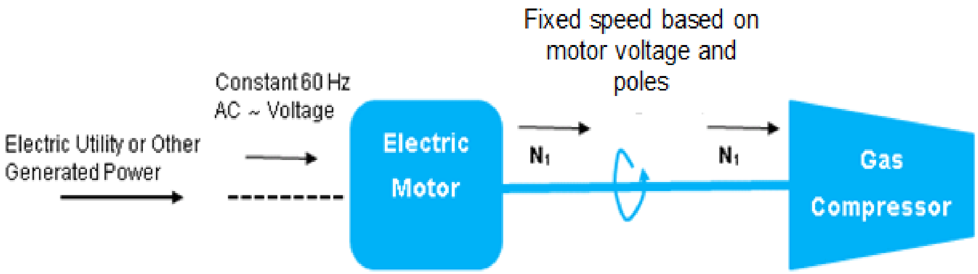

Figure 3-18: Single Motor Directly Driving Gas Compressor at Fixed Speed

In order to be able to vary the compressor speed in a direct drive train, a variable frequency drive unit, which varies the input frequency and voltage supplied, thus changing the effective synchronous speed of the motor, is used. The motor speed changes in proportion to the VFD controller. Figure 3-19 shows the direct drive train with VFD incorporated to handle speed variation with the operating limits of the compressor.

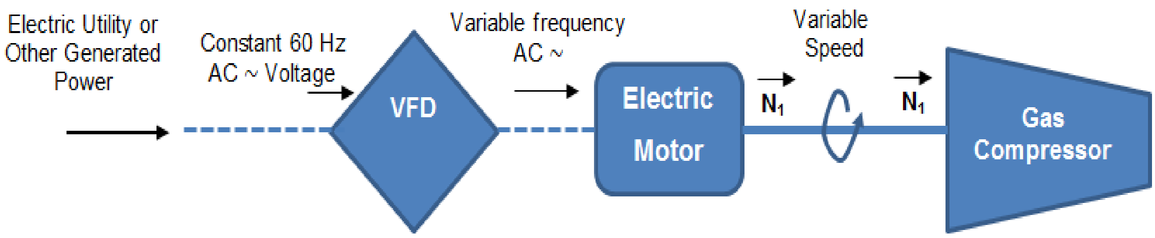

Figure 3-19: Direct Drive Train with VFD

The use of a multi-speed motor is an alternative method of accommodating compressor variable speed capacity control requirements without the use of VFD. The major disadvantage of this method is that the compressor speed can only change between limited points as the maximum available multi-speed motor is a four-speed motor.

3.4.2.2 Motor with Gearbox

A speed changing gear is required if the fixed speed motor speed does not fall within the compressor speed operational window. The gearbox is used to increase the motor speed to match the compressor speed requirements. Figure 3-20 shows the drive train of elctric motor with gearbox.

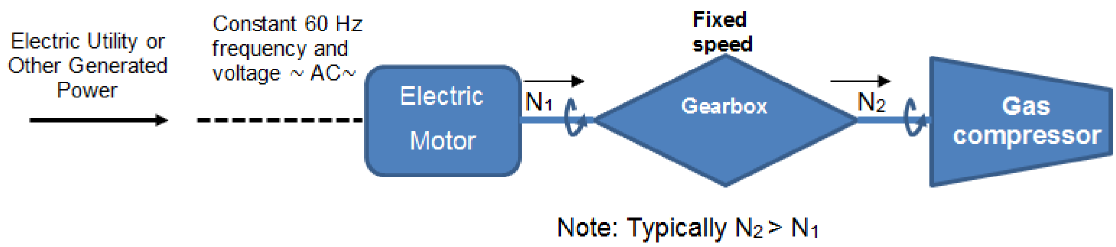

Figure 3-20: Electric motor with Gearbox - without VFD

3.4.2.3 Variable Speed Hydraulic Drive (VSHD)

A variable speed hydraulic drive which uses a mechanical gearbox in combination with a variable speed hydraulic pump or motor may be used to vary the speed and torque supplied to the compressor. Hydraulic couplings are also used to decouple the electric motor from the drive and also serve to effectively dampen any torque ripples produced by the electric motor. Figure 3-21 shows the drive arrangement for variable speed hydraulic drive.

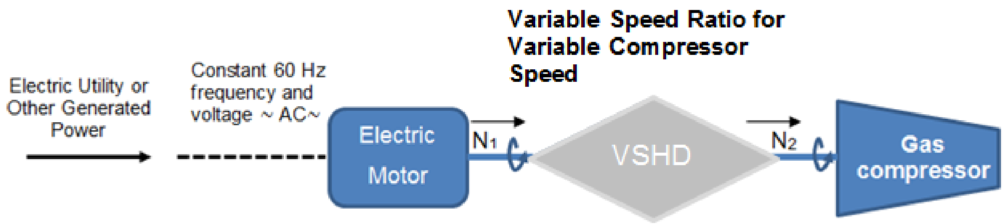

Figure 3-21: Drive Train with Variable Speed Hydraulic Drive

3.4.3 Motor Performance

Motor performance is characterized in terms of torque, speed and delivered shaft power. Motor apparent power requirements and permissible loading of the motor beyond rated conditions are characterized using the power factor and service factor. In the performance an of electric motor, current and torque are often expressed as a percentage of the full load values. Torque represents the motor rotational work necessary to match the resistance to turning of the shaft caused by the driven load.

Some electric motors are designed to run at 50% to 100% of rated load. Maximum efficiency is usually near 75% of rated load. Thus, a 10-horsepower (hp) motor has an acceptable load range of 5 to 10 hp; peak efficiency is at 7.5 hp. A motor’s efficiency tends to decrease dramatically below about 50% load.

3.4.4 Development of Equivalent Circuit of an Induction Motor

In developing an equivalent circuit of an induction motor, the similarity between a transformer and induction motor is considered. The primary of the transformer is similar to the stator of the induction motor and the rotor corresponds to the secondary of the transformer. It follows from this analogy that the stator and the rotor have their own respective resistances and leakage reactance. A magnetizing reactance exists because the rotor and the stator are magnetically coupled. The air gap in an induction makes the magnetic circuit relatively poor, thus the corresponding magnetizing reactance will be relatively smaller than that of transformer. The hysteresis and eddy current losses in an induction motor can be represented by a shunt resistance, as was done for the transformer.

3.4.5 Torque and speed curve

To fully analyse the performance of an induction motor, it is imperative to analyse the electric circuit which can be represented thus:

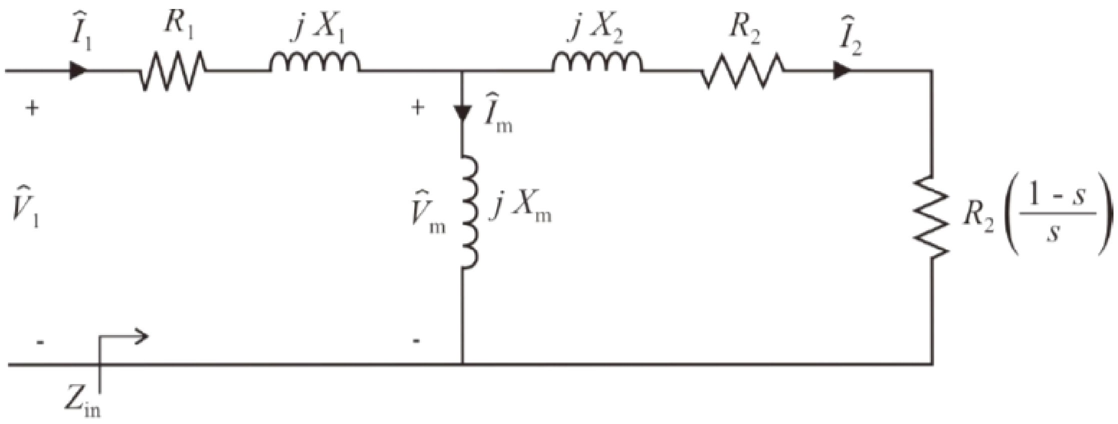

Figure 3-22: Equivalent circuit of an induction motor

Figure 3-22 shows the equivalent electric circuit of an induction motor which consists of the traditional five parameters (i.e. stator resistance R_1, stator leakage reactance X_1, magnetizing
Reactance X_m, rotor leakage reactance X_2, and rotor resistance R_2.

From Figure 3-22, it can be seen that the input impedance Z_{in} can be determined once the slip is calculated as defined in equation 3.7.
The input impedance can be calculated as shown in equation

$$Z_{in} = R_1 + jX_1 + jX_m || \left(\frac{R_2}{s} + jX_2\right) \qquad (3.9)$$

Equation 3.9 gives equation 3.10

$$Z_{in} = R_1 + jX_1 + \frac{jX_m\left(\frac{R_2}{s} + jX_2\right)}{\frac{R_2}{s} + j(X_m + X_2)} \qquad (3.10)$$

With the knowledge of the motor source voltage, $\widehat{V_1}$, the stator current can be computed using equation 3.11.

$$\hat{I}_1 = \frac{\widehat{V_1}}{Z_{in}} \qquad (3.11)$$

The rotor current can be determined from current divisions as

$$\hat{I}_2 = \left(\frac{jX_m}{\frac{R_2}{s} + jX_2 + jX_m}\right)\hat{I}_1 \qquad (3.12)$$

The rotor current is flowing through the term $R_2/2$, which may be represented as the series of combination of a pure resistance R_2 and a back-emf term $R_2\left(\frac{1-s}{s}\right)$. The mechanical torque can then be computed as the power into the back-emf term divided by the mechanical speed. This results in

$$\tau_m = \frac{3I_2^2 R_2}{\omega_m}\left(\frac{1-s}{s}\right) \qquad (3.13)$$

Figure 3-23 shows the percentage torque and synchrous speed characteristic for an induction motor.

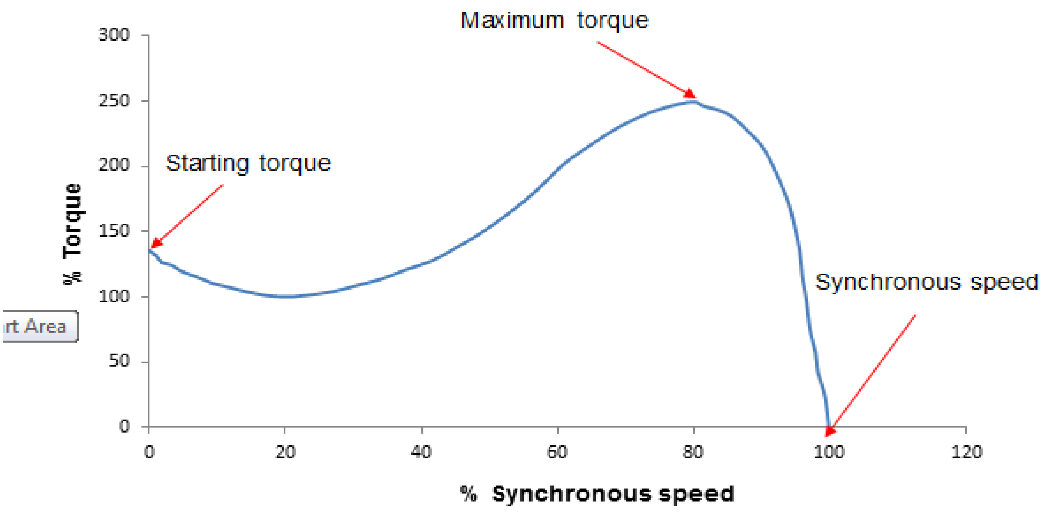

Figure 3-23: : %Torque- % Synchronous speed curve of induction motor

The torque versus speed relationship for the induction motor must be analysed carefully to ensure that all compressor required operating points may be met. The torque produced by an induction motor is a function of shaft power and the shaft speed, where the torque reduces with speed for constant power. This can be expressed as

$$\tau = 9.5493\,\frac{P_m}{n_r} \qquad (3.14)$$

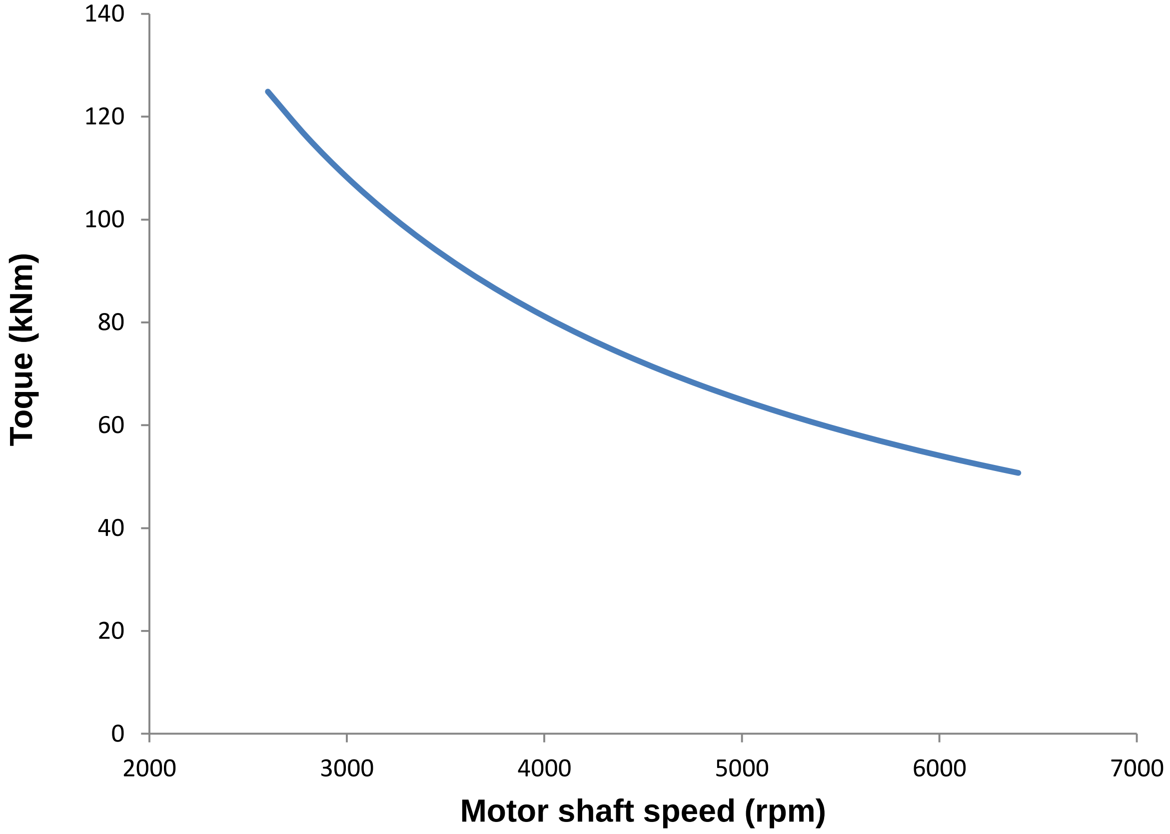

Figure 3-24: Torque- Speed curve of 34 MW Induction Motor

Table 3-4: Performance specification of Cypress HPL Motor

Parameters	**Value**
Rated power	34 MW
Maximum speed	6200 rpm
Efficiency at rated power	98 %
Number of pole	8
Rated voltage	4160 V

Table 3-5 Performance specification of Electric Motor Model MV7624

Parameters	**Value**
Rated power	24 MW
Maximum speed	7000 rpm
Efficiency at rated power	98 %
Number of pole	6
Rated voltage	3120 V

The electrical power input to

a. the 34 MW induction motor is given by

$$P_{in} = \frac{34}{0.98} = 34.69 \approx 34.7\, MW \qquad (3.15)$$

b. the 24 MW induction motor is given by

$$P_{in} = \frac{24}{0.98} = 24.89 \approx 24.9\, MW \qquad (3.16)$$

The torque developed at rated power of 34 MW is given by

$$\tau = \frac{9.5493 \times 34 \times 10^6}{6200} = 52367.1 \qquad (3.17)$$

$$\tau = 52367.1 \text{ Nm} \qquad (3.18)$$

And similarly the torque developed at rated power of 24 MW is

$$\tau = \frac{9.5493 \times 24 \times 10^6}{7000} = 32740.5 \qquad (3.19)$$

$$\tau = 32740.5 \text{ Nm} \qquad (3.20)$$

For a constant shaft power the torque decreases with increase in speed. The torque produced by a 34 MW induction motor running at 6300 rpm is 52367.1 Nm and for a rated power of 24 MW induction motor, the torque is 32740.5 at a speed of 7000 rpm.

3.4.6 Motor and Drive Train Efficiency

The motor losses and associated drive train component losses affect the cumulative efficiency for the drive input to the compressor. The following loss mechanisms were considered in estimating the motor losses:

- Stator I^2R losses which are due to stator windings
- Rotor I^2R losses which are due to rotor windings
- Core losses which are losses in iron due to fundamental magnetic field
- Friction and windage losses; mechanical losses which are due to friction, bearing and windage.
- Drive train component losses.

3.4.7 Electric Power Supply Sources

Two sources of electric power supply are open to pipeline operators: power supply from utility company which may or may not be reliable, and on-site power generation which is an added capital and running cost on the part of the operator, but with higher reliability. If power is generated onsite, more flexibility is available in terms of voltage selection and possibly motor start up options. If power supply is from an utility company, the pipeline operating company pays the bill. This has no capital investment on the part of the operator but the reliability of the supply is beyond the operator’s control, but depends on the utility company. The electric supply system reliability is generally a function of the voltage level of the supply. Transmission and sub-transmission voltage levels have a better record of reliability than distribution voltage levels. A dual feed from two different directions is more reliable than a single feed. In arranging for electric service, the utility and pipeline operator should have a clear understanding of probabilities and possible effects of unexpected weather conditions on the electric power supply.

3.4.8 Transmission loss

Electric-power transmission is the bulk transfer of electrical energy, from generators at power plant to substations and ultimately to consumers. This is distinct from electric power distribution, which is the local wiring between high-voltage substations and customers. Transmission loss occurs in transmission lines over long distance, which can be as high as 20% of the transmitted power, depending on the resistance of the transmission and the voltage of transmission. Electric power is transmitted over long distances with high-voltage lines because the transmission losses are much smaller than with low-voltage lines. All wires used for transmission from the generating station to the substation have a total resistance, R_{TT}

$$R_{TT} = \frac{\rho_{wire} \times L_{wire}}{A} \qquad (3.21)$$

Where ρ_{wire}, L_{wire}, and A are the resistivity, length and cross-sectional area of the transmission wire.

For a power demand of the substation

$$P_T = I\,V \qquad (3.22)$$

Equation 3.22 implies that the substation draws a current which is equal to

$$I = \frac{P_T}{V} \qquad (3.23)$$

Consequently, the higher the voltage, the smaller the current drawn for the same power. This analysis translates to a smaller transmission loss, which is a function of current drawn as shown in equation 3.24

$$P_{Loss} = I^2 R \qquad (3.24)$$

Alternatively, the transmission (power) loss can be expressed in terms of the power demand as

$$P_{Loss} = \frac{P_T^2 R}{V^2} \qquad (3.25)$$

For a fixed power demand and R as small as it can be by using very large cable, the transmission loss decreases strongly with increasing voltage.

3.4.9 Motor Life and Life Cycle Cost (LCC)

Horsepower ratings and life expectancy of electric motor supplied by manufacturers are well-established and are based on the following standardized operating conditions:

- ambient or surrounding air temperature less than 104°F (40°C)
- altitudes lower than 3,300 feet (about 1 km) above sea level (decreased air

  density reduces motor cooling)
- clean ventilation openings
- strict adherence to nameplate service factor limitations
- nameplate (rated) voltage supplied at motor terminals.

The main cause of reduced motor life is heat. When a motor is operated under the above conditions, the expected working life of the motor as stated on the motor nameplate is almost guaranteed and a continuous-duty electric motor will produced its rated horsepower output without overheating or damaging the insulation on the motor coil windings. Industrial motors can have a 20 to 30 year service life under proper operating conditions (Duhaime, 2012). Any operation at internal motor temperatures beyond nameplate ratings will reduce motor life. Electric motors can and will carry an overload. However, prolonged overload can shorten the motor life.

Many factors will affect the short term and long-term life of the motor. Some of the factors are;

1. Variations in Motor Loading. Operating the motor at above the rated load will generate more heat thereby leading to a higher temperature rise than one operating below the rated load. The higher the temperature rise, the higher the possibility of reduced motor life.

2. Load inertia. Each motor has specified standard inertia value. If a motor is accelerated with higher inertia during start-up, a higher heat build-up and motor stress ensues.

3. Frequent stops and starts. A motor starting draws six to seven times the full load current. This causes high short-term rotor copper losses and heat build-up. Frequent stops and starts in the long run cause the winding to fatigue due to high current loading and cyclic heat build-up.

4. Electrical supply voltage and frequency fluctuations. These fluctuations can lead to increased motor current with consequent winding temperature rise and increased electrical stress on motor windings which can lead to premature motor failure.

5. Operating altitude. Motors operating at high altitude beyond 1000 m (3300 ft) experience high temperature rises because at this altitude ambient air is less dense and dissipates less heat. This rise in temperature obviously has a negative effect on the motor life.

3.4.10 Motor Life Cycle Cost (LCC)

Life cycle cost is the systematic economic consideration of all whole life costs and benefits of the motor over a period of analysis or expected motor life while fulfilling the performance requirements. This analysis is recommended to assess the large cost items in the motor installation and operation project. LCC is the capital cost (purchase and installation), plus maintenance and operation costs (based on energy prices) over its life time. This computation was done bearing in mind the life expectancy of the motor.

$$Electricity\ cost = \left(\frac{Motor\ power\ rating}{conversion\ efficiency} + P_{Loss}\right) \times electricity\ tariff \qquad (2.26)$$

$$Transmission\ loss, P_{Loss} = \frac{P_T^2}{V^2}\left(\frac{\rho_{wire} \times L_{wire}}{A_{wire}}\right) \qquad (2.27)$$

$$LCC = \sum_{1}^{n\ years} (Electricity\ cost + capital\ cost + O\&M\ cost) \qquad (2.28)$$

3.4.11 Off-site Emission Analysis

Although electric motor drives do not produce on-site emission but a closer look and thought of the sources of electricity for running the motors will show that the use of electric drive is also responsible for pollution of the environment. Figure 3-25 shows the net electricity generation by fuel and it can be seen that about 40% of the net generation is produced by coal fired power plants. These plants generally have efficiencies less than the efficiency of gas turbine used for electricity generation. A coal fired power plant produces between 900 to 1400 kg of CO_2 for each megawatt-hour it generates, depending on the plant efficiency and coal type (Brun and Kurz, 2008b). The CO_2 emissions associated with electric motor drive is also directly influenced by the transmission and motor losses. This contributes to the power transmitted and consequently increases the emissions. For a typical compression station of about 7.5 MW, electric drive will lead to the emission of about 180000 kg of CO_2 per day.

Gas turbine drive in a compression station utilizes natural gas from the same pipeline as a fuel. It is well known that natural gas is a fossil fuel with the lowest carbon production footprint and a simple cycle gas turbine (operating at a nominal 35% efficiency) produces about 0.5 kg of CO_2 per hp per hour (Brun and Kurz, 2008a). For a 7.5 MW compression station using gas turbine as driver, less than 90000 kg of CO_2 is produced which is less than half of that produced when electric motor drive is used for the same compression station.

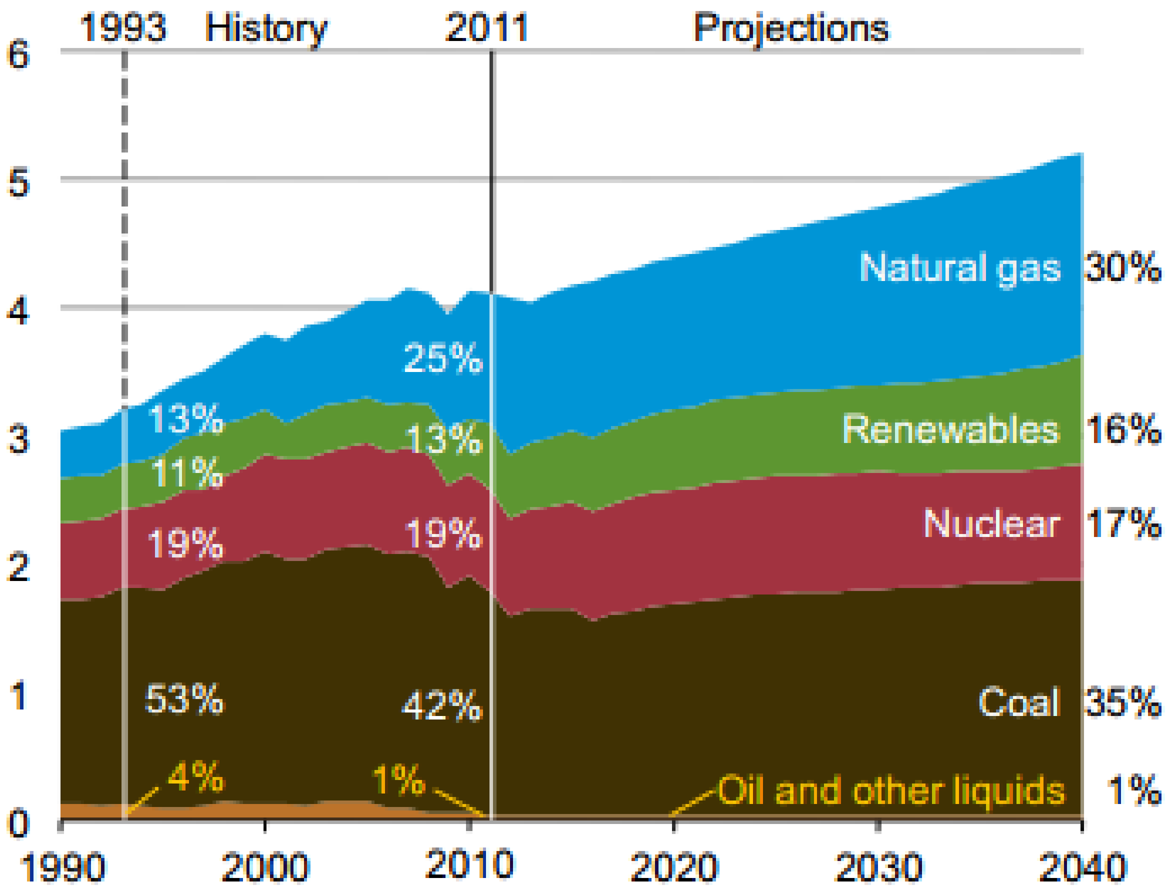

Figure 3-25: World net electricity generation by fuel (trillion kWh/year) (US EIA, 2012)

In order to deliver the required power to a 34 MW electric motor driven compressor station from a power station 50 km away, the power that needed to be transmitted should be above 115% of the required power and in some cases lower power is transmitted and then a step-up power transformer is used to step the power transmitted up to meet the power requirement of the compressor station.

$$P_{TRN} = \frac{34}{\eta} + Transmission\ loss \qquad (2.29)$$

$$P_{TRN} = \frac{34}{\eta} + \frac{P_T^2}{V^2}\left(\frac{\rho_{wire} \times L_{wire}}{A_{wire}}\right) \qquad (2.30)$$

Figure 3-26 shows the transmission loss variation with cable length. This generation of this huge loss obviously adds to the emission associated with electric drive. Figure 3-27 shows the total power transmitted taking into account the transmission losses.

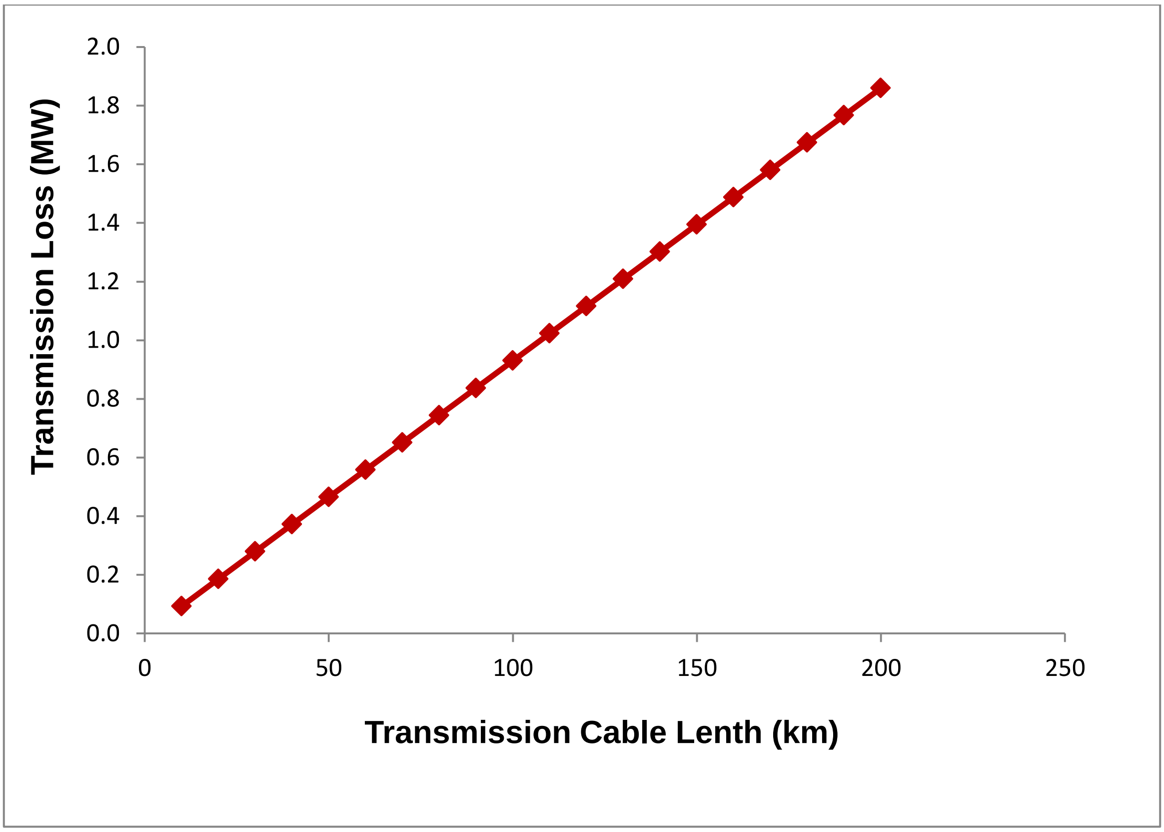

Figure 3-26: Transmission loss against cable length

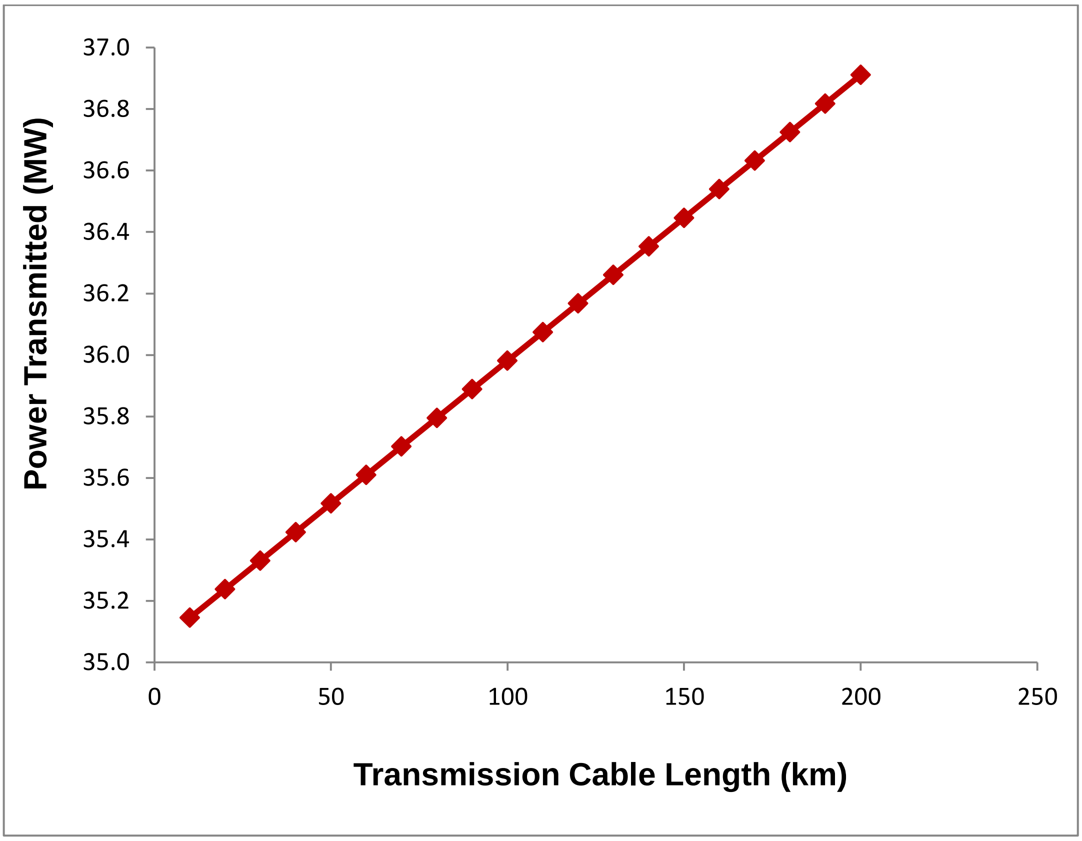

Figure 3-27: Power transmitted against length of transmission cable

For coal fired power plant supplying a 34 MW compressor station running for 8760 hours in a year will produce over 400 ton of CO_2. This is huge compared to the emission from gas turbine using natural gas as fuel.

Electric drive may at first seem better in terms of environmental pollution, but from the electricity generation today and in the near future, the use of electric motor contributes more to environmental pollution than does gas turbine. 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.

4 ECONOMICS OF CAPITAL PROJECTS

Natural gas pipeline projects are judged to be capital intensive and so the capital investment appraisal technique to assess the viability of any option consider is of utmost importance. The search for reliable techniques of project appraisal as a result of investors' continuous concern on project profitability dates back decades (Akalu, 2003). There exist in literature several appraisal techniques which can be employed by organizations in taking a decision which involves high capital investment, including natural gas pipeline transportation systems (Sillignakis, 2003). Choosing an appraisal technique depends not only on the cost of the investment, but also on the expected life of the project and type of company (Akalu, 2003).

4.1 Capital Investment (CI) Appraisal

A capital investment appraisal is a financial assessment which gives the profitability or otherwise of a capital project considering all the necessary economic criteria. Some techniques of capital investment appraisal among others which are employed in assessing the financial viability and profitability of projects are shown in Figure 4-1

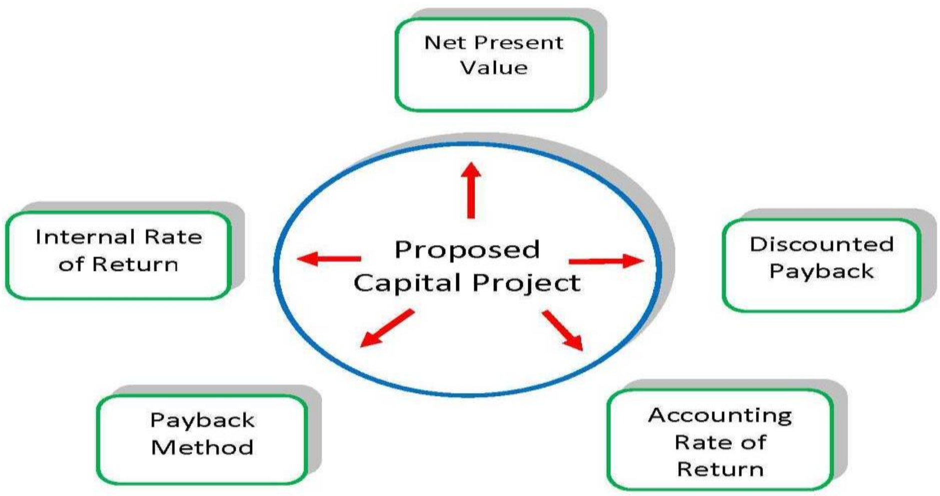

Figure 4-1: Techniques of Capital Investment Appraisal

4.1.1 Payback Technique

The payback method of appraisal is a technique which estimates the time needed for a project to recover the initial investment and further stand on its own. The recovery of an investment can be established by considering the net cash flow for certain period of time. This method does not take into account non-cash items, such as depreciation and gains on the sales of fixed assets.

4.1.2 Discounted Payback

This is similar to payback method but a better appraisal, since the time value of money is taken into account. A discounted net cash flow is considered in calculating the payback period. The procedure involves calculating future net cash flow and estimating an appropriate rate of interest and finally reducing the net cash flow to the present value by multiplying them by a discount factor.

4.1.3 Accounting Rate of Return (ARR)

This capital investment appraisal technique compares the profit that can be earned by a project to the amount of initial investment capital that would be required for the project. It is usually expressed as a percentage, i.e.;

$$ARR = \frac{Profit}{Capital\ employed} \times 100 \qquad 4.1$$

The decision using ARR is guided by higher rate of return, therefore projects with low rate of return are naturally not preferred. ARR does not take into account the present worth of the investment involved.

4.1.4 Net Present Value (NPV)

NPV is a CI appraisal which measures the cash in-flow, discounting it over the life span of the project and given the present worth. The main objective of investment appraisal the drive towards a positive NPV. The NPV method shows the importance of the cash received now over cash received in the future. The risk of waiting is presented as a discount which gives less money today than in the future. The advantage of getting money today and not waiting for tomorrow because of the uncertainty, is that the money can be invested and the difference made up over time. Basically, NPV is a mathematical calculation which involves calculating the annual net cash flows and discounting using an estimated appropriate rate of interest, thus giving the present values of the cash flow, which are added together to obtain the NPV. The project is financially viable and may be accepted if the NPV is positive but a negative NPV products that the project if embarked upon may not break-even. A project with higher NPV is preferred to one with a lower NPV when two projects have positive NPVs. NPV is considered to be highly acceptable method of CI appraisal. It does take into consideration the timing of the net cash flows, the project's profitability, and the return of the original investment. The NPV capital investment appraisal method was applied in the present study.

$$NPV = \sum_{i=3}^{T} \frac{Revenue\ stream}{(1+r)^i} - \sum_{i=0}^{T} \frac{capital\ \&\ operating\ cost}{(1+r)^i} \qquad (4.2)$$

4.1.5 Internal Rate of Return (IRR) Method

IRR is similar to the NPV method. However, instead of discounting the expected net cash flows by a predetermined rate of return, the IRR method is concerned with the rate of return which gives the total NPV equal to the total initial cost. IRR presents the efficiency of the capital investment. A project whose cost of capital investment is higher than IRR is considered not viable and is rejected.

4.2 Natural Gas Pipeline Economics

4.2.1 Introduction

There are three scenarios that represent the uses of pipeline. These are pipelines may be constructed

i. to transport natural gas for the owner of the pipeline,
ii. to sell gas to another company, or (iii)
iii. to transport some other company's gas.

The economics involved in the selection of pipe diameter, compressor station, driver option and other related facilities will vary slightly for each scenario. As an owner company transporting its own gas, minimal facilities will probably be built although all regulatory requirements will still have to be met. This is the scenario being considered in this research. Regulatory requirements impose strict guidance on type of facilities, and the cost that may be passed on to customers requesting gas transportation. Many factors must be taken into account in order to arrive at an accurate cost of service and consequently cost of transporting cubic meter of natural gas. The two main cost components are those related to the pipeline system and the cost related to the compressor system (Arsegianto et al., 2009). These costs are further split into capital cost, which comes only once at the beginning of the project, and operating and maintenance cost which varies every year through the production life of the project.

4.2.2 Capital Cost

Cost is an important element in the design, construction and operation of a natural gas pipeline system (Omonbude, 2009). The main contributors to pipeline construction costs include operating pressures, diameter, distance and very importantly, the terrain of the area through which the pipeline passes (Cornot-Gandolphe et al., 2003).

The capital cost of a pipeline project consists of the following major components:

- Pipeline
- Compressor stations
- Main valve stations
- Meter stations
- Pressure regulatory station
- SCADA and telecommunication
- Right of way acquisition
- Engineering and Construction management

4.2.2.1 Pipeline system cost analysis

The pipeline cost consists of those costs associated with the pipe material, coating, pipe fittings, and the actual installation or labour cost. Pipeline cost is an important cost parameter that is considered in natural gas pipeline economics. The cost of pipe is a function of diameter, thickness and the total length of pipe. Pipe diameter and thickness are used to establish the tonnage of pipe per unit length. Using Equation 4.1 for pipe weight, the cost of pipe required for a given pipeline length is found from equation 4.3.

$$PMC = 0.024t(D - t)LC_T \tag{4.3}$$

Where C_T is the pipe material cost per tonne ($/ton), D, t and L are diameter, thickness and total length of the pipe respectively.

In the present study, the estimate of pipe material cost per ton (C_T) was obtained from manufacturers of steel pipes and all other costs associated with pipeline are estimated from available literature. The labour cost to install the pipeline can be represented in dollars per unit length of pipe. This amount will depend on whether the pipeline is installed in open country, fields, or city streets. Such figures are generally obtained from contractors who will take into consideration the difficulty of trenching, installing pipe, and back-filling in the area of construction. A generalized estimate of pipeline construction cost is quite difficult because this depends on the location. Laying a pipeline through a rural area may cost far less than through a dense urban area (Parker, 2004).

For estimation purposes, there is a wealth of historical data available for construction cost for various pipe sizes. Sometimes the pipe installation cost is expressed in terms of dollars per in. diameter per mi of pipe.

Table 4-1: Typical Pipeline Installation Costs (Menon, 2005)

Pipe Diameter (inch)	Average Cost ($/in-dia/mi)
8	18,000.00
10	20,000.00
12	22,000.00
16	14,900.00
20	20,100.00
24	33,950.00
30	34,600.00
36	40,750.00

4.2.2.2 Compressor station cost analysis

The economic success of a gas compression operation depends to a large extent on the operation of the compressor stations. First cost (capital cost), operating cost (especially fuel cost), life cycle cost and emissions are important criteria that need to be considered for the economic success of the compression station. The selection of driver type and type of compressor certainly has an impact on the cost, fuel consumption, emissions and the general economics of the compression system. In this study a centrifugal compressor was considered and two drive options were studied viz; gas turbine and electric motor drive.

The capital cost for this project was not only based on the first cost of compressors and prime movers (drivers), but also on necessary systems required for the proper operation of the compressor station. These include coolers, valves, instruments and filters. The driver selection must follow basic criteria of the compression station, which is that the power demand of the compressor must be met at site condition (especially worst ambient temperature scenario) and not at ISO conditions.

4.2.3 Operating/Energy Cost

Once the pipeline, the compressor stations, and ancillary facilities are constructed and the pipeline is put into operation, there will be annual operating costs over the useful life of the pipeline, which is 30 years for the present research. Operating cost is significantly controlled by fuel cost. Efficiency and operating range are two performance parameters of the compressor and its driver that are important for the economic evaluation of the compressor station. Efficiency is directly related to the cost of fuel consumed in order to bring a certain amount of gas from suction pressure to a pre-determined discharge pressure. For a cost effective system, it requires a high isentropic efficiency of the compressor, high thermal efficiency of a gas turbine drive option and a high conversion and mechanical efficiency of the electric drive option. Operating range describes the range of possible operating conditions in terms of flow and head at an acceptable efficiency, within the power capability of the driver. For the gas turbine drive option, the fuel consumption over the varying ambient conditions of the compressor station was obtained from the off-design simulation results using TURBOMATCH. Part of the results for fuel gas turbine consumption with respect to ambient temperature variation has been presented in chapter three of this thesis. The energy cost associated with the use of electric drive option for the same pipeline was obtained considering the conversion efficiency which determines the electric power supplied to meet the power requirement of the compressor.

$$\mathrm{Energy}_{\mathrm{FUEL}} = \mathrm{V}_{\mathrm{FUEL}} \times \mathrm{CV}_{\mathrm{FUEL}} \times \mathrm{VC}_{\mathrm{FACTOR}} \tag{4.4}$$

The annual fuel energy used can be computed from equation 4.5

$$\frac{\mathrm{E}_{\mathrm{FUEL}}}{\mathrm{annum}} = \mathrm{E}_{\mathrm{FUEL}} \times \text{annual operating time (sec)} \tag{4.5}$$

Energy cost can then be calculated from the present price of gas as

$$\text{Energy}_{\text{COST}} = \text{Annual Energy} \times \text{gas price} \qquad (4.6)$$

4.2.4 Maintenance cost

Maintenance cost is the cost incurred in order to the pipeline system in good working condition to meet the growing demand for natural gas. It is an important parameter in the economic analysis of the pipeline project. It is also a cost that contributes to the determination of the economic viability of a project by establishing the net present value and the life cycle cost of the project. Maintenance costs includes the parts and labour to keep the equipment running at or above a certain power level. This include routine maintenance (like change of lube oil and spark plugs in gas engines) and overhauls. Maintenance events can be scheduled or condition based. The maintenance cost calculation takes account of the degradation of the power plant. Degradation factors from OEM's confidential data were used in the computation of the maintenance cost over the working life of the plant. Maintenance affects availability, which in turn affects the economics of the project. Some maintenance events, usually unscheduled, require the complete shutdown of the plant, while others, usually scheduled do not always require complete shutdown. When a plant is shut down completely, its availability is affected, which impacts on the income which could have been made during the period. Maintenance costs may be kept to the bare minimum by maintaining a schedule maintenance programme, as suggested by the OEM. Unscheduled and improper maintenance negatively affects the availability due to more rapid performance degradation and a higher chance of unplanned shutdowns.

4.2.5 Economic pipe size

Transporting a particular throughput of gas through a pipeline can be done using several pipe sizes, but one of the pipe sizes will give the lowest transportation cost. The pipe size that gives the lowest transportation cost is the economic pipe size for the particular throughput. This may vary with throughput, but a range of throughput will definitely have an economic pipe size. The computation of economic pipe size considers the power requirement as a result of varying pipe sizes and the consequent cost implications in the form of capital and operating costs. In a range of pipe sizes, the pressure drop is maximum with the smallest size and consequently requires more compression power to maintain the throughput as well as pressure. Although the smallest pipe size would obviously have least cost, the capital and operating costs of the require gas turbine may not make it the economic pipe size. This is obviously a conflicting condition which requires striking a balance to obtain a pipe size which ultimately gives the lowest transportation cost. An analysis of pipe cost and operating cost over a range of pipe sizes is presented in chapter five to illustrate this conflicting condition.

5 TECHNO-ECONOMIC MODULE DEVELOPMENT

This chapter considers the development of techno-economics of each of the modules. It presents the results obtained from each of the modules which were developed in FORTRAN codes.

5.1 Gas Properties

Gas properties are important parameters in the analysis of gas flow, as well as the compression process. The compressibility, gas composition and specific gravity affects the energy required to compress certain volume of the gas as well as pressure drops in pipeline. The composition of the natural gas is presented in Table 5-1.

Table 5-1: Composition of natural gas

Components	% Composition	Molecular Weight	%Weight
Methane (CH_4)	90	16.04	14.44
Ethane (C_2H_6)	3	30.07	0.90
Propane (C_3H_8)	3	44.09	1.32
Butane (C_4H_{10})	2	58.12	1.16
Carbon Dioxide (CO_2)	1	44.01	0.44
Nitrogen (N_2)	1	28.01	0.28
		Molecular weight of gas =	18.54

5.1.1 Specific gravity

The specific gravity of the natural gas, which is the measure of how heavy the gas is compared to air at a particular temperature, can be calculated thus;

$$G = \frac{M_{Wg}}{M_{air}} = \frac{18.54}{29} = 0.64 \qquad (5.1)$$

5.2 Pipeline Module

In developing the pipeline module an existing pipeline was used as a baseline. Figure 5-1 shows the pipeline route profile from Sarir gas field to Tobruk city gate.

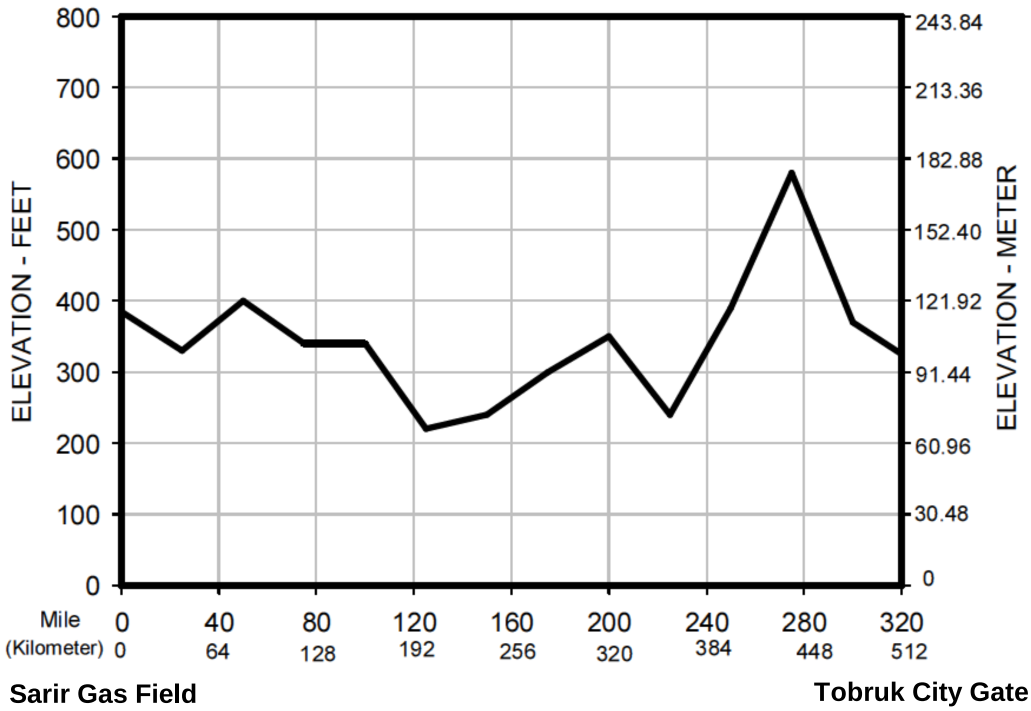

Figure 5-1: Pipeline Route Profile

The pipeline route has variation in elevation along the pipe length. It becomes necessary to obtain the equivalent pipe length, which takes into account the varying elevation. The equivalent length of the pipeline is obtained using equation 5.2.

$$L_e = L_i j_i \qquad (5.2)$$

where L_i is the length of each segment of the entire pipe length. j_i is calculated for each segment of pipe and can be expressed as

$$j = \frac{e^s - 1}{s} \qquad (5.3)$$

The elevation adjustment parameter s which is expressed in equation 5.4 depends on gas gravity, gas compressibility factor, the gas temperature and the elevation difference.

$$s = 0.0684G\left(\frac{H_2 - H_1}{T_f Z}\right) \qquad (5.4)$$

where H_1 and H_2 are upstream and downstream elevations respectively.
The equivalent length of pipeline, considering the route and pipe segments, can be expressed as 5.5.

$$L_e = j_1L_1 + j_2L_2 + j_3L_3 + \cdots \ldots\ldots\ldots\ldots (5.5)$$

The pipeline module was developed in Fortran code in order to make it an integral part of a series of modules which are Fortran based, the important of this, being the gas turbine performance code (TURBOMATCH). The module analyses the flow of natural gas in the pipeline. The case study considered in this research falls into the class of large diameter pipeline, high pressure, and high flow rate. Consequently the general gas equation 5.6 was used to analyse the natural gas flow, considering the equivalent pipeline length and obtaining transmission factor as a function of the friction factor. Equation 5.6 represents the equation to technically determine the pipe size necessary for any particular throughput.

$$D = \left[\frac{Q\left(\frac{P_b}{T_b}\right)}{5.747 \times 10^{-4}F}\left(\frac{P_1^2 - e^s P_2^2}{GT_f L_e Z}\right)^{-\frac{1}{2}}\right]^{0.4} \qquad (5.6)$$

where the transmission factor F is related to the friction factor f by

$$F = \frac{2}{\sqrt{f}} \qquad (5.7)$$

The friction factor can be obtained from the Colebrook-White equation, which presents the relationship between the friction factor and the Reynolds number, pipe roughness and inside diameter of pipe. Equation 5.8 is the Colebrook equation for calculating the friction factor in gas pipeline in turbulent flow.

$$\frac{1}{\sqrt{f}} = -2Log_{10}\left(\frac{e}{3.7D} + \frac{2.51}{Re\sqrt{f}}\right) \qquad (5.8)$$

where e is the absolute pipe roughness.

The pipes considered in this research are smooth pipes, and therefore the Colebrook equation 5.8 reduces to

$$\frac{1}{\sqrt{f}} = -2Log_{10}\left(\frac{2.51}{Re\sqrt{f}}\right) \qquad (5.9)$$

Equation 5.9 is an implicit equation in f because f appears on both sides, the solution is sorted by trial-and-error approach. A FORTRAN programme was developed to solve the iteration problem of equation 5.9 until convergence is obtained.

5.2.1 Compressibility Factor

Compressibility factor is a measure of how close a real gas is to an ideal gas. It is a function of gas gravity, gas temperature and gas pressure. A fairly simple equation for quickly calculating the compressibility factor, when the gas gravity, temperature and pressure are known is the California Natural Gas Association (CNGA) method and this equation is given as;

$$Z = \frac{1}{\left[1 + \left(\frac{344400\ P_{avg} \times 10^{1.785G}}{T_f^{3.825}}\right)\right]} \qquad (5.10)$$

5.2.2 Pipe Analysis

Transporting natural gas through pipeline over long distances is adjudged to be cost effective and this makes the procedure attractive. Pipe analysis is an important aspect to ensure safety and minimize cost. Pipe diameter, wall thickness and material of construction are parameters which must be properly analysed in any techno-economic model involving natural gas pipeline transportation.

The pressure required to transport a given volume of gas through a pipeline is one of the factors which controls pipe selection. The pipe internal pressure is a parameter that can cause permanent deformation if allowed to reach or exceed the yield strength of the pipe. Obviously, the pipe should have sufficient strength to handle the internal pressure safely. The natural gas pipeline may also be subjected to external pressure, where the pipes are buried owing to the weight of the soil above due to load transmitted above the soil. The deeper the pipe is buried, the higher will be the soil load on the pipe, and the lower will be the pressure transmitted to the pipe due to vehicles above the ground. The effect of internal pressure is more than that of the external pressure in some cases involving buried pipelines transporting gas; therefore, the internal pressure dictates the necessary minimum pipe wall thickness.

The minimum wall thickness required to withstand the internal pressure in a gas pipeline will depend upon the pressure, pipe diameter and pipe material. The larger the pressure or diameter, the larger the wall thickness required.

5.2.3 Pipe Tonnage

Pipe tonnage is an important parameter used frequently in natural gas pipeline economics. This guides the calculation of pipe total material cost and consequently leads to the establishment of the life cycle cost. The weight per meter of a pipe is given by equation 5.11 (Menon, 2005).

$$W_p = 0.0246t(D - t) \tag{5.11}$$

Where t, D and $W_p$are the thickness (mm), diameter (mm) and weight per unit length (kg/m) of pipe respectively.

5.2.4 Main Valve Station (MVS)

Valves are important part of a pipeline system, they are installed on pipelines and piping systems to isolate sections of piping for maintenance, to direct the fluid from one location to another, to shutdown flow through pipe sections, and to protect pipe and prevent loss of fluid in the event of rupture. Design codes and regulatory requirements dictates that sections of be isolated by installing mainline block valves at certain fixed spacing on long distance pipelines transporting natural gas and other compressible fluids. The spacing of these valves depends upon class location which in turn depends on the population density around the pipeline.

Table 5-2: Maximum Spacing between Main Valves (Menon, 2005)

Class Location	Valve Spacing
1	20 miles
2	15 miles
3	10 miles
4	5 miles

The baseline pipeline used in this research is a class 1 location and consequently main valve stations were installed at every 32 km along the pipeline. This is an added cost to the pipeline cost.

5.2.5 Meter Stations and Regulators (MSR)

Accounting for the amount of gas transported from one point to another along a gas pipeline makes gas flow measurement an important aspect of pipeline system. The flow rate of the gas has to be measured at a number of locations for the purpose of monitoring the performance of the pipeline system and more particularly at places where custody transfer takes place. Depending on the purpose for metering, whether for performance monitoring or for sales, the measuring techniques used may vary according to the accuracy demanded (Mokhatab and Raymand, 2009). Meter and regulator stations are installed about 40 km apart on a long pipeline.

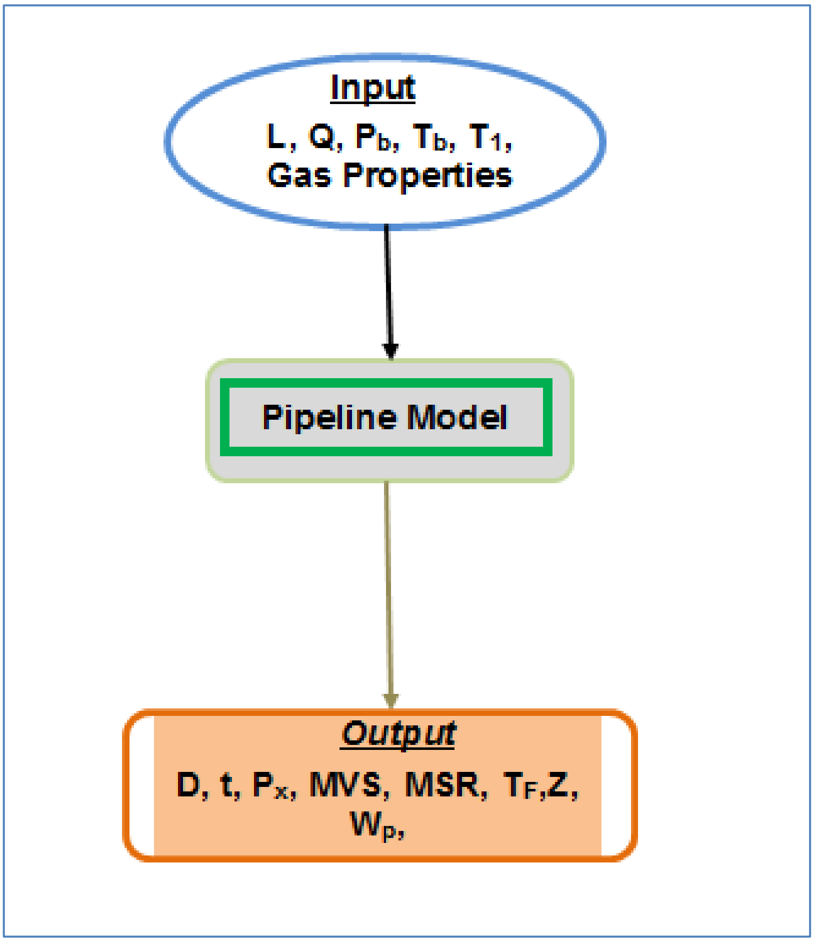

Figure 5-2: Pipeline Model

The pipeline model calculates for a given throughput and gas properties, the required pipe size and automatically picks the next nominal pipe size. The module is made robust enough to calculate for a particular throughput, varying pipe sizes the pipeline inlet necessary to maintain a constant pipeline discharge pressure. The pipeline model also determines for a specific pipe size and varying throughput the pipeline inlet pressure, to maintain constant discharge pressure at the city gate. Establishing these variations is important in order to know the trend of drive power required (calculated in the compressor station module), which consequently affects the economics of the system. Figure 5-3 to Figure 5-5 shows the result of the pipeline module. Other parameters such as pressure drop along the pipeline, number of MVS and MSR, gas flowing temperature, pipe weight and compressibility of gas are computed and made inputs to other models.

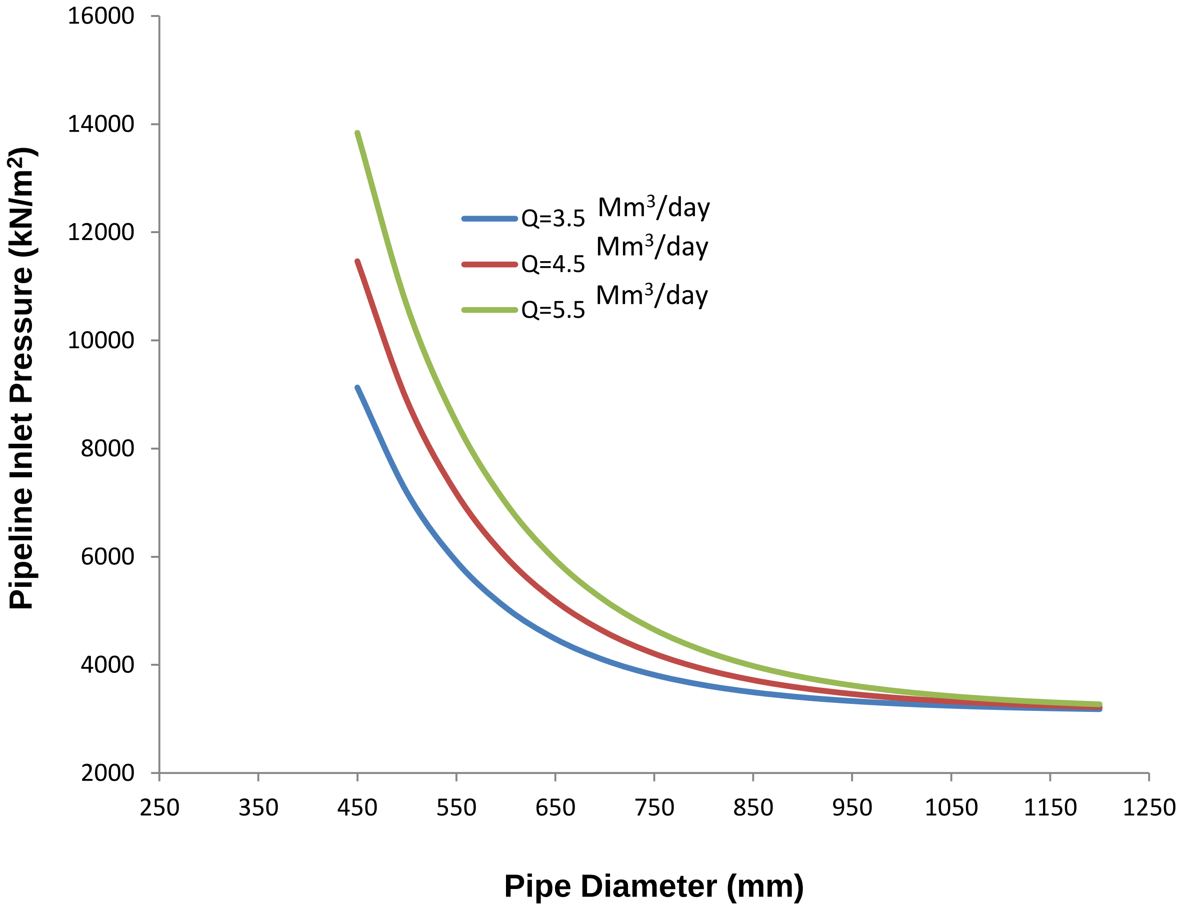

Figure 5-3: Pipeline inlet pressure variation with pipe diameter

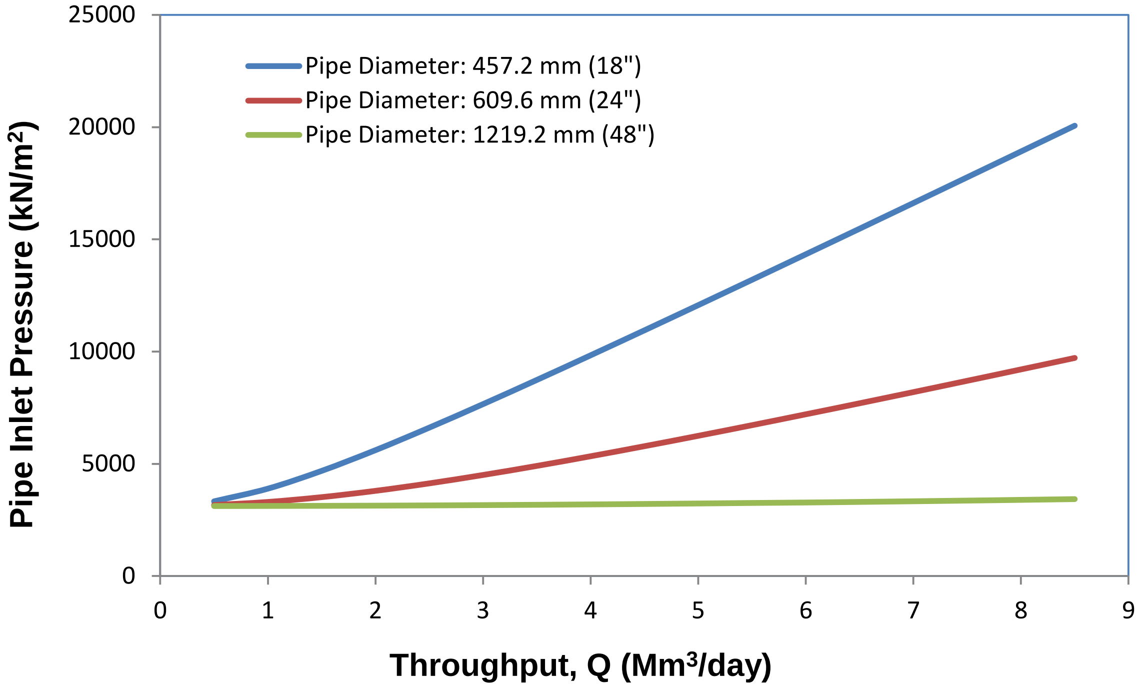

Figure 5-4: Pipe inlet pressure variation with gas throughput

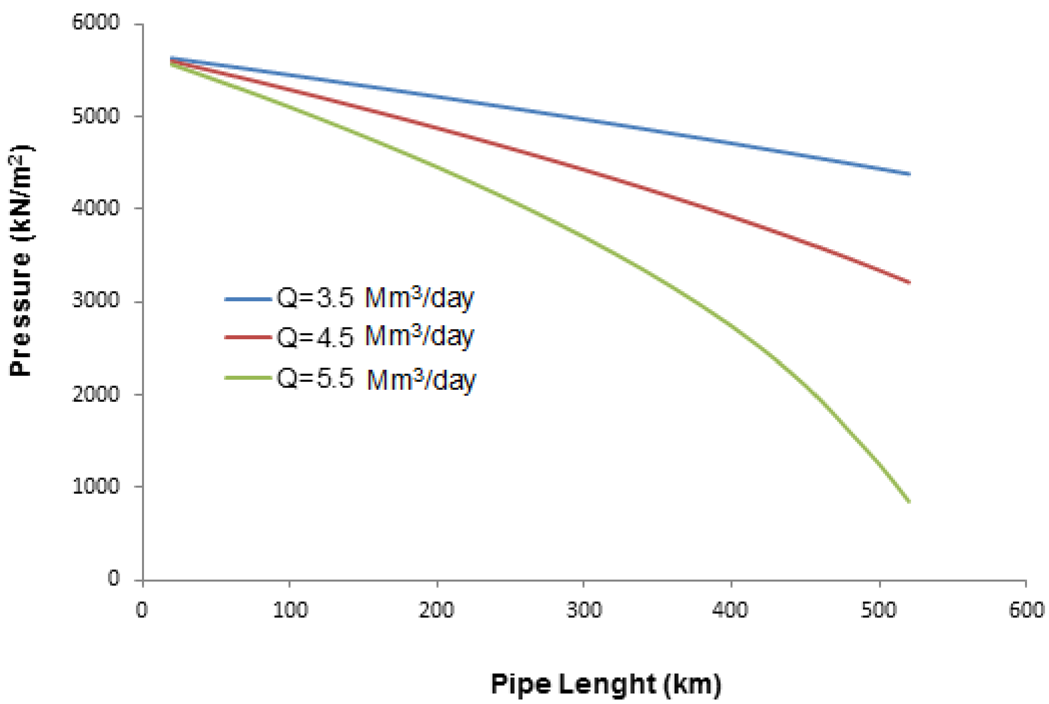

Figure 5-5: Pressure gradient along the natural gas pipeline

5.2.6 Discussion of Pipeline Module Results

Figure 5-3 presents the effect of varying pipe diameter on the pipeline inlet pressure for different throughput. This was done maintaining the discharge pressure at the city gate at 3000 kPa (30 bar). The pipeline inlet pressure is the compressor station discharge pressure. The inlet pressure rises sharply from 5.9 x 10^3 kPa for a throughput of 4.5 Mm^3/day through a pipe diameter of 600 mm (NPS 24) to 11.5 x 10^4 kPa for the same throughput through a 450 mm (NPS 18) pipe size. This is because of the higher pressure drop in smaller pipes, as a result of increase in the resistance to flow which is due to high frictional effect. This is an indication of pipe under-sizing, the effect of which is translated to rise in the power requirement for compression as presented by the compressor station module. Beyond the pipe size of 750 mm (NPS 30), the pipe inlet pressure to maintain a delivery pressure of 3000 kPa at the city gate gradually reduces and the effect of pipe sizes on the inlet pressure fizzles out as the pipe diameter becomes larger. It can be seen from Figure 5-3 that the effect of pipe sizes on inlet pressure has the same pattern for the throughput of 3.5, 4.5, 5.5 Mm^3/day and this effect diminishes with larger pipe sizes. The effect of pipe sizes on the pipe inlet pressure diminishes with large pipe sizes because the pressure drop is minimal.

For different pipe sizes the effect of throughput on the inlet pressure required to maintain a discharge pressure of 3000 kPa (30 bar) at the city gate in Tobruk is presented in Figure 5-3. For a pipe size of 457.2 mm (18”), the pipeline inlet pressure necessary to maintain a constant discharge pressure of 30 bar is 8735.9 kN/m^2 for a throughput of 3.5 Mm^3/day and is 17762.8 kN/m^2 for a throughput of 7.5 Mm^3/day; this gives a about 103.3 % rise. The percentage rise for a 609.6 mm (24”) diameter pipe is 77.6 % in the same flow regime. For a 1219.2 mm (48”) the inlet pipeline pressure is 3175.7 kPa for a throughput of 3.5 Mm^3/day and 3373.4 kPa for a throughput of 7.5 Mm^3/day, this gives a percentage increase of 6.2 %. This implies that for any particular throughput, the pipeline inlet pressure which controls the compression power is lower with large pipe sizes with consequent lower driver power. This is because the difference of squares of pressure indicating pressure drop reduces with increased pipe size. An economic saving is made in compression but with an increase in pipe material cost. Striking a compromise between these two important cost parameters will lead to a cost effective pipeline system. The economic module presented in the latter part of this thesis deals with this issue.

Figure 5-5 shows the pressure gradient along the 512 km natural gas pipeline. Maintaining the inlet pressure at 5900 kPa, the discharge pressure at the city gate 512 km away varies with throughput. For a throughput of 5.5 Mm^3/day, the discharge pressure is 850 kPa and for a throughput of 3.5 Mm^3/day the discharge at the same city gate is 4383.4 kPa. This confirms the fact that an increase in throughput must be matched with a corresponding increase in pipe inlet pressure, and consequently compressor drive power to maintain a constant discharge pressure.

The internal pressure in a pipe as a result of the flowing gas causes the pipe wall to be stressed, and if allowed to reach the yield strength of the pipe material, it could cause permanent deformation of the pipe and ultimate failure. Although external pressure from the soil above buried gas pipeline contribute to the total pressure, the internal pressure is more significant and the minimum wall thickness required to withstand the internal pressure in a gas pipeline depend upon the pressure, pipe diameter, and pipe material. The larger the pressure or diameter, the larger would be the wall thickness required. Higher strength steel pipes will require less wall thickness to withstand the given pressure compared to low-strength materials

When the same flow that enters a pipe is the same with the flow at discharge, a constant pipe size and thickness may be used. However, if there are injections or deliveries which may cause the gas pressure and throughput to vary considerably, then pipes of same nominal diameters but different thickness along the pipe segment where increase pressure is expected may be used. The minimum wall thickness required to withstand the internal pressure in a gas pipeline will depend upon the pressure, pipe diameter, and pipe material.

5.3 Compressor Station

The compressor station is the heart of the natural gas pipeline system and its functionality is very important to the success of the transportation system. Compressor stations are installed on gas pipeline to provide the pressure needed to transport gas from one location to another. The two important units in the compressor stations which concerns this research are the compressors and their drive units. Maximum Allowable Operating Pressure (MAOP) of the pipe controls the maximum discharge pressure from a compressor station necessary to transport a certain volume of natural gas through a long-distance pipe. This invariable affects the number of compressor stations along a gas pipeline. The locations and pressure at which these compressor stations operate are determined by the allowable pipe pressures, power available, environmental and geotechnical factors (Menon, 2005).

5.4 Pipeline Compressors in Series and Parallel

The Two main arrangements of compressor units which are possible in compressor stations are the series and parallel configurations. In series operation of compressors, each unit compresses the same amount of gas but may be at different pressure ratio. The overall pressure ratio is achieved in stages of compression. Figure 5-6 shows the series arrangement of compressor units.

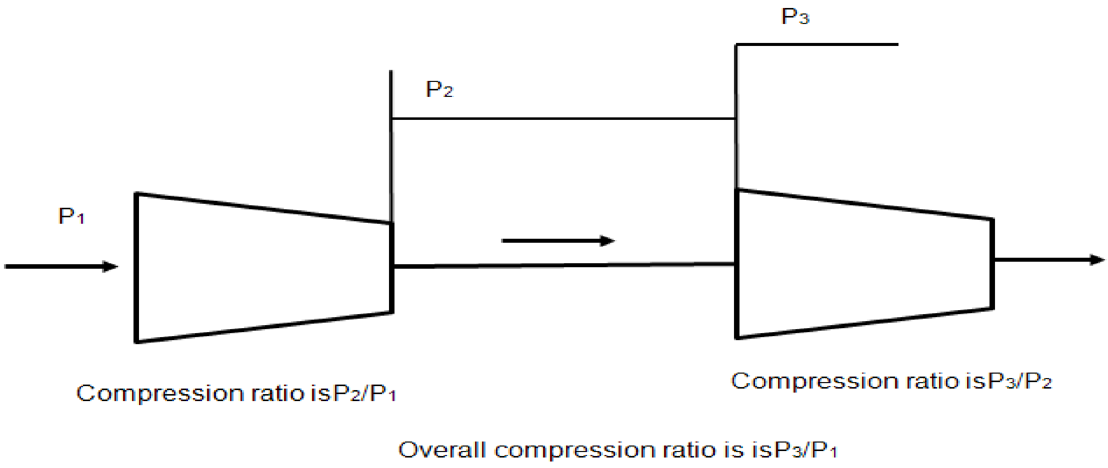

Figure 5-6: Compressors in series

In some compression stages, intercooling may be necessary between two stages because of the temperature of the compressed gas which follows equation 5.12.

$$T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \qquad (5.12)$$

High gas temperatures are not desirable, since the throughput of the gas decreases with flow temperature.

In parallel arrangements of compressor units, the compression ratios are usually the same, but this arrangement can handle large volume of gas. Figure 5-7 shows the parallel arrangement of the compressor units. Unlike in the compressors in series arrangement, the temperature of the discharged gas from parallel compressors will not be high, since it has not gone to multiple compression stages. The gas temperature on the discharge side of each parallel compressors will be the same as that of a single compressor, with the same compression ratio. An after cooler is required at the compressor discharge before entering the pipeline to cool the temperature of the gas in order to achieve efficient gas transportation, and also to operate at temperatures not exceeding the limits of the pipe coating material. The pipe coating materials generally requires gas temperature not to exceed 60^{o}C to 65^{o}C (Menon, 2005). There is a considerable increase in discharge gas temperature with increase in compression ratio. In order to achieve high compression ratio, it is necessary to have multi-compression stages, in order that each stage of compression does not exceed the acceptable range of compression ratio. In general, if N_c compressors are installed in series to achieve a required compression ratio r, each of the compressor stage will operate at a compression ratio of

$$r = (r_i)^{\frac{1}{N_c}} \qquad (5.13)$$

Where r is compression ratio, r_i is the overall compression ratio and N_c is the number of compressors in series. Power requirements is minimized by using identical compressors in series to provide the overall pressure.

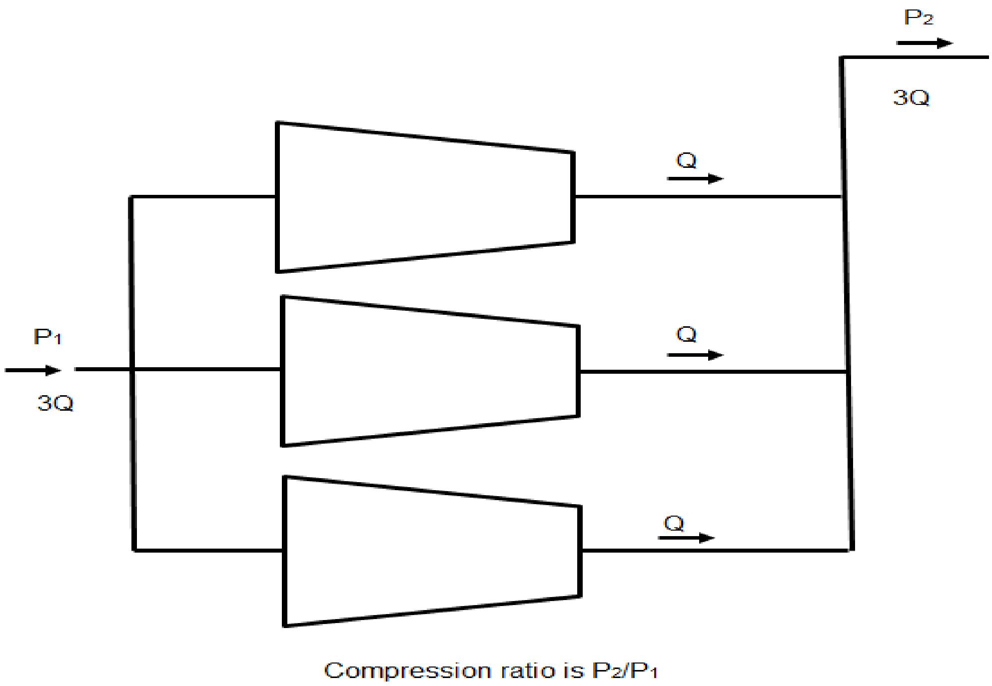

Figure 5-7: Compressors in parallel

The question of which arrangement is better has attracted the attention of many authors (Santos, 1997; Santos, 2000; Kurz et al., 2003; Ohanian and Kurz, 2002), who have considered the advantages and disadvantages of series, parallel or series-parallel compressor unit arrangements in compressor stations. Dos Santos, (2004) in his techno-economic analysis of the two arrangements, concluded that the series arrangement of the compressors gives a better result than the parallel arrangement when there is no consideration for standby units, and parallel units provides better results than series, when stand by units is a requirement and more flexibility is desirable.

5.5 Compressor Station Module

The continuous flow of natural gas in a pipeline is made possible by the help of compressor stations which boost the pressure at a predetermined interval along a pipeline. These stations are generally made up of basic components such as compressor and driver units, scrubber/filters, cooling facilities, emergency shutdown systems, and an on-site computerized flow control-Supervisory Control and Data Acquisition (SCADA) and dispatch system that maintains the operational integrity of the station (Carter, 1998). This module computes the required compressor power necessary to handle a certain flow through a specified pressure ratio.

$$H_P = ZRT_1 \frac{n}{n-1}\left(r_P^{\frac{n-1}{n}} - 1\right) \qquad (5.14)$$

The gas power is computed from the general form of the thermodynamic head equation for a polytrophic process. The shaft power is subsequently computed by considering mechanical, leakage and coupling losses. The drive power required by the compressor is

$$P_D = \frac{mH_P}{\eta_C} \qquad 5.15$$

where η_C is the compressor efficiency and m is the mass flow rate of gas.

The module is made sufficiently robust to compute the number of main valve stations, meter stations and regulators all depending on the length of pipeline, and also establish the optimum compressor station position along a pipeline using a genetic algorithm toolbox in MATLAB. This module receives input such as pipeline inlet pressure, which is necessarily the compressor station discharge pressure from the pipeline model. Pipeline inlet pressure varies with pipe diameter for constant throughput and city gate discharge pressure. The module uses this established information to determine the drive power required for each case. It determines the effect of varying throughput and constant pipe sizes on the drive power required.

5.6 Hydraulic Balance

Hydraulically balanced pipeline is adopted for the analysis of the number of compressor stations. This involves setting the pressures and flow rate to be the same since there are no intermediate injections or deliveries along the pipeline and this consequently implies that every compressor station will require the same amount of drive power. This has a great advantage since all the compression equipment is identical for all the compression stations, with all adding the same amount of energy to the flowing gas. This arrangement will reduce the inventory of spare parts and minimize maintenance. Hydraulically balanced compressor stations requires less total drive power than if the stations were not located for hydraulic balance.

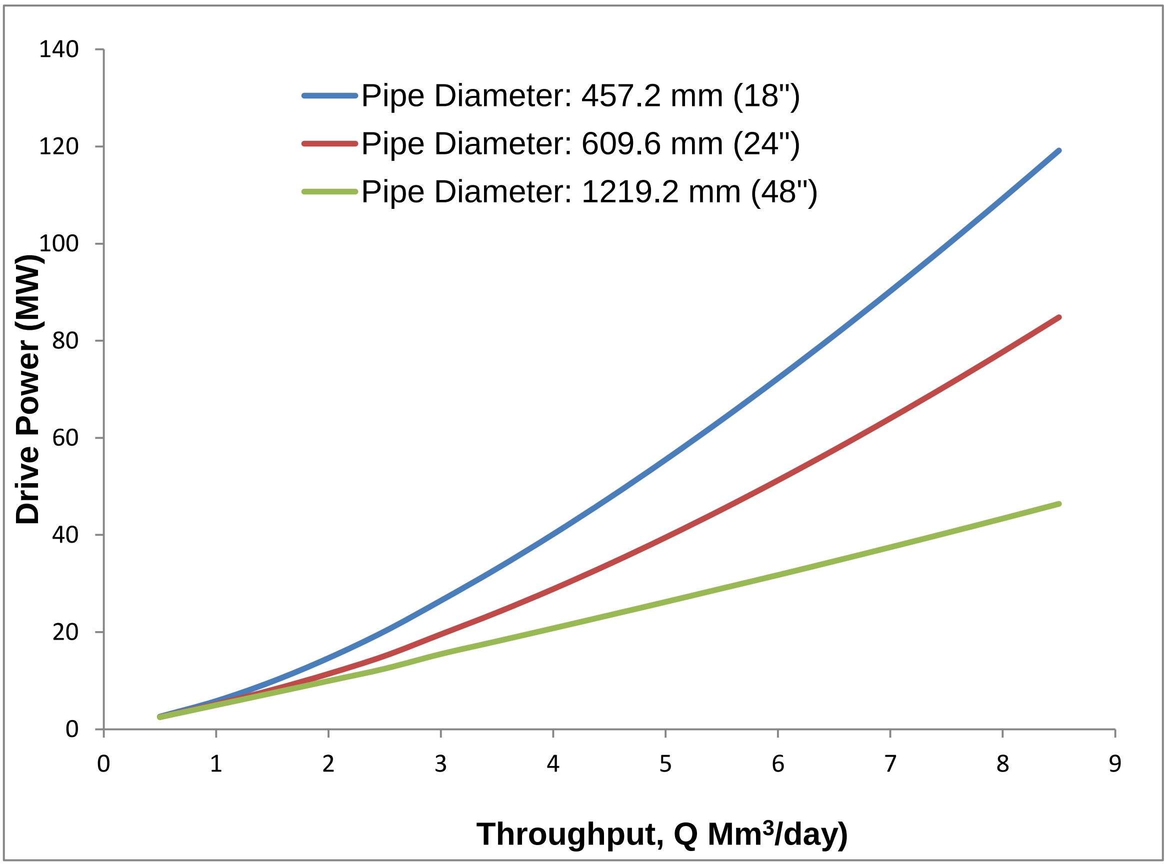

Figure 5-8: Effect of Throughput on compressor drive power

5.7 Discussion of Compressor station Model Results

Figure 5-8 shows the variation of drive power requirement with throughput for varying pipe sizes. It can be seen that the increase in drive power for pipe diameter 457.2 mm (18") is higher than for 609.6 mm (24") and 1219.2 mm (48"). For a 457.2 mm pipe, the drive power required for 2.5 Mm^3/day is 20.2 MW and 47.7 MW for a throughput of 4.5 Mm^3/day. This amounts to 136.1% increase power required for about 80% increase in throughput. For a pipe size of 1219.2 mm (48"), 80% increase in throughput leads to about 88.4% increase in drive power. This implies that at high throughput the drive power required is lower in large pipe sizes and higher in small pipe diameters. This is because resistance to flow due frictional effects decreases with increase diameter and vice versa (Schroeder, 2001).

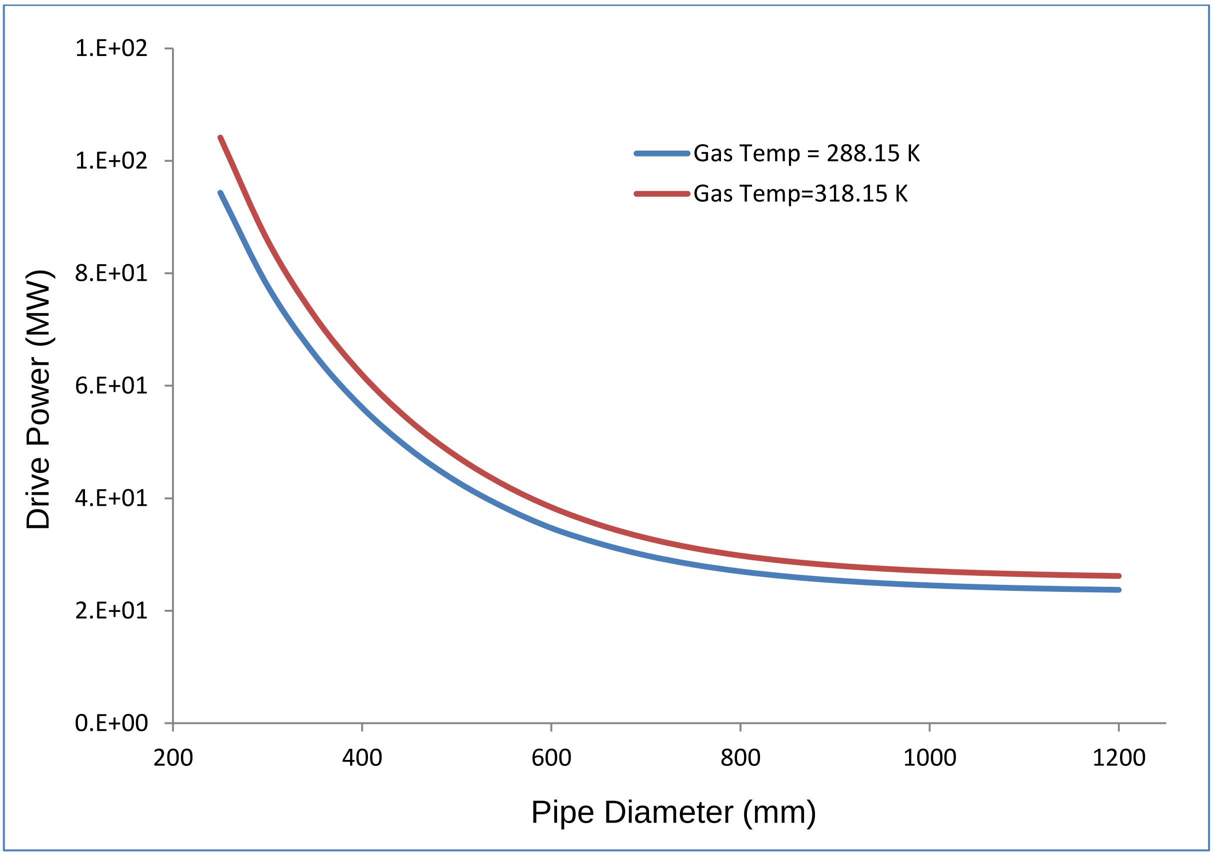

Figure 5-9: Drive power variation with pipe diameter

The effect of varying pipe sizes and gas temperature at inlet into the compressor station on drive power is presented in Figure 5-9. The drive power increases with increased gas inlet temperature to maintain a constant throughput. The efficiency of the overall compression process is reduced; an improvement in the efficiency of compression can be achieved by cooling the gas before a stage of compression (Osiadacz and Chaczykowski, 2001). A drive power of 48.7 MW is required to compress natural gas of 4.5 Mm^3/day through a pipe size of 457.2 mm (18”). A gradual reduction in drive power is noticed with increase pipe size until a pipe size above 600 mm (NPS 24) where the drive power seems constant. This gives an estimate of the economic pipe size of 609.6 (24”) for a throughput of 4.5 Mm^3/day. A conclusion on the economic pipe size for a particular throughput can only be established through the results of the economic module which will be presented in later part of this chapter.

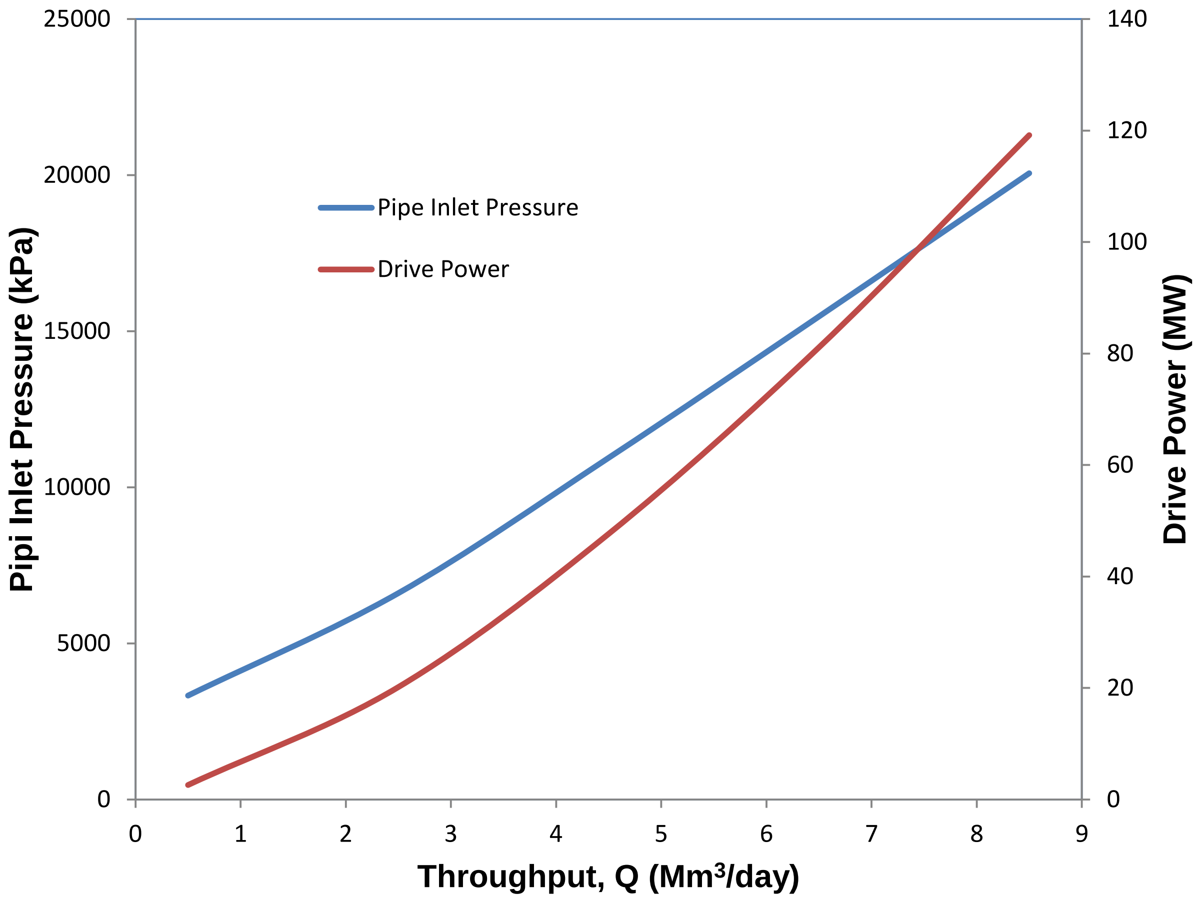

Figure 5-10: Pipe inlet pressure and drive power variation with throughput for Pipe Diameter = 457.2 mm (18”)

Figure 5-10 presents the effects of varying throughput on drive power and consequently on pipe inlet pressure for a pipe size of 457.2 mm. For a pipe size of 457.2 mm, a throughput of 2.5 Mm^3/day and inlet pipeline pressure of 6610.4 kPa in order to maintain a discharge pressure at the city gate of 3000 kPa, a drive power of 20.2 MW is required. And for a throughput of 4.5 Mm^3/day and inlet pipeline pressure of 10945.7 kPa to maintain a discharge pressure of

3000.0 kPa a drive power of 47.6 MW is required. This means for 80 % rise in throughput through a 304.8 mm pipe, a 78% rise in inlet pipeline pressure is necessary to maintain a constant discharge pressure at the city gate, and this invariably requires an increase in drive power of 43599.3 kW.

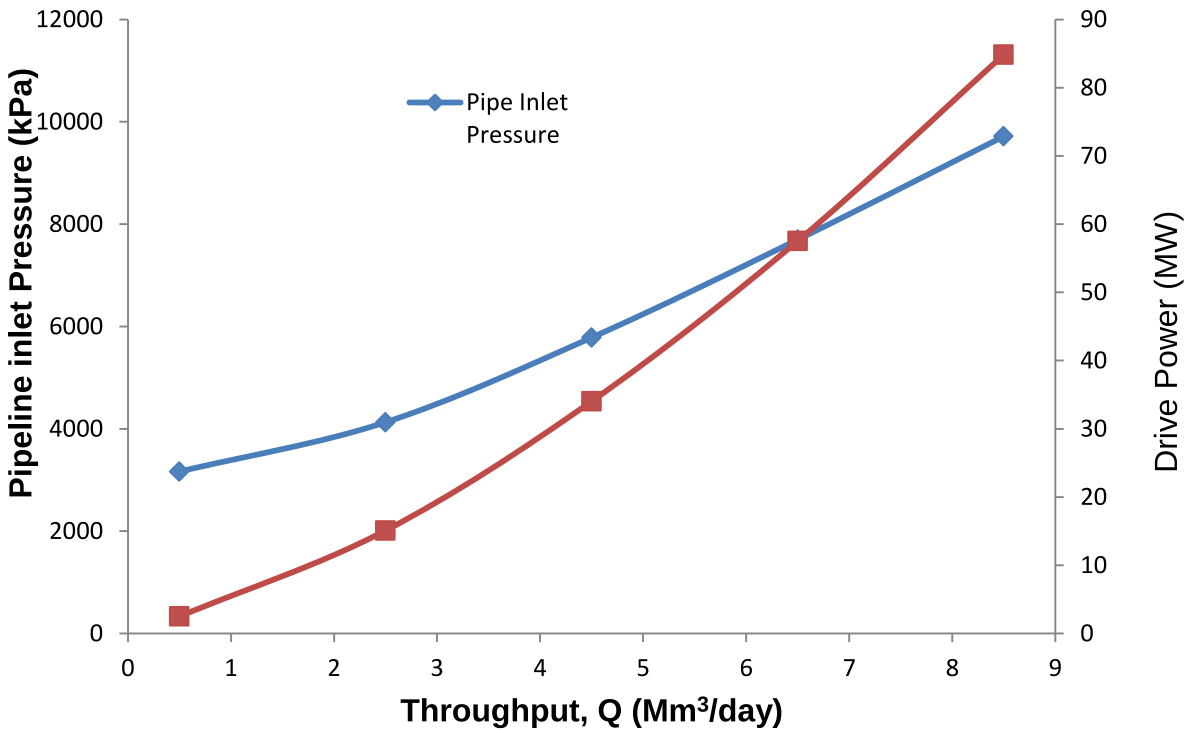

Figure 5-11: Pipe inlet pressure and drive power variation with throughput for pipe diameter = 609.6 mm (24")

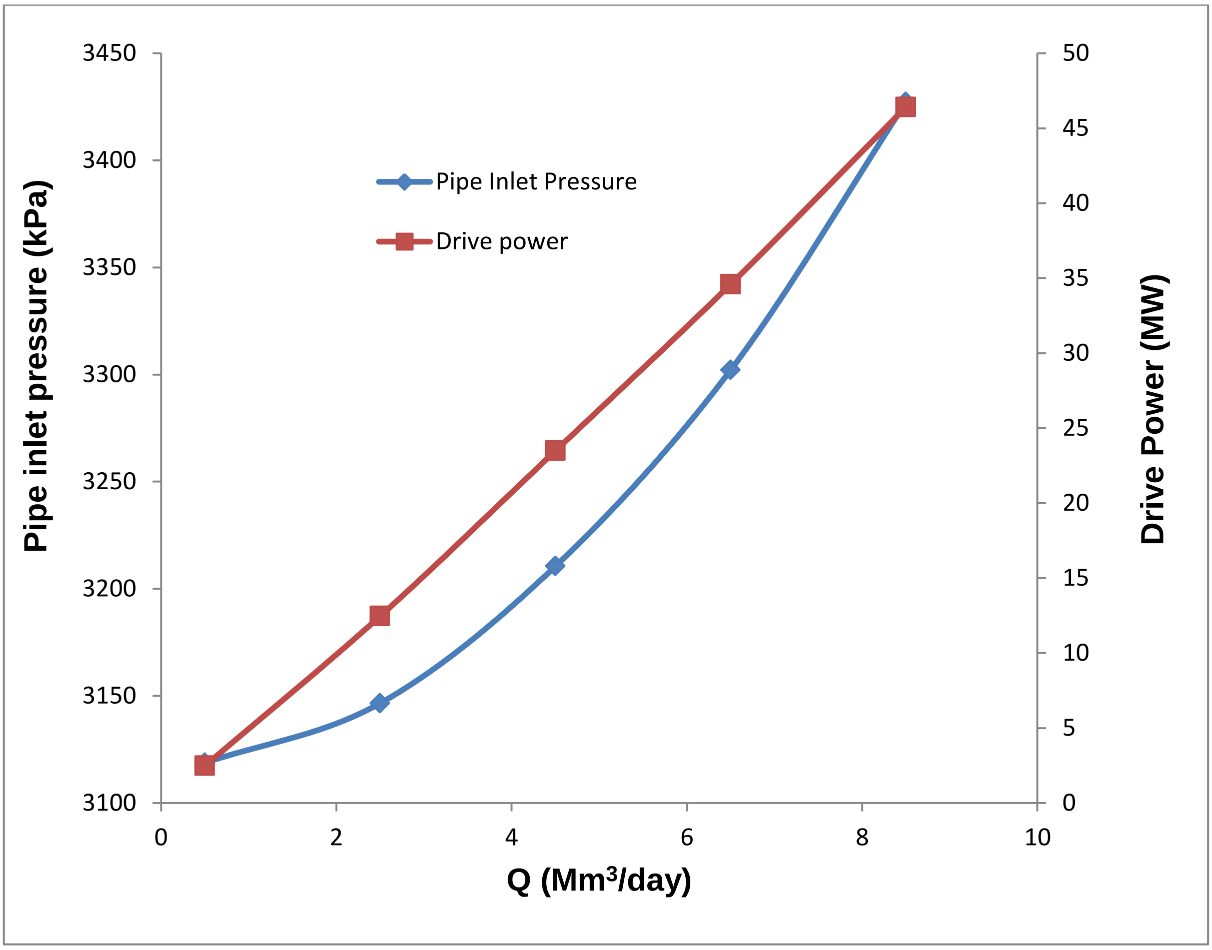

Figure 5-12: Pipe inlet pressure and Drive power variation with throughput for Pipe Diameter = 1219.2 mm (48")

For a pipe size of 609.6 mm –NPS 24 (Figure 5-11), a throughput of 2.5 Mm^3/day and inlet pipeline pressure of 4128.4 kPa, in order to maintain a discharge pressure of 3000.0 kPa at the city gate, a drive power of 15.1 MW is required. And for a throughput of 4.5 Mm^3/day and inlet pipeline pressure of 5783.6 kPa, in order to maintain the same discharge pressure of 3000.0 kPa, a drive power of 34.0 MW is required. This implies that for 80% increase in throughput through a pipeline of 609.6 mm, a 40% rise in inlet pipe pressure is necessary to maintain a discharge pressure of 3000.0 kPa at the city gate, and this consequently requires an increase of 18.9 MW.

Figure 5-12 presents the results for a pipe size of 1219.2 mm. For a throughput of 2.5 Mm^3/day and inlet pipeline pressure of 3146.7 kPa, a drive power of 12.5 MW is required. Similarly for a throughput of 4.5 Mm^3/day and pipeline inlet pressure of 3210.6, the required drive power is 23.5 MW. This implies that for 80 % increase in throughput through a 1219.2 mm (48") pipe size, 2.0 % rise in inlet pipe pressure is necessary to maintain a constant discharge pressure of 3000.0 kPa at the city gate, and this requires an increase of 11.0 MW in the drive power.

This huge increase in the required drive power for a flow through 457.2 mm pipe size makes it uneconomical for the present flow requirements of 4.5 Mm^3/day. For a pipe size of 609.6 mm and 1219.2 mm, the trend is similar but the difference in the increase of the required power is far less with the 1219.2 mm and a compromise between the increase in pipe cost for 1219.2 mm and increase in compression cost for 609.6 mm is required in order to establish the economical pipe size. This conclusion is drawn in the economic module.

5.8 Economic Module

The economic module is essential to the entire TERA methodology. Receiving important data from the pipeline, compressor station and gas turbine simulation models, the economic module computes the capital cost as it relates to all the equipment, operating and maintenance cost for the entire project life. The module takes into account the degradation of gas turbine which could affect fuel consumption as the years roll by. Lifing was not carried out but factors from OEM's confidential data was used in the computations. The module calculates the Life-Cycle cost of the system by establishing the present values of all the costs associated with the plant over its useful life. It finally presents the net present values of all the cash flow all through the plant life. The initial phase of the project was the construction phase which took three years during which no revenue is accrue. The calculation of revenue starts after the construction and commission phase and lasted over the life of the project. The module, which is developed in FORTRAN code, is executed with a switch, 1 for gas turbine economics and 2 for electric motor economics. The results are written to a separate file for graphic presentation. Table 5-2 presents the economic assumptions taken in the economic analysis and which serves as user defined inputs for the module.

Table 5-3: Economic Assumptions

Parameters	Values
Interest rate on Loan	10 %
Production Life	30 Years
Equity	30% of capital
Discount rate	10 %
Federal Income tax rate	30 %
Year to commission	3 years

5.9 Case Study I: Gas Turbine as the Prime Mover for Baseline Natural Gas Pipeline

This case study considers the techno-economic analysis of gas turbine as a prime mover for the baseline natural gas pipeline. The baseline gas pipeline is a 609.6 mm 24 inch pipe spanning from Sarir to Tobruk city gate which is 512 km distance. A 4.54 million cubic meter per day of natural gas is transported with a compressor station requiring a drive power of 34 MW. The performance module results of the gas turbines were presented in an earlier chapter. The results of the integration of all the modules which have the economic module as the core is presented here. The economic appraisal used is the discounted cash flow to analyse the economics of using gas turbine as a prime mover. In order to be able to compare the use of gas turbine and electric motor as pipeline compressor drivers, the transportation cost is analysed for the two types of prime movers studied in this thesis.

Figure 5-13 to Figure 5-23 presents the results of the economic module, the life cycle cost and sensitivity analysis of the project based on certain constraints or varying parameters which were studied for gas turbine option.

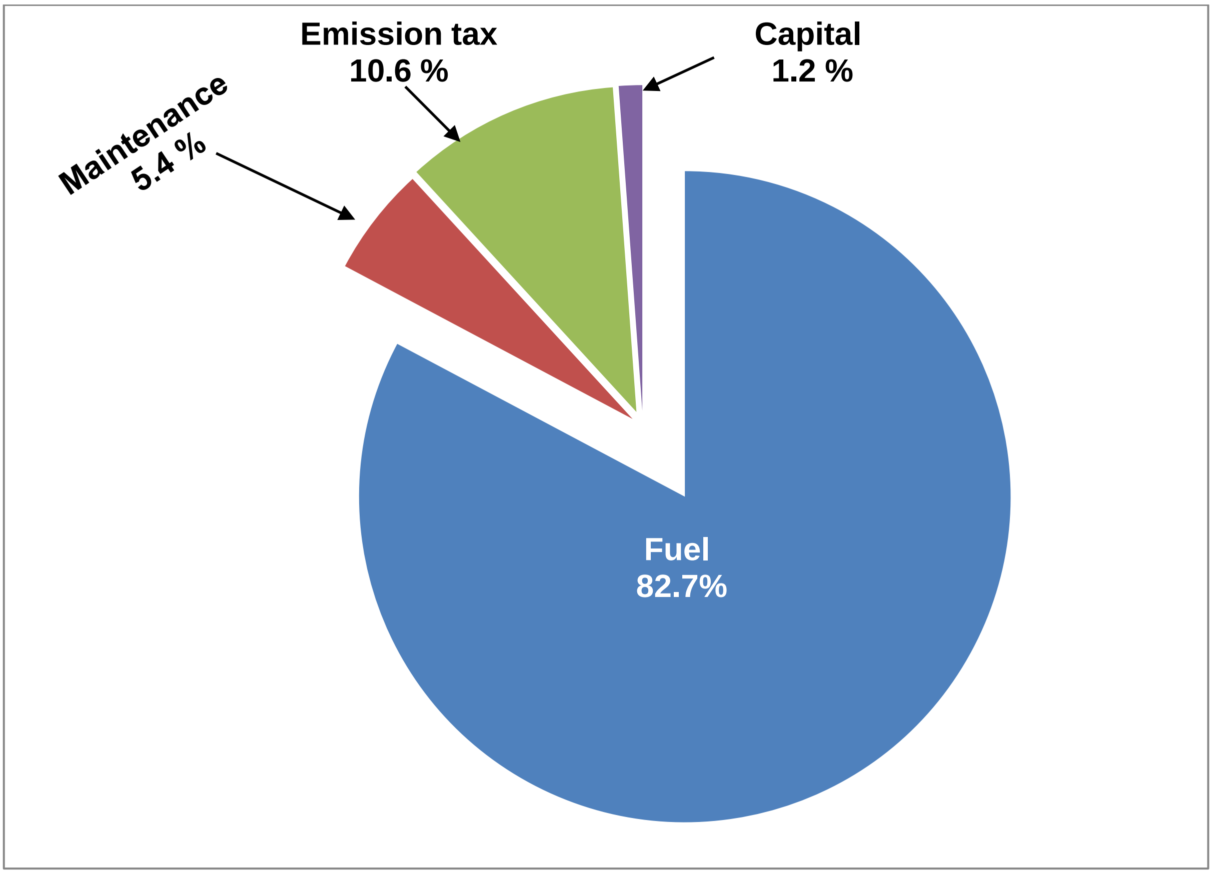

Figure 5-13: Costs breakdown for the baseline plant

Figure 5-13 shows the costs breakdown for the baseline plant. It shows that the fuel cost dominates the entire cost over the life of the plant. Emission is about twice the maintenance cost and capital cost is the lowest of all the cost, all these trends are expected. The life cycle cost of the baseline plant is $1.498 billion.

5.9.1 Effects of Throughput on Operating and Total Cost for Gas Turbine Option

Figure 5-14 shows the effect of throughput on the operating cost for the gas turbine option. It also shows the sensitivity of the effect to pipe sizes and gas prices. For a throughput of 2.5 Mm^3/day through a pipe size of 304.8 mm, the operating cost is $0.697 billion and for the same throughput through a 609.6 mm pipe the operating cost is $0.318 billion, this gives a significant difference of $0.379 billion. This is a huge difference, although there is an effect of increase in pipe material cost as a result of using larger pipe size. For a 4.5 Mm^3/day throughput, the operating cost for using 304.8 mm is 125% higher than using a 609.6 mm pipe sizes. The overall effect is seen when all the cost parameters are put into perspective.

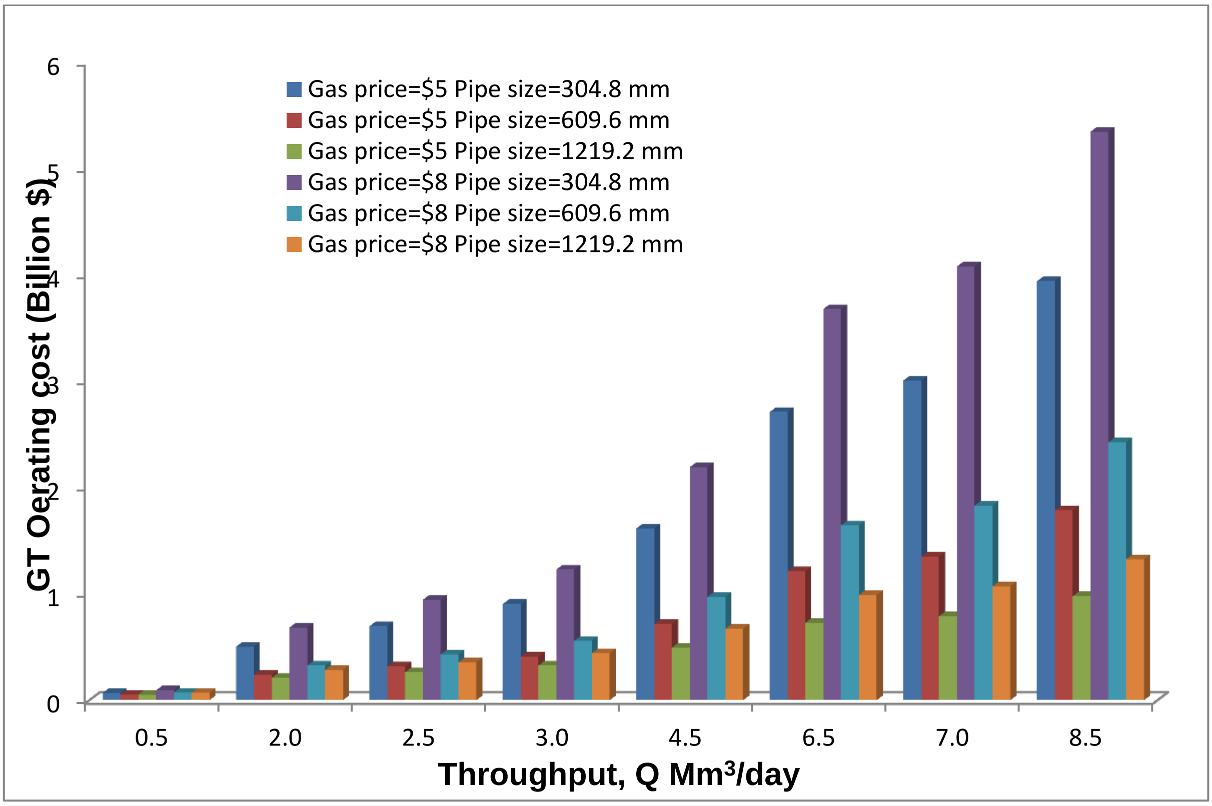

Figure 5-14: GT Operating cost against with gas flow rate for varying pipe sizes and gas price

The higher the throughput the higher is the compressor drive power required for a specified pressure ratio and subsequently the higher the operating cost. For 80 % increase in throughput through a 304.8 mm pipe size there is corresponding increase of 131.9 % increase in operating cost which is $779.2 million. The operating cost increases with increase in gas price, which is to be expected.

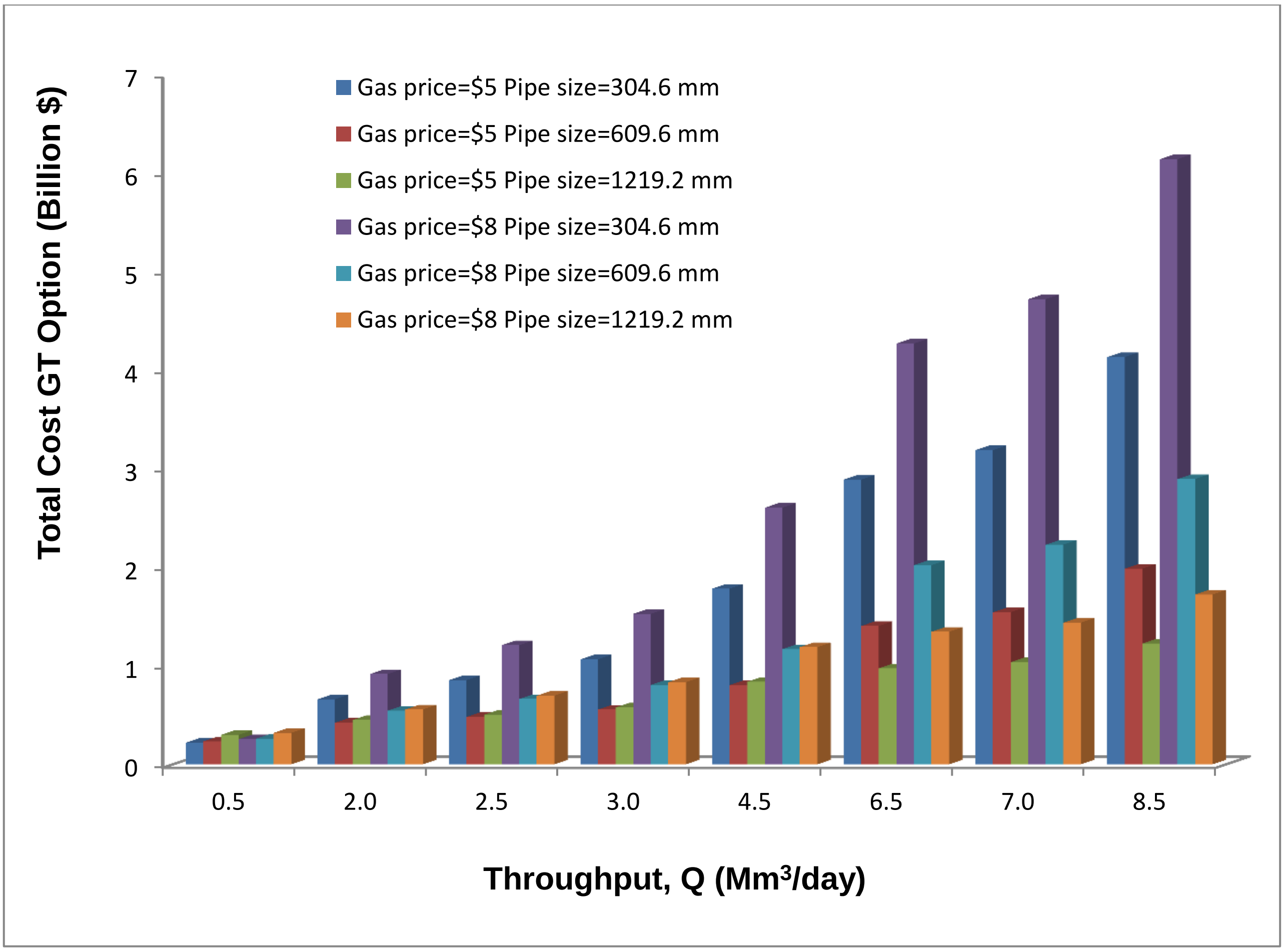

Figure 5-15: GT Total cost against Gas flow rate for varying pipe sizes and gas price

Figure 5-15 depicts the effect of throughput on total cost. The trend is similar to that of operating cost. The total cost for using 304.8 mm pipe size for a throughput of 2.5Mm^3/day is $0.86 billion for gas price of $5.0/GJ while for the same gas price and throughput through 609.6 mm, the total cost is \$0.481 billion and \$0.504 billion for a pipe size of 1219.2 mm. It can be noted that there is a change in the trend of having total cost reducing with increase in pipe size; this is because the pipe material cost for 1219.2 mm dominates the cost effect. At a throughput of 2 Mm^3/day, the total cost of using 609.6 mm at a gas price of \$5.0/GJ is \$0.133 billion lower than using 1219.2 mm pipe size at a gas price of \$8.0/GJ. As the throughput increases, this difference continues to reduce until a throughput of 5.8 Mm^3/day where they become equal. Beyond this throughput, the trend changes and the usage of 1219.2 becomes cheaper than 609.6 mm. This obviously is because two cost component are controlling this effect, viz the pipe material cost and compression cost. As the pipe material cost rises due to increased pipe size, the compression cost reduces, due to reduction in compression power required.

5.9.2 Effect of pipe size on pipe, operating and total cost for GT Option

Figure 5-16 shows the effect of pipe sizes on the two main cost components, pipe material cost and the operating cost, and the comparative effect on the economics of the entire project. For a pipe size of 1219.2 mm the pipe material cost is \$109 million and the operating cost is \$0.42 Billion for gas price of \$5.0/GJ and \$0.67 Billion for a gas price of \$8.0/GJ. Reducing the pipe size for the project to 609.6 mm will give a saving of \$55.1 million in pipe material cost, but an increase in operating cost to the tune of \$0.19 Billion for a gas price of \$5.0/GJ and \$0.30 Billion for a gas price of \$8.0/GJ is incurred. This shows that although a saving in pipe material is obtained by reducing the pipe size, this saving is far less than the increase in operating costs which makes the under sizing of pipe economically unviable and negates the profitability of the entire project.

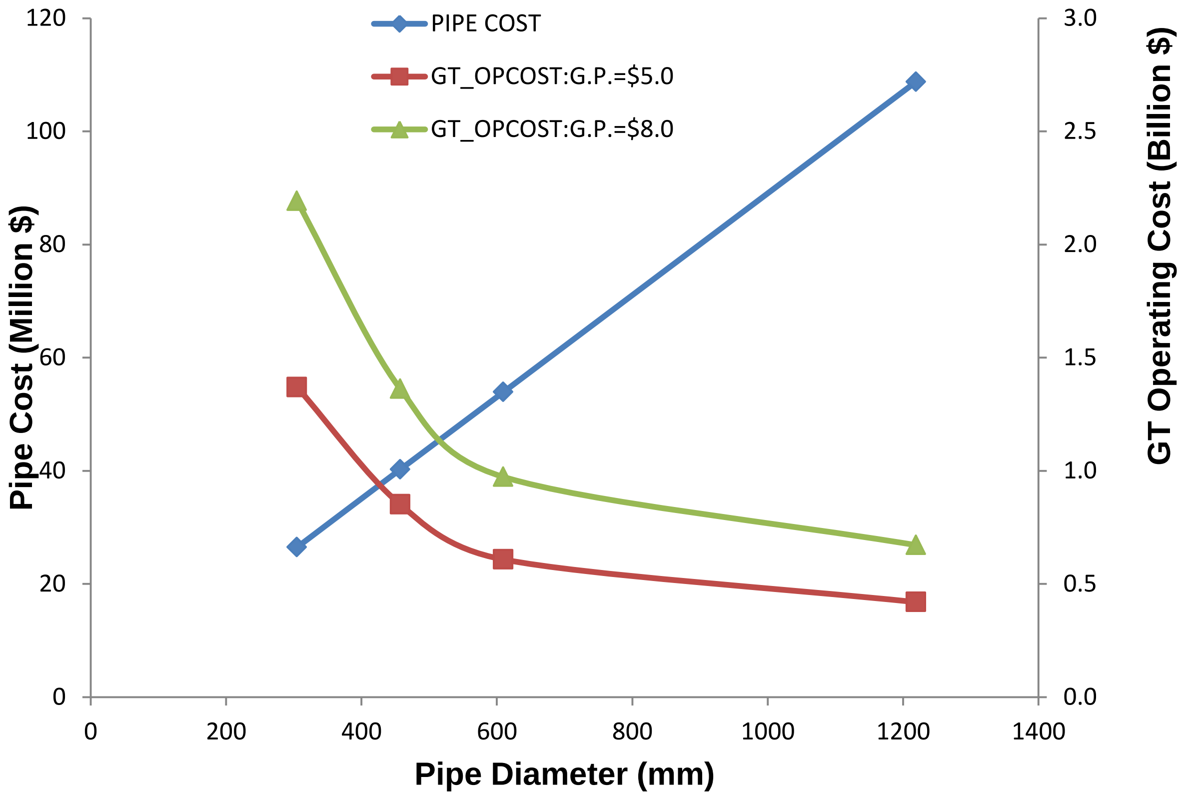

Figure 5-16: GT Operating cost & Pipe cost variation with Pipe diameter

5.9.3 The effect of throughput on NPV

Figure 5-17 and Figure 5-18 presents the effect of throughput on the Net Present Value (NPV) which is the appraisal technique used in this research. The NPV indicates whether a project is economically viable or profitable over a period of time or not. For a throughput of 4.5 Mm^3/day the NPV is $1.55, $1.66, $1.685 Billion employing 304.8 mm, 609.6 mm and 1219.2 mm pipe sizes respectively. For an increase in throughput to 6.5 Mm^3/day, the NPV is $2.19, $2.39, $2.44 Billion using 304.8 mm, 609.6 mm and 1219.2 mm pipe sizes respectively. It is generally seen that the NPV increases with increase in throughput. An increase of 2 Mm^3/day gave rise to $0.73 Billion increase in NPV.

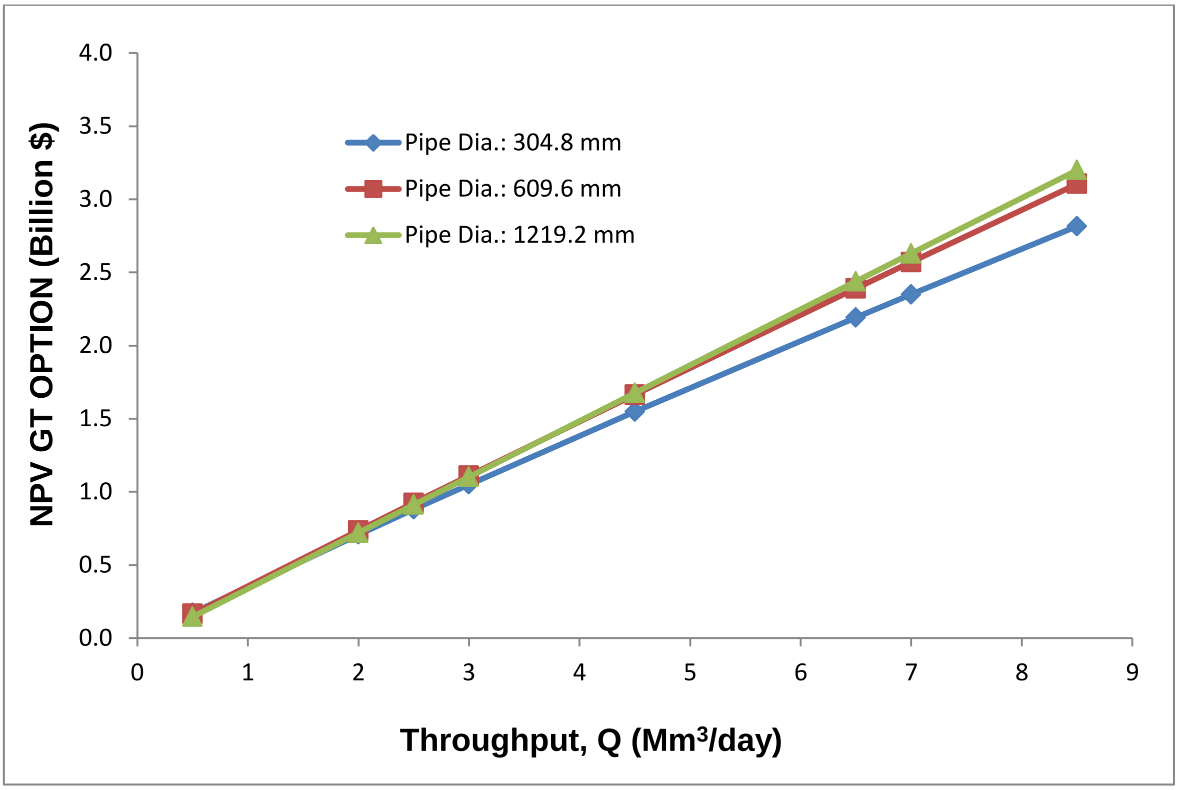

Figure 5-17: NPV against Gas flow rate for varying pipe sizes (GT Option)

Figure 5-18 shows that for any increase in throughput there is corresponding increase in drive power required, which consequently means increase in the capital and operating investment. The huge increase in NPV, despite the increase in capital and operating investment, is due to large economies of scale attributed to natural gas pipeline transportation system(Subero et al., 2004). The NPV also increases with increase in pipe sizes for the same throughput. This is because of increase in drive power which contributes to the controlling cost component – operating cost.

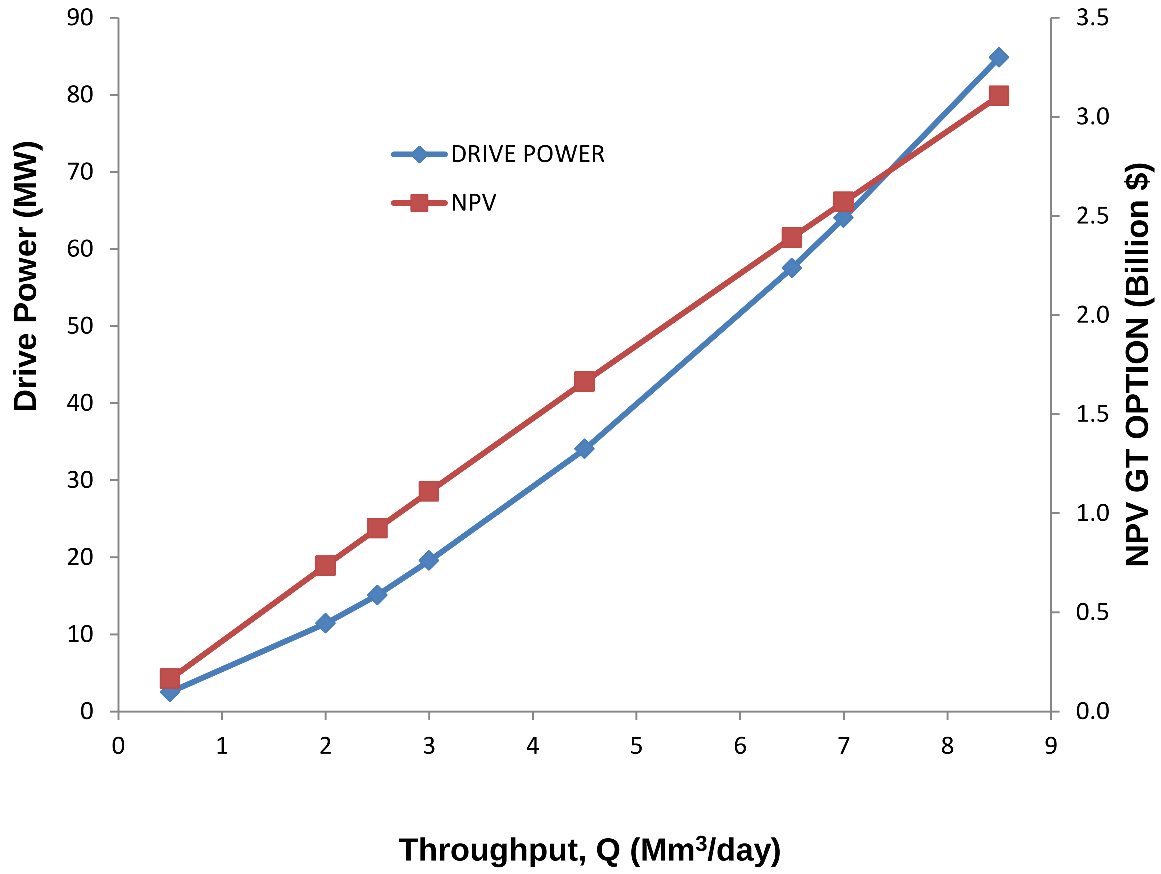

Figure 5-18: Drive power and NPV against gas throughput (GT Option)

5.9.4 Effect of discount rate on NPV

Figure 5-19 shows the effect of varying discount rate on NPV for different throughput. This is a financial risk study. For a throughput of 4.5 Mm^3/day, the NPV is $3.09 Billion at a discount rate of 5 % and 1.03 billion at a discount rate of 15 % which 200 % reduction in NPV. At a discount rate of 5%, the NPV is $291 million and $4.44 billion for 0.5 Mm^3/day and 6.5 Mm^3/day respectively. This significant difference of $4.15 billion is reduced to $491 million at a discount rate of 30%. Since discount rate is the rate at which future value is discounted, it follows that an increase in discount rate will correspondingly reduce the NPV.

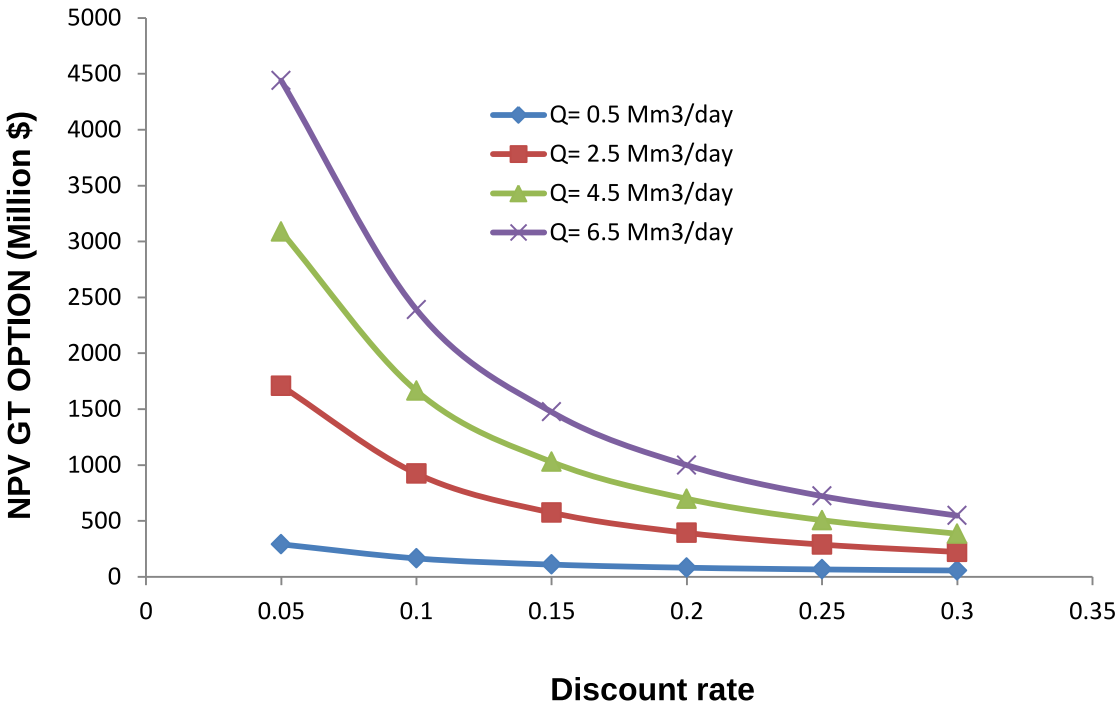

Figure 5-19: NPV against discount rate for varying throughput (GT Option)

5.9.5 Effect of Project life on NPV

Figure 5-20 present the effect of project life on NPV at different discount rate. The NPV is \$2.52, \$2.84 and \$3.09 billion for a project life of 20, 25 and 30 years respectively.

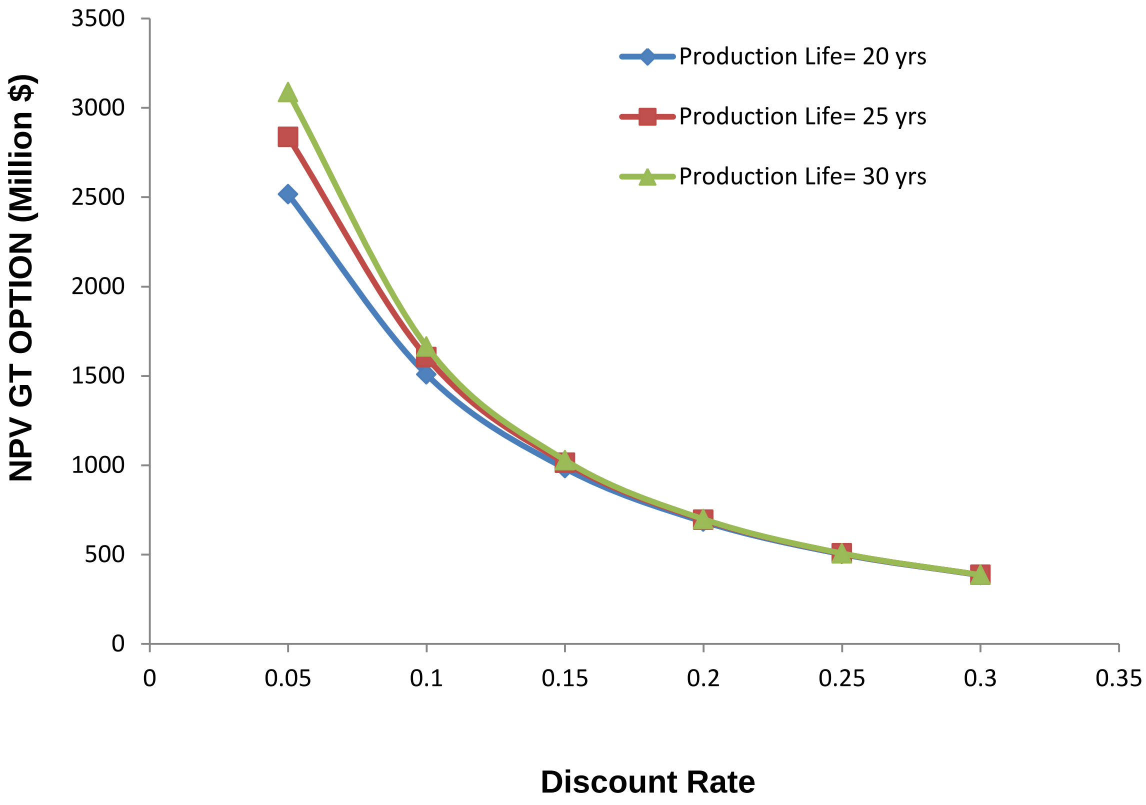

Figure 5-20: NPV against discount rate for varying project life (GT Option)

At low discount rate the effect of plant life is pronounced but with increase in discount rate the effect of the project life becomes less significant. The graph also shows that as plant life increases there is lesser difference in the values of NPV, this may be attributed to the degradation of plant reaching a level where it is becoming worth less to continue to maintain thereby bringing the maintenance over the years to a normalised average. It is worth designing for longer plant life at low discount rate as shown in Figure 5-19. For example at a discount rate of 5%, the NPV for 20 year plant life is $2.52 billion while for 25 year plant life is $2.84 billion, this yields an increase of $0.32 billion. But for 30 year plant life the NPV is $3.09 billion yielding an increase of $0.25 billion over that of 25 year plant life. It can be seen that the return seems to decrease with increased plant life.

5.9.6 Effect of Gas price and Discount rate on NPV

Figure 5-21 shows the effect of discount rate and gas price on NPV. Increasing the discount rate implies taking more financial risk and by so doing the NPV value drops. The value of the NPV can be seen to increase with increase gas price and the converse as the gas price reduces. It is obvious that when the gas price reduces, the NPV must reduce because there is drop in the revenue. The effect of the discount rate on the NPV is more pronounced than the effect of gas price. This makes the financial risk an important factor to consider in developing a natural gas pipeline project since it has more effect on the projected profit than the price of gas used to run the plant. Depending on the region of the pipeline, the risk may be high and this will impact on business decisions by an operator who will be looking for greater returns for projects with higher risks. This ultimately will guide the negotiation and final business decision between the pipeline operator and the gas company.

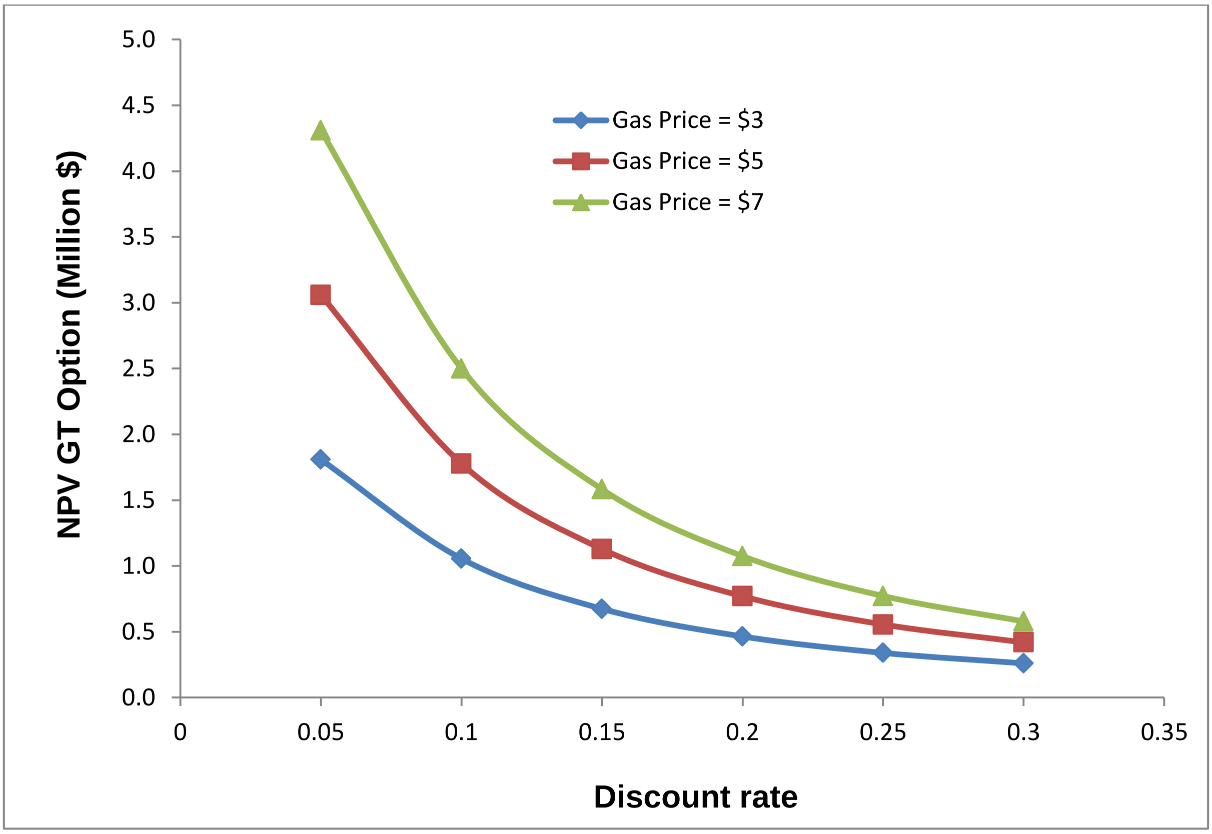

Figure 5-21: NPV versus Discount rate & Fuel price

5.9.7 Effect of throughput on Gas Transportation Cost (GT Option)

Figure 5-22 presents the effect of throughput on gas transportation cost for varying pipe sizes. The transportation cost is \$0.10/m^3, \$0.11/m^3 and \$0.15/m^3 at \$5.0/GJ gas price for 0.5 Mm^3/day through 304.8 mm, 609.6 mm and 1219.2 mm pipe sizes respectively. The figure depicts the optimum natural gas throughput for each pipe sizes and this defines the economic pipe size.

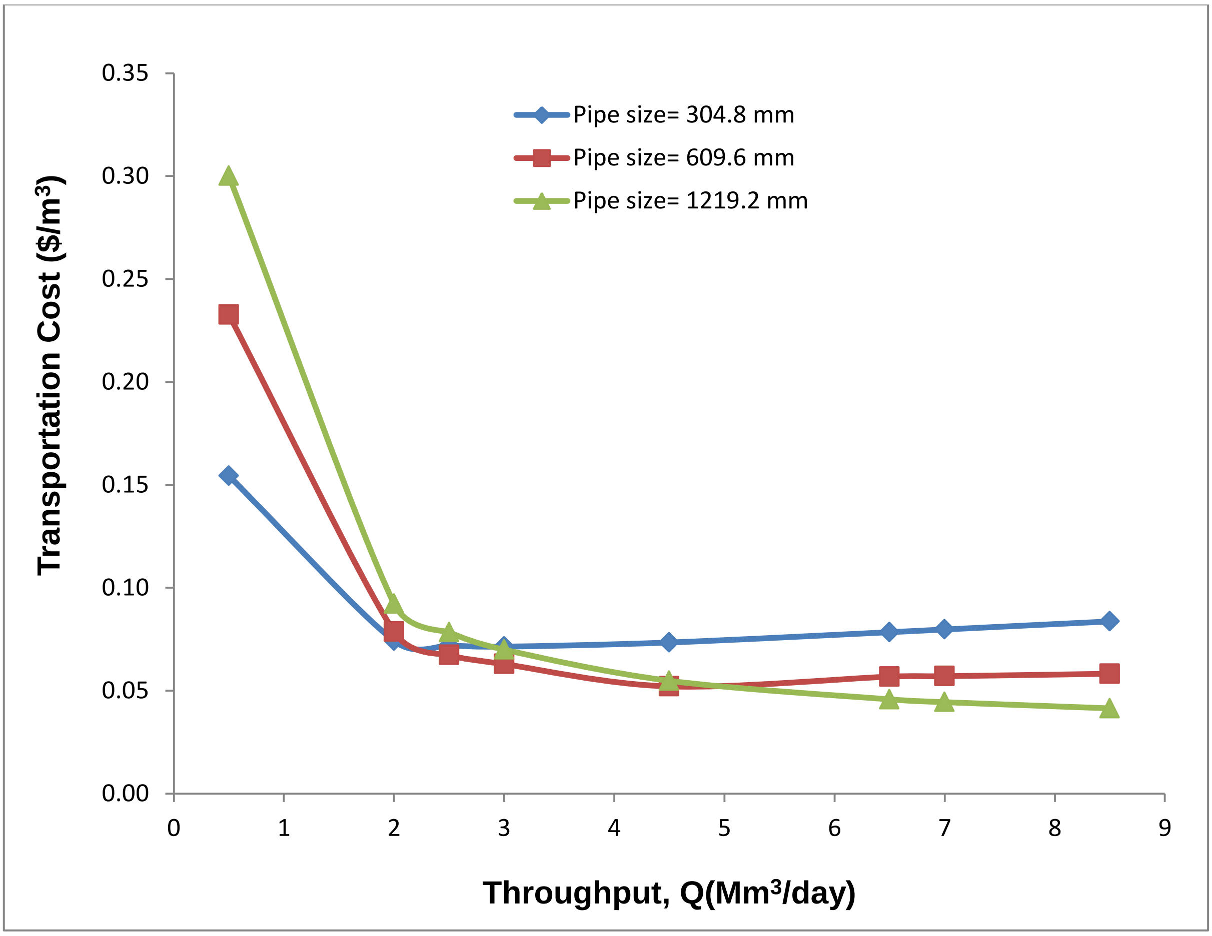

Figure 5-22: Cost per unit of gas variation with throughput for different pipe sizes

The economic pipe size for 0.5 Mm^3/day is 304.8 mm (12”), with a transportation cost of $0.15/$m^3$ which is equivalent to $3.95/GJ, is because the high pipe cost for large pipe sizes controls the overall cost at this point. Although this is seen to be the lowest cost of transporting natural gas, it is grossly uneconomical as the transportation cost is far beyond the expected transportation cost by pipelines and this suggest that transporting a 0.5 million cubic meter per day of natural gas over long interstate pipeline is uneconomical. For a 2.5 Mm^3/day throughput, the transportation cost is $0.072/$m^3$, $0.067/$m^3$, and $0.078/$m^3$ at $5.0 gas price and through pipe sizes of 304.8 mm, 609.6 mm and 1219.2 mm respectively. As the flow increase the trend of the effect changes and small pipe sizes become less economical and large pipe sizes become more profitable. There exist a changing point in the inter-play between pipe material cost and the compression cost. At very high throughput such as 7.0 Mm^3/day, it is noted that pipe size of 1219.2 mm continues to take the lead on the profitability, as it presents the least transportation cost. For each of the pipe sizes a point is noted which is the throughput where they present lowest cost, and beyond this point there is noticeable rise in the transportation cost. At a gas price of $5/GJ, it is 2 Mm^3/day for 304.8 mm, 4.5 Mm^3/day for 609.6 mm and for this study 8.5 Mm^3/day for 1219.2 mm. Although this result has not reached the minimum of 1219.2 mm pipe sizes, it is believed that beyond 8.5 Mm^3/day, a point exist where the transportation cost will increase.

5.10 Case Study Two: Electric Motor as the Prime Mover for Baseline Natural Gas Pipeline

In case study II, electric motor was considered for the prime mover option for the same baseline natural gas pipeline as presented in case study I. A 34 MW induction motor with a conversion efficiency of 98% was considered. The details of the electric motor performance was presented in chapter three of this thesis. The result of the integrated modules with electric motor module and economic module is presented and discussed here.

5.10.1 The Cost Components of Induction Motor Operation

The two cost component of induction motor operation are maintenace and operating cost. Maintenance cost is usually estimated by percentage of the power rating while operating cost is quite straight forward as the electricity consumed can be obtained from a meter reading, or calculated knowing the efficiency of the electric motor and subsequently with the knowledge of the tarrif applied, the operating cost can be computed. The load applied on an electric motor controls the efficiency, and consequently the kW required to deliver an expected output power. As the load reduces, the efficiency drops and as it increases the efficiency rises, until the rated load where the efficiency is maximum.

The cost of power transmission loss can be high, depending on whether the power is being transmitted at high or low voltage. Power transmitted at high voltage transmits low current and consequently low power loss.

5.10.2 Effects of Throughput on Operating and Total Cost (Electric Motor Option)

Figure 5-23 presents the effect of throughput on the operating cost for electric motor drive option. The operating cost increased from $0.0615 billion to $0.623

billion as the throughput increased from 0.5 Mm^3/day to 2.5 Mm^3/day for a pipe size of 304.8 mm and electricity tariff of \$0.05/kWh. This amounts to a difference of \$0.562 billion. For 609.6 mm pipe size the operating cost increased from \$0.0472 billion to \$0.285 billion as the throughput increased from 0.5 Mm^3/day to 2.5 Mm^3/day which amounts to a difference of \$0.237 billion. The rise in operating cost as a result of increase in throughput reduces with increase in pipe size. The sensitivity shows that the higher the electricity tariff, the higher the operating cost. Pipe size 1219.2 mm is seen to have the least rise in operating cost as drive power is minimal with flow through it. Figure 5-23 also shows that the operating cost tends to zero as the throughput reduces to zero.

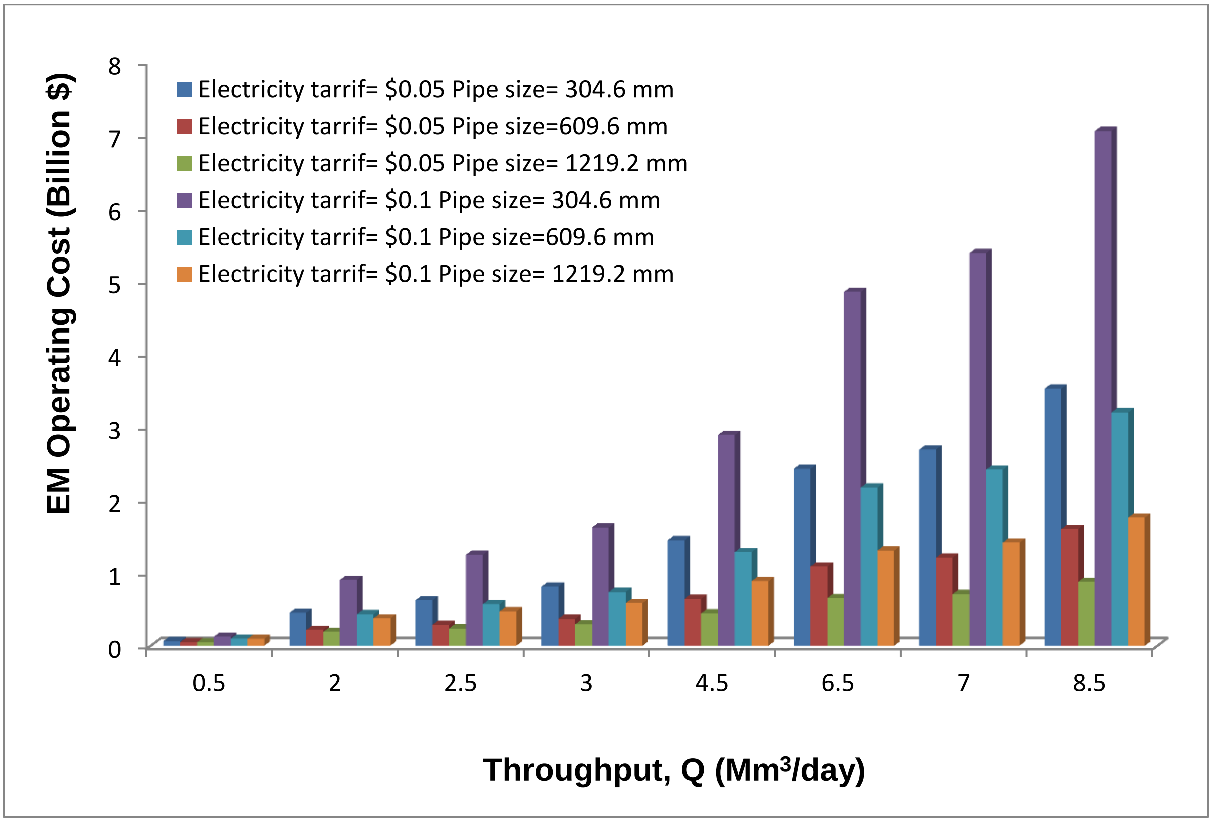

Figure 5-23: EM Operating cost against with gas flow rate for varying pipe sizes and electricity price

Figure 5-24 shows that the effect of throughput on the total cost, this follows the same trend as the operating cost. The total cost increase from \$0.220 billion to \$0.853 billion as the throughput increased from 0.5 Mm^3/day to 2.5 Mm^3/day through a 304.8 mm pipe size and at \$0.05/kWh.

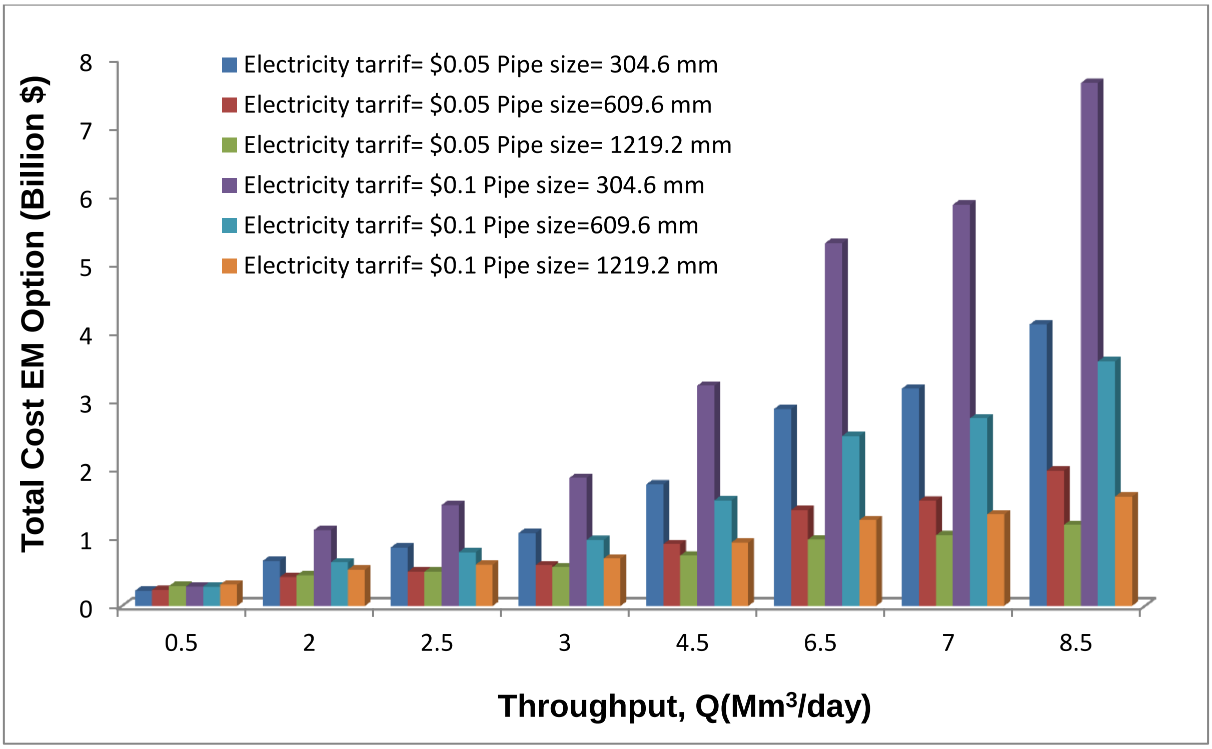

Figure 5-24: EM Total cost against Gas flow rate for varying pipe sizes and electricity price

For a pipe size of 609.6 mm, the total cost for a throughput of 4.5 Mm3/day at electricity tariff of \$0.05/kWh is equal to the total cost of the same throughput through a 1219.2 mm pipe size at electricity tariff of \$0.1/kWh. This is because for the smaller pipe size the effect of the increase in drive power was reduced with small electricity tariff, and for the larger pipe size the effect of reduced drive power was increased by the high electricity tariff and pipe material cost. Beyond the throughput of 4.5 Mm^3/day, 1219.2 mm pipe size continues to have lower total cost for an electricity tariff of \$0.05/kWh.

5.10.3 Effect of pipe size on operating and pipe material cost

Figure 5-25 presents a comparative analysis of the effect of pipe size on pipe cost and operating cost.

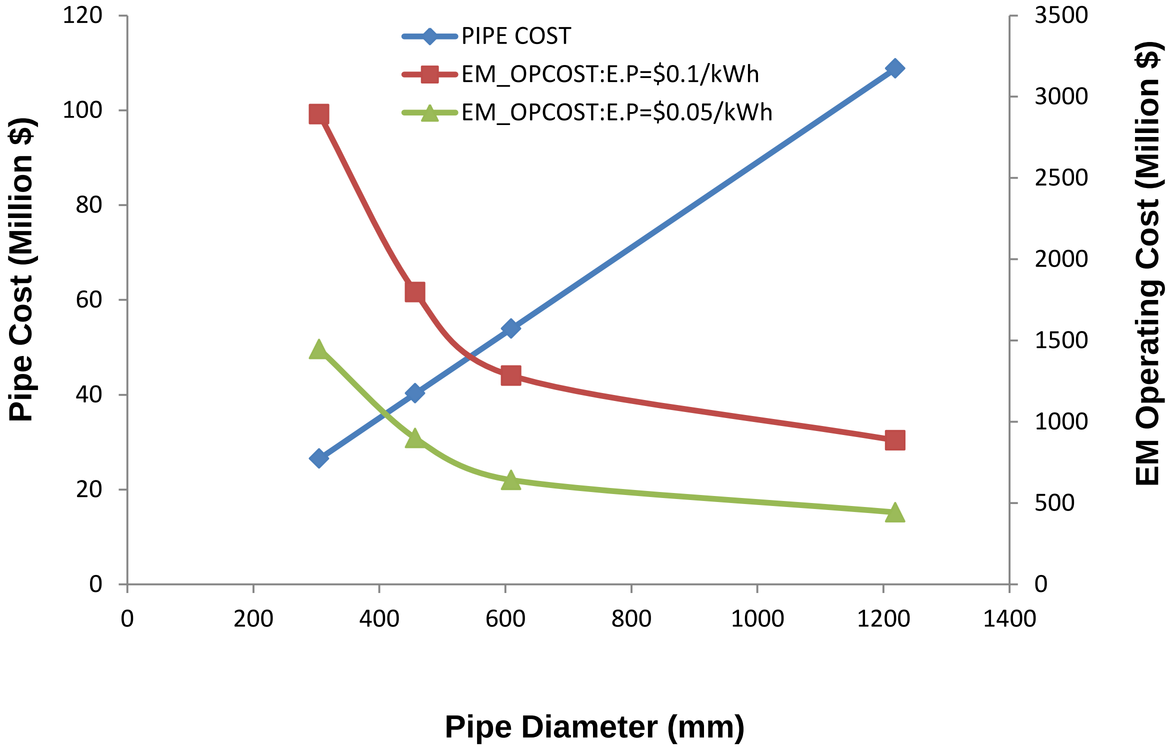

Figure 5-25: EM Operating cost & Pipe cost against with Pipe diameter for varying electricity price

The pipe material cost for the usage of 609.6 mm size is $53.9 million and $26.5 million for a pipe size of 304.8 mm. This gives a savings of $27.4 million in material cost. On the other hand the operating cost for an estimated electricity tariff of $0.05/kWh is $641.9 million using a pipe size of 609.6 mm and for a pipe size of 304.8 mm and the same electricity tariff, the operating cost is $1.4 billion. This amounts to an increase of $804 million. It shows that although under-sizing of pipe apart from technical issues has negative economic impact on the natural gas pipeline project. The saving in pipe cost is only about 3.4 % of the increase in the operating cost.

5.10.4 Effect of throughput on NPV (EM Option)

Figure 5-26 presents the effect of throughput on the Net Present Value for the Electric Motor (EM) option for varying pipe sizes. For a throughput of 4.5 Mm^3/day through a 609.6 mm pipe size the NPV is \$2.7 billion and for a throughput of 7.0 Mm^3/day, the NPV is \$4.2 billion. This is an increase of about 55 % in NPV. But for a 4.5 Mm^3/day through a 304.8 mm pipe size, the NPV is \$2.6 billion and \$4.0 billion for a throughput of 7.0 Mm^3/day. This gives a percentage increase of 53.8%.

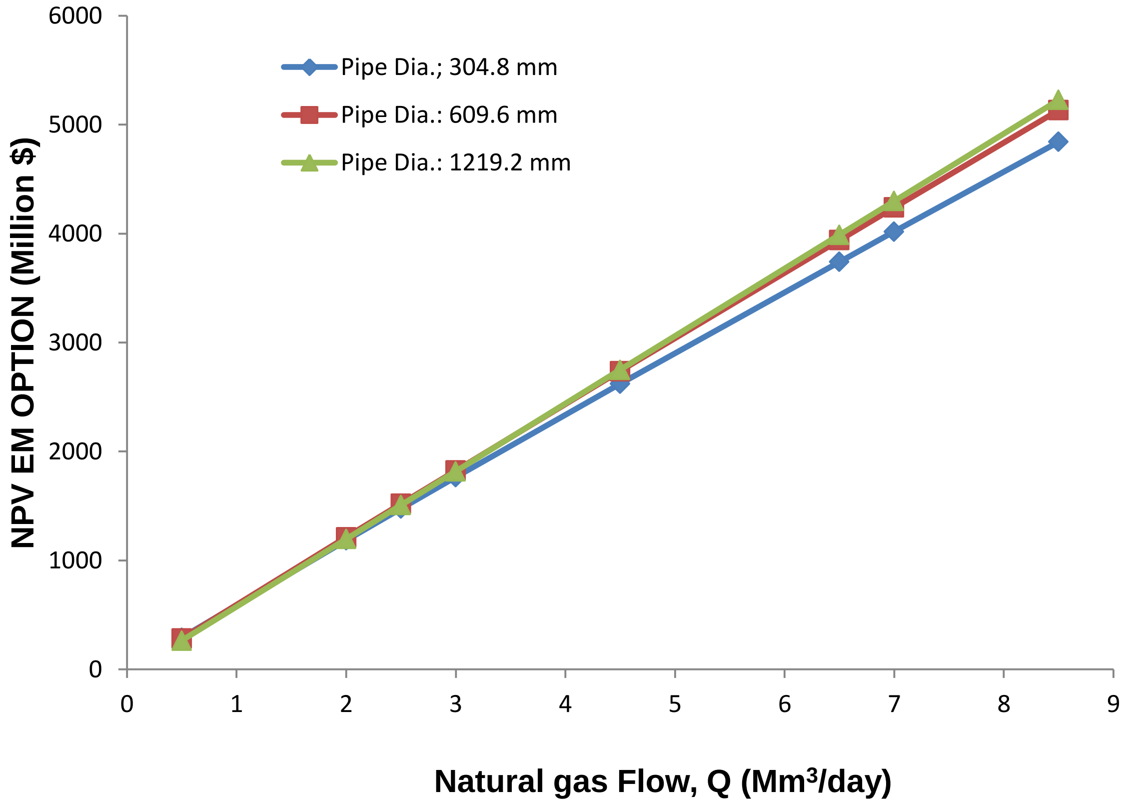

Figure 5-26: NPV against Gas flow rate for varying pipe sizes (EM Option)

The increase in throughput undoubtedly increases the compressor drive power required, as shown in Figure 5-27. This rise in drive power will also cause an increase in the capital and operating costs.

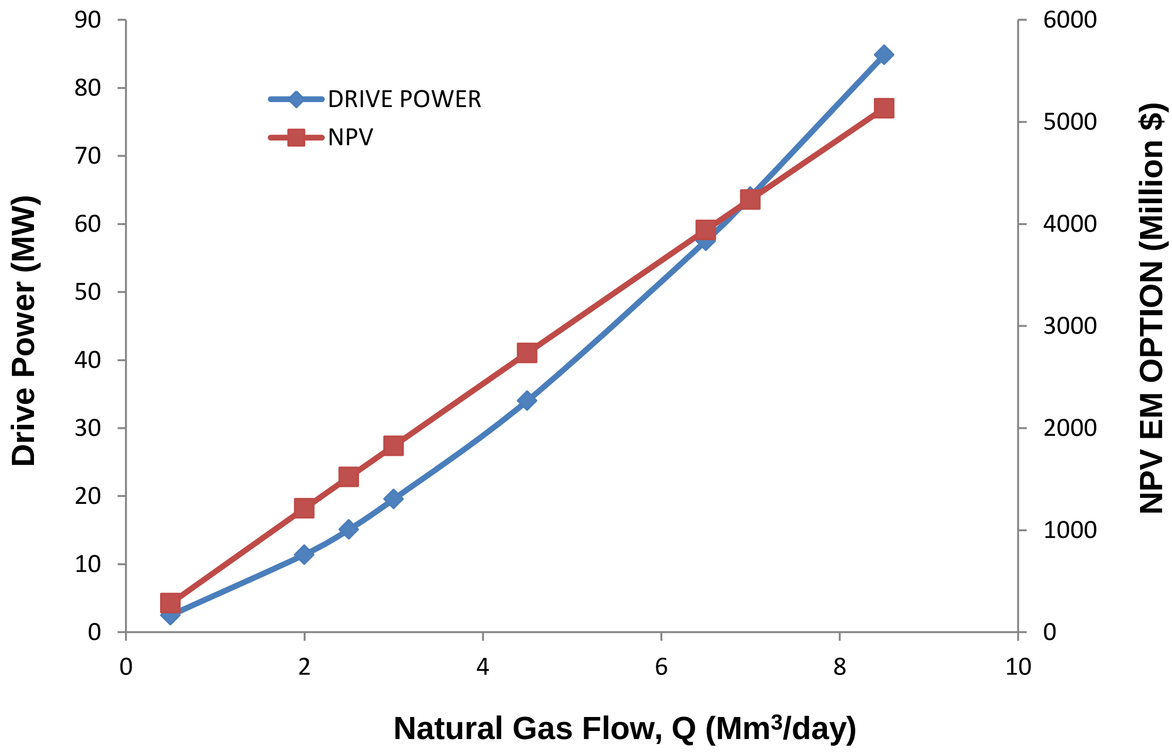

Figure 5-27: Drive power and NPV against gas throughput (EM Option)

Figure 5-27 shows the trend of variation of drive power and NPV as the throughput changes. The increase in NPV, which puts together all the cost components and cash flows over the project life, despite the increase in capital and operating costs, further confirms the possession of economies of scale by pipeline transportation systems.

5.10.5 Effect of throughput on Gas Transportation Cost (EM Option)

Figure 5-28 presents the effect of throughput on gas transportation cost for varying pipe sizes using electric motor as driver. For a throughput of 0.5 Mm^3/day, the transportation cost is \$0127, \$0.192 and \$0.249 for pipe sizes of 304.8 mm, 609.6 mm and 1219.2 mm respectively. It is seen that 304.8 mm presents the lowest cost at an electricity tariff of \$0.05/kWh. This amount is equivalent to \$2.95/ GJ, which is higher than what is expected of pipeline transportation cost and this suggest that 0.5 million cubic meter per day is not economical to be transported over long interstate pipelines.

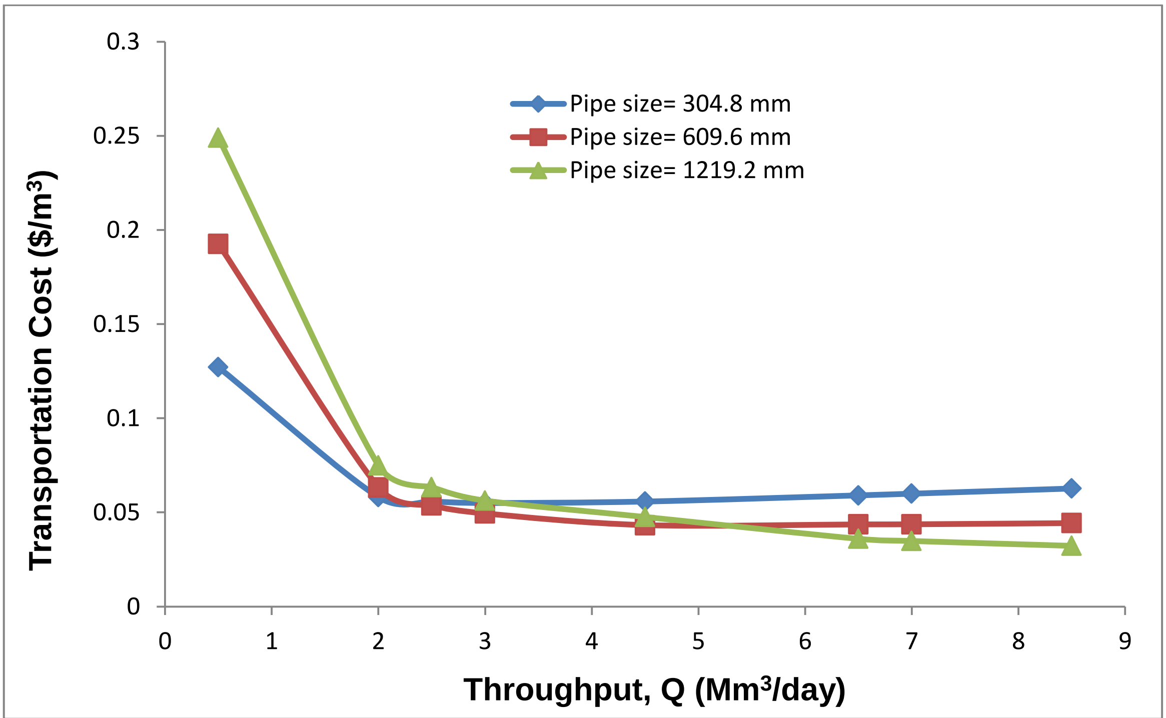

Figure 5-28: Transportation Cost per unit of gas against throughput for varying pipe sizes

An increase in throughput from 0.5 Mm^3/day to 2.0 Mm^3/day gave rise to a sharp drop in the gas transportation cost across all the pipe sizes and 609.6 mm pipe size has the least transportation cost of \$0.054 although that is not the optimum point for it. Beyond 2.5 Mm^3/day, an increase in transportation cost is noted for pipe size 304.8 mm. This implies that the optimum throughput for 304.8 mm is 2.5 Mm^3/day which yields \$0.055 transportation cost. For a 609.6 mm pipe size and electricity tariff of \$0.05/kWh, the transportation cost rise as the throughput goes beyond 6.5 Mm^3/day. For 1219.2 mm pipe size the transportation cost continues to drop all through the throughput studied for the studied electricity tariffs. It is believed that there will be an increase in the transportation cost at a point as the throughput continues to rise. This point will be the optimum throughput for 1219.2 mm pipe size. At a throughput of 8.5 Mm^3/day, 1219.2 mm presents the lowest gas transportation cost of \$0.032.

6 ECONOMIC OPTIMIZATION OF COMPRESSOR STATION POSITIONING

6.1 Introduction

Optimization is an act, process, methodology or procedure(s) used to make a system or design as effective or functional as possible, especially the mathematical techniques involved. It could involve maximizing or minimizing a function called the objective function, subject to certain constraints imposed on the variables of the function. The objective function and constraints can be linear or nonlinear; the constraints can be bound constraints, equality or inequality constraints, or integer constraints. Traditionally, optimization problems are divided into Linear Programming (LP; all functions are linear) and Nonlinear Programming (NLP). Figure 6-1 depicts the general optimization process. It shows that the optimization process varies the input to achieve the desired output.

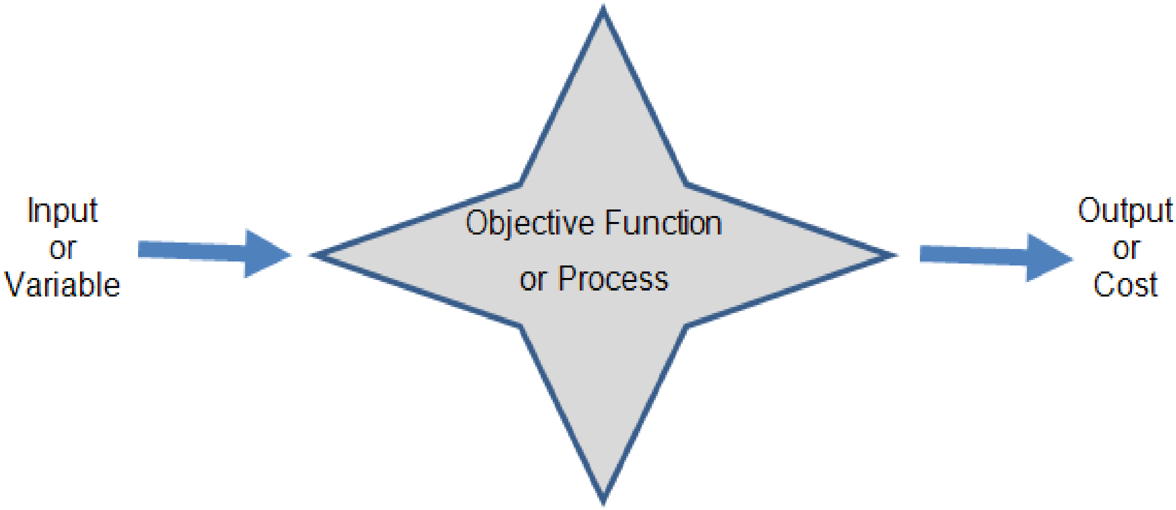

Figure 6-1: Optimization process

Optimization algorithms can be divided into six categories as shown in Figure 6-2 with each categary not necessarily mutually exclusive. A dynamic optimization could be constrained or unconstrained and the variable may be discrete or continuous.

Figure 6-2: Six categories of optimization algorithm, adopted from (Haupt and Haupt, 2004)

Trial and error optimization involves adjusting variables without detailed knowledge of the process leading to the output. When the output of an optimization is a function of time, the optimization is said to be dynamic, and static if the converse is the case.

Optimization algorithm can also be classified according to method of operation into deterministic and probabilistic algorithm.

6.2 Overview of Genetic Algorithm

Genetic Algorithms (GA) which are part of the group of Evolutionary Algorithms (EA) are direct, parallel, stochastic methods for global search and optimization, which imitates the evolution of the living beings, described by Charles Darwin. GAs are powerful and broadly applied stochastic search and optimization techniques which are the widely known current types of evolutionary computation methods, with the GA community having its attention turned to optimization problems in industrial engineering over the past decade (Gen and Cheng, 2000; Ḃeack, 1996; Fogel, 2005; Michalewicz, 1998). GA is based on the basic phenomena of chromosome gene code interchange, a process fully based on random selection of data and modification.

Generally, a genetic algorithm has four basic components as presented by (Michalewicz, 1998):

- A genetic representation of solution
- A procedure for the creation of the initial population of solutions
- Genetic operators responsible for the altering the genetic composition of children during reproduction
- Values for the parameters of GAs

The basic steps followed by Genetic Algorithm involve the random creation of an initial population from the supplied data, which represents a population of individual solutions covering the entire range of possible solutions. This means data for population is selected in a random manner. All the individuals in the search space are encoded using a mathematical object or a fixed length character string.

The GA usually attempts to find the best solution to the problem by genetically breeding the individuals in the population over a number of generations.

The next step is to evaluate the data and to check if the stopping criteria are met. The positive response will stop iteration and present the result. The negative response will give rise to the chance of parent selection. After a random selection of parents, crossover is performed based on crossover probability. To make things clear, the points of crossover and the selection of parents, both are done randomly. After this step, the mutation will be performed. In this case, the number of points of mutation depends on the probability of mutation, generally a user input. Now the data are evaluated using the equations. The data that give us best results, for example, best cost choice, or best operational time choice, or any other criteria, is chosen to be the best fitted data.

Then a step called Roulette Wheel is used to replace the worst fitted data. The general process is to replace the worst fitted value by the best fitted value. Sometimes mean fitness value is used and data with fitness values below 50% of the mean fitness value are replaced by the best fitted data. If the best fitted value of the table of values achieved by this process is closer to the criteria given by the user, then the older table is replaced by this new table. This process is schematically shown in Figure 6-3.

The Iterative process in the GA repeats the following instructions from step 2 on the population until the stopping criterion are met:

1. Assigning a fitness value to each individual in the population using the fitness measure
2. Selecting individuals from the population with a probability based on
    a. Fitness. Three genetic operations are applied to the selected individuals to create a new population, these operations are:
    b. Reproduction, by simply copying an existing individual into the new population.

c. Creating a new individual from an existing individual by randomly mutating the string, usually at a single randomly chosen position.

d. Creating new strings from two or more existing strings by genetically recombining substrings using the crossover operation at randomly chosen crossover points.

3. The best individual is noted as the result of the genetic algorithm for the run. This individual may or may not represent a solution, or an approximate solution to the problem.

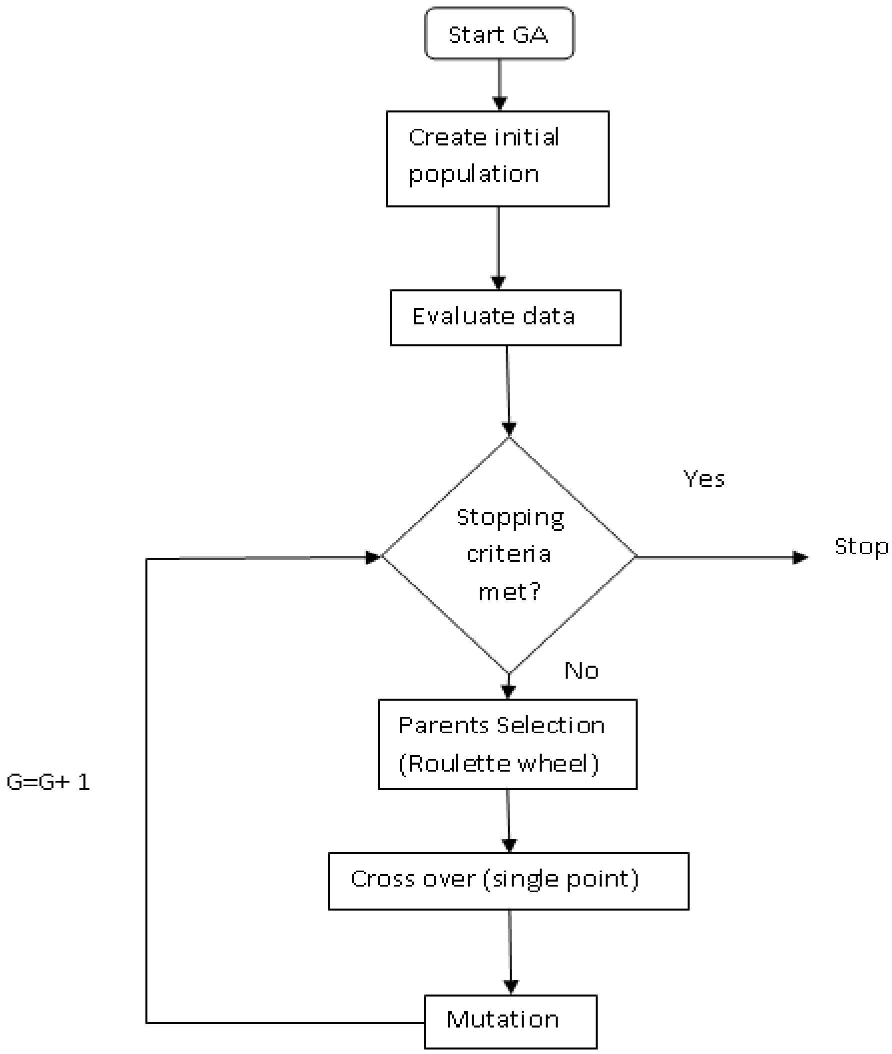

Figure 6-3: Steps of a Simple Genetic Algorithm Process

Each iteration of this iterative process is termed a **generation.** GAs is usually iterated from about 50 to 500 or more generations, with the entire set of generations called a **run**. One or more highly fit chromosomes results at the end of a run and because the process is characterized by randomness, the results of two runs with different random number seeds will generally give results with different detailed behaviours. The statistics of the report showing the best fitness found in a run and the generation in which the individual with that best fitness was discovered, this is averaged over many different runs of the genetic algorithm of the same problem.

The above description gives the basic process in a genetic algorithm; further discussion on each step of the GA process will be presented in the following sections of this thesis.

In employing a GA scheme to solve a problem, it is important for the representation scheme to be capable of finding a solution to the problem, in other words, it should be able to converge with an acceptable result, showing the minimum or maximum of the objective function as the case may be. Four preparatory steps are also significant before initiating a GA to solve a problem using fixed length strings, these are:

- The representation scheme
- The measure of the fitness of individuals
- The parameters and variables for controlling the algorithm
- Designation for terminating a run or arriving at a successful conclusion.

The fitness measure is a rule that guides the evolutionary process. Each individual in the population is assigned a fitness value with number of individuals in a generation usually being from tens to thousands and this number controls how fast the algorithm can move from generation to generation. The number of generations required before a solution is reached can be many thousands. With the selection of smaller population, more generations are required before a final solution is reached, but the algorithm moves faster from generation to generation.

6.3 Main Components and Genetic Algorithm Method

The main components which constitute the working elements of the GA for solving the optimization problem are discussed in this section. In the present research, GA Toolbox in MATLAB was used for the initial trials and thereafter a MATLAB code was developed for the optimization.

6.3.1 Chromosomes

In human genetics, chromosomes contain genes that carry the inherited cell information, and every gene codes particular protein which determines the appearance of different peculiarities. Similarly, for the genetic algorithms, the chromosome is a binary string that represents set of genes, which code the independent variables (Wright et al., 1998). Every chromosome represents a solution of the given problem. The genes could be Boolean, integers, floating point or string variables, as well as any combination of these. A set of different chromosomes (individuals) forms a generation. By means of evolutionary operators, like selection, recombination and mutation an offspring population is created (Chipperfield and Fleming, 1995).

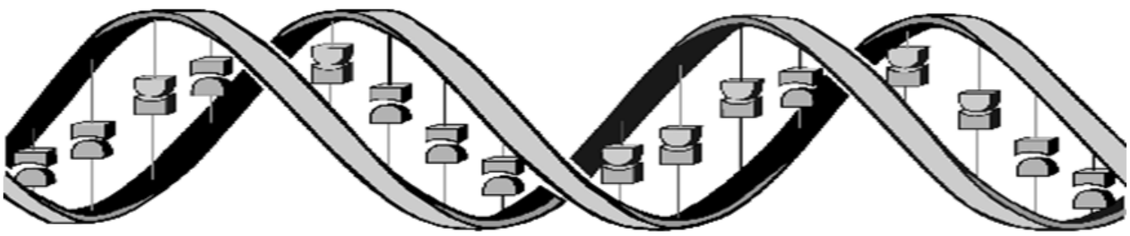

Figure 6-4: Chromosomes

An initial population is randomly created to serve as a starting point for the genetic algorithm once a suitable representation has been decided upon for the chromosomes. From empirical studies, over a wide range of function optimization problems, a population size of between 30 and 100 is usually recommended.

6.3.2 Selection

Selection is a process in which the individuals which will be applied to the genetic operations and which will create the offspring population are chosen. This process has two main purposes:

1. To choose individuals, which will take part in the generation of next population or will be directly copied (elitism);
2. To give an opportunity to individuals with comparatively lowvalue of the fitness function to take part in the creation process of the next generation. This allows us to preserve the global character of the search process and not allow a single individual to dominate the population, and thus bring it to local extremism

Just like in natural genetics, the selection is based on survival of the fittest in the genetic algorithm. It is based on the evaluation of the fitness function and the initial population is evaluated and each individual is given a figure of merit based on its

fitness as assessed by the objective function. The GA selects individual from the population to be copied into the next generation, where the probability of selection is based on the fitness of the individual. For the present optimization which deals with minimization of the total cost, the individuals with small value of the fitness function will have higher chances for recombination and consequently for generating offspring.

A popular and simple method of selection involves each chromosome being assigned a probability of selection for the next generation which is proportional to the ratio of the individual's measure of merit divided by the total of the figure of merit for the entire population and this easily implemented using an imaginary roulette wheel divided into as many segments as there are individuals in the population (Goldberg, 1989; Deb, 1999; Hariz, 2010).

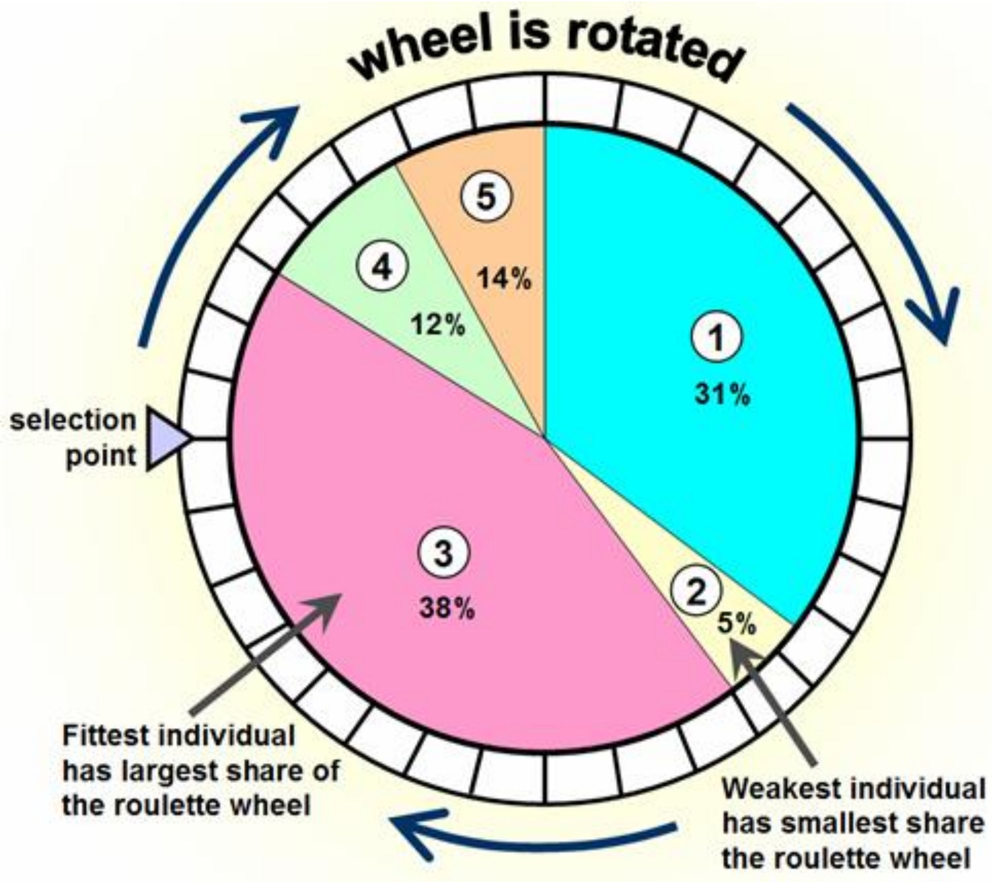

Figure 6-5: Selection process using roulette wheel for five individual chromosomes with measure of fitness of 0.31, 0.05, 0.38, 0.12 and 0.14

Each individual is allocated a size of segment proportional to its fitness measure. Figure 6-5 shows the roulette wheel selection process for five individuals.

It is obvious that although this is a random process, the probability that an individual is selected is proportional to its size on the wheel which is a function of its fitness measure.

6.3.3 Crossover

Crossover is genetic operator in genetic algorithm that combines two selected chromosomes (parents) to produce a new chromosome (offspring) with the intent that the offspring may be better than both the parents if it takes the best characteristic from each of the parents. Crossover can be done in many ways such as single point crossover, two point crossover, uniform crossover and arithmetic crossover.

6.3.4 Single Point Crossover

A crossover point is randomly selected within the chromosome (Parent X) and a binary string from beginning of the chromosome to the selected crossover point is copied for the new chromosome, the rest of the new chromosome is copied from parent Y. Figure 6-6 shows a single point crossover of parents X and Y.

Figure 6-6: Single point crossover

6.3.5 Two Points crossover

Two crossover points are randomly selected and binary string from beginning of chromosome to the first crossover points is copied from parent X. From the first crossover to the second crossover point is copied from parent Y and the last part is copied from parent x to form a new chromosome (offspring). This is depicted in Figure 6-7.

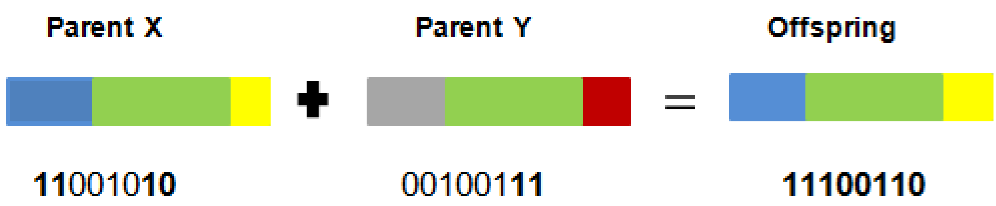

Figure 6-7: Two point crossover

6.3.6 Uniform Crossover

The offspring is formed here by randomly copying bits from parent X and Y.

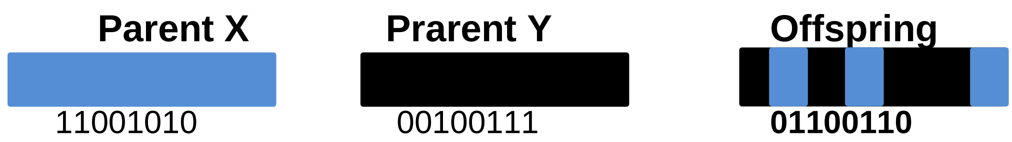

Figure 6-8: Uniform point crossover

6.3.7 Mutation

The final basic operator of GA, mutation, is achieved by randomly altering a small percentage of an individual in a population. In GA, mutation serves the crucial role of either (a) replacing the genes lost from the population during the selection process so that they can be tried in a new context or (b) providing the genes that were not present in the initial population. This is meant to keep the diversity in the population and thereby increases the likelihood that the algorithm will generate individuals with better fitness values (Kumar, 2011).

Unlike in crossover where the two parents are involved in the process, mutation is achieved only by altering one parent. This process can introduce traits which are not in the original population and prevents the GA from converging too fast, before sampling the entire search surface (Haupt and Haupt, 2004). Figure 6-9 shows an example of mutation in which the binary string 1110**0**100 having the fifth position being chosen randomly to mutate and forming a new binary string of 1110**1**100.

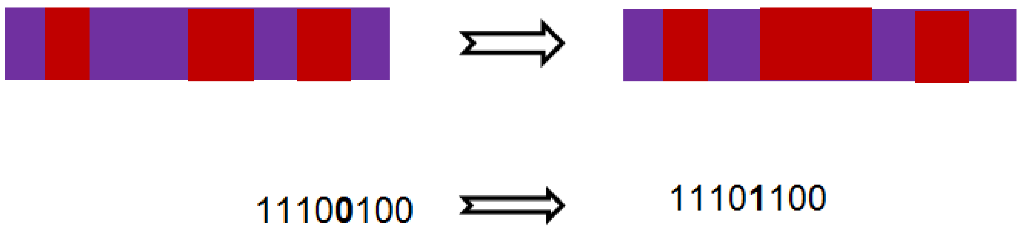

Figure 6-9: Mutation operator

6.3.8 Elitism

This involves copying the fittest chromosomes into the new population. This process ensures that the fittest chromosomes are not lost in the crossover and mutation to form new population, and consequently can vary rapidly increase the performance of GA.

In this study, a single objective is considered independently for each of the compressor drive options. The fitness value is evaluated for each individual in the generation and the outcome is used to select those individuals to be used to create the next generation.

6.3.9 Total Cost Objective Function

The first step in the implementation of a GA is the establishment of an objective function. This is a function that calculates the fitness of each member of the population or simply the function to optimize. The optimization in MATLAB is set to minimize by default but in order to maximize an objective function, a negative is simply set to the function and it is minimized. In this research, the objective function which is the total cost pertaining to the application of gas turbine as a driver for pipeline compressor is meant to be minimized. The objective function is given in equation 6.1

$$Objective\ function = C_{GTCAP} + C_{GTFuel} + C_{GTOM} + C_{GTEMIsON} + C_{COMPRE} \tag{6.1}$$

The variable parameter used here is the available gas turbine power. The parameter varies between a minimum and a maximum with the intent of finding a gas turbine power that will yield the minimum total cost with due cognisance to the constraint of maximum allowable operating pressure (MAOP) of the pipeline which dictates the maximum gas turbine power.

The objective function for the Electric motor option follows that of gas turbine except that they have different values.

$$Objective\ function = C_{EMCAP} + C_{EMkW} + C_{EMOM} + C_{COMPRE} \tag{6.2}$$

The variable used here is available electric power drive. This parameter also varies between a minimum and a maximum value with the intent of finding the electric motor drive power which will yield the least total cost.

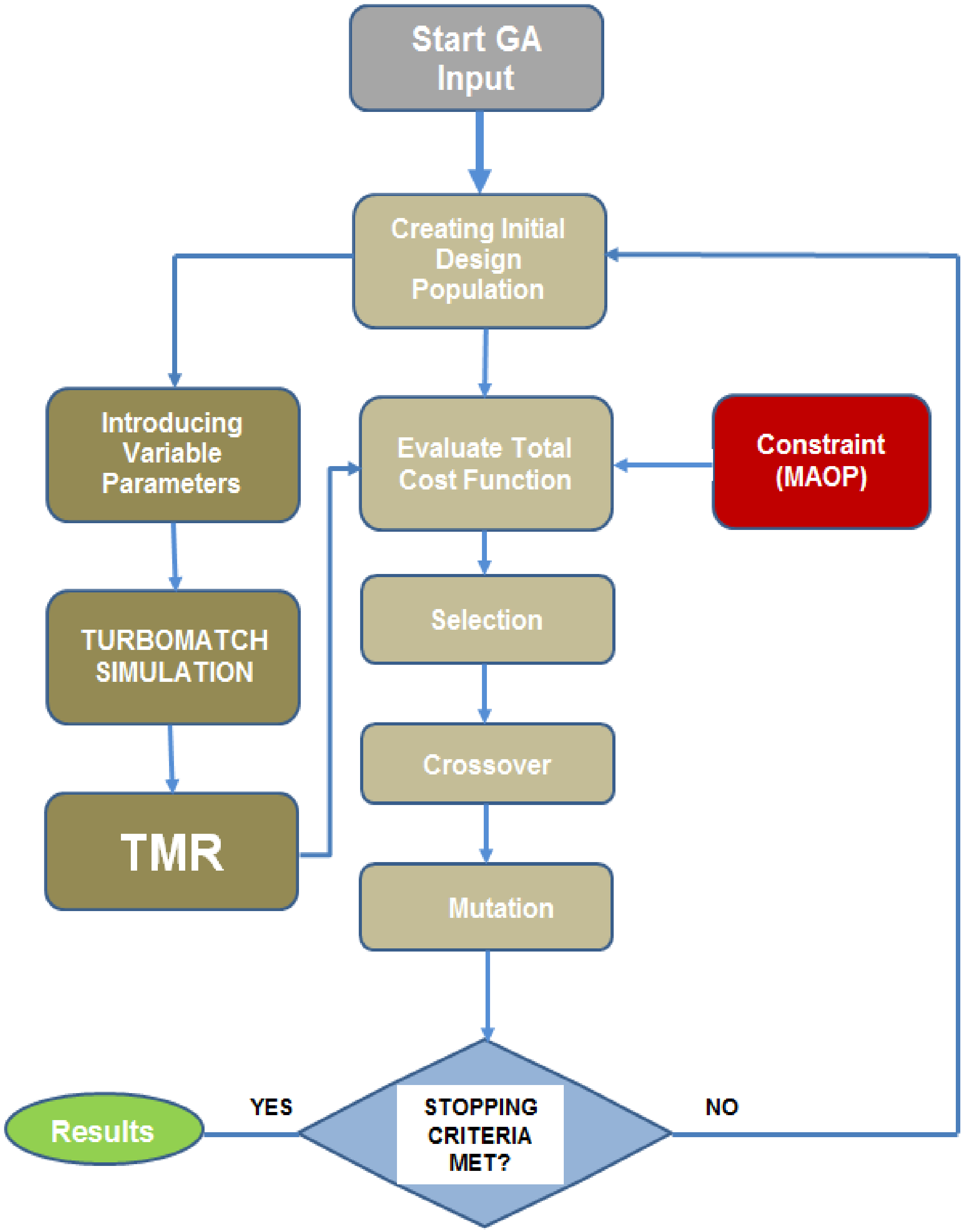

Figure 6-10: Optimization Flow Chart

6.3.10 Effect of Population Size

Population size is actually the population of solutions available for the genetic algorithm to run through in any one particular generation. Larger populations usually takes longer time to move from one generation to another but will find the optimal individual for a particular problem within few generations. Larger populations also enable the genetic algorithm to search more points and consequently obtain better result. Figure 6-11 to Figure 6-14 shows the fitness value in terms of total cost versus the number of generations for population sizes of 30, 40, 50 and 60 respectively.

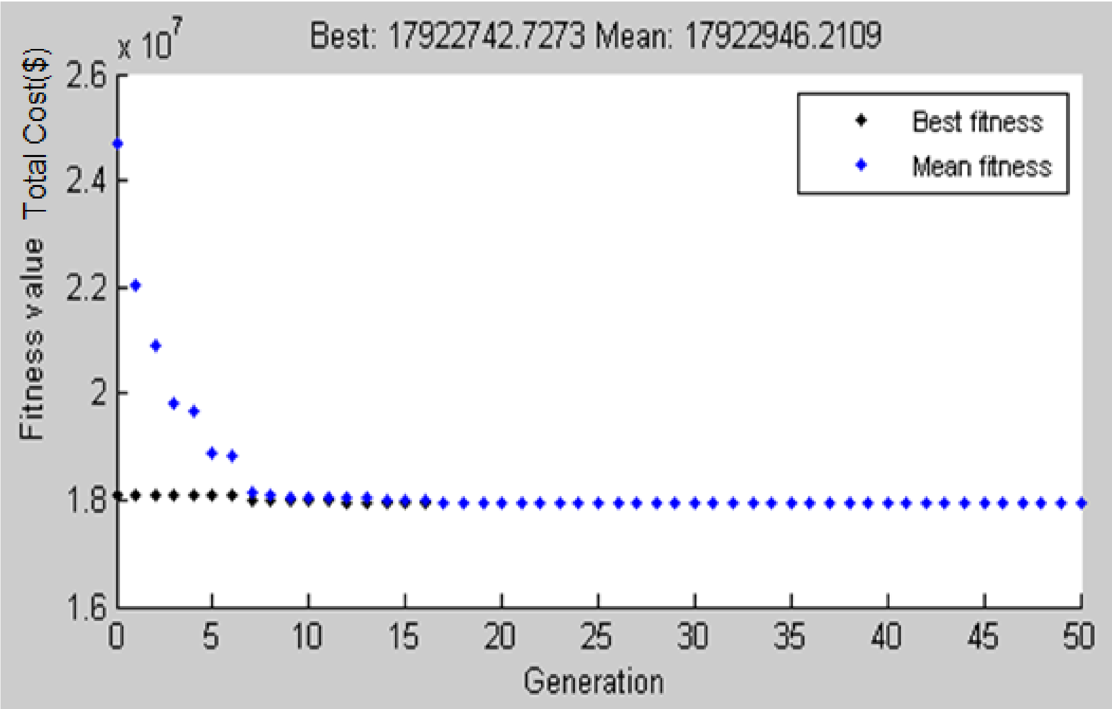

Figure 6-11: Fitness value versus Generation for Population size of 30

It is seen that as the population size increases, the number of generations to find an optimum reduces. The best fitness values which represent the least total cost in each of the runs are \$17922742.7275, \$17922742.2244, \$17922742.1613 and \$17922742.1657 for population sizes 30, 40, 50 and 60 respectively.

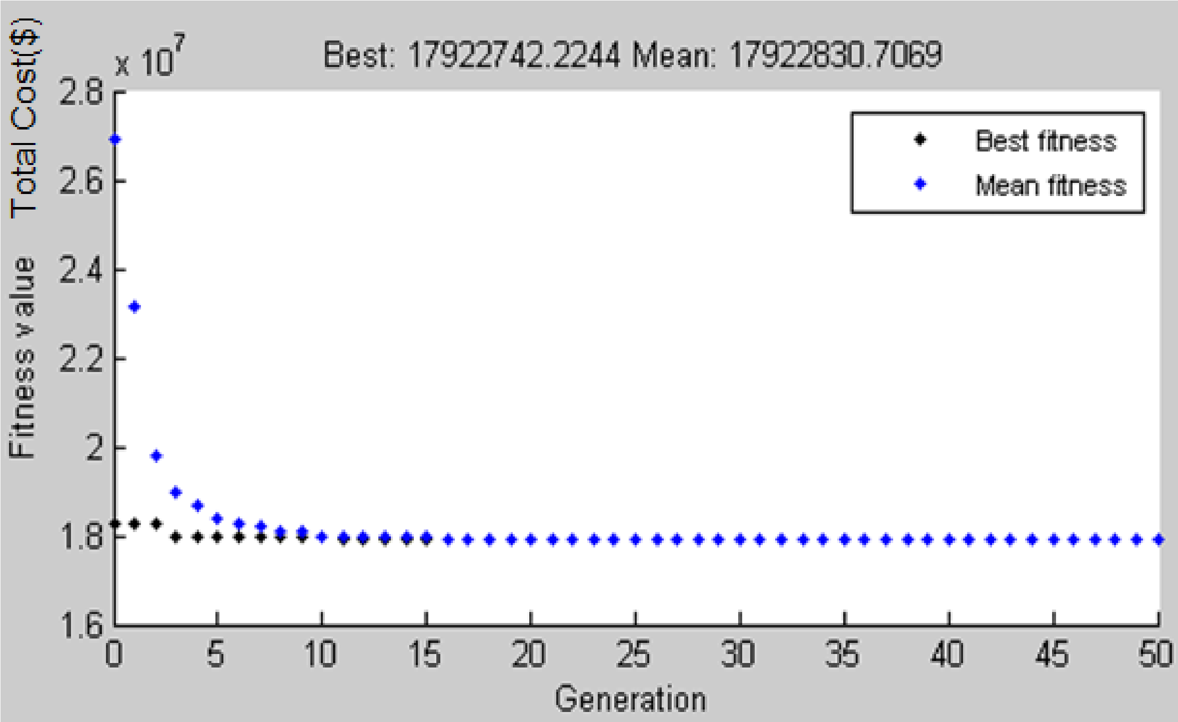

Figure 6-12: Fitness value versus Generation for Population size of 40

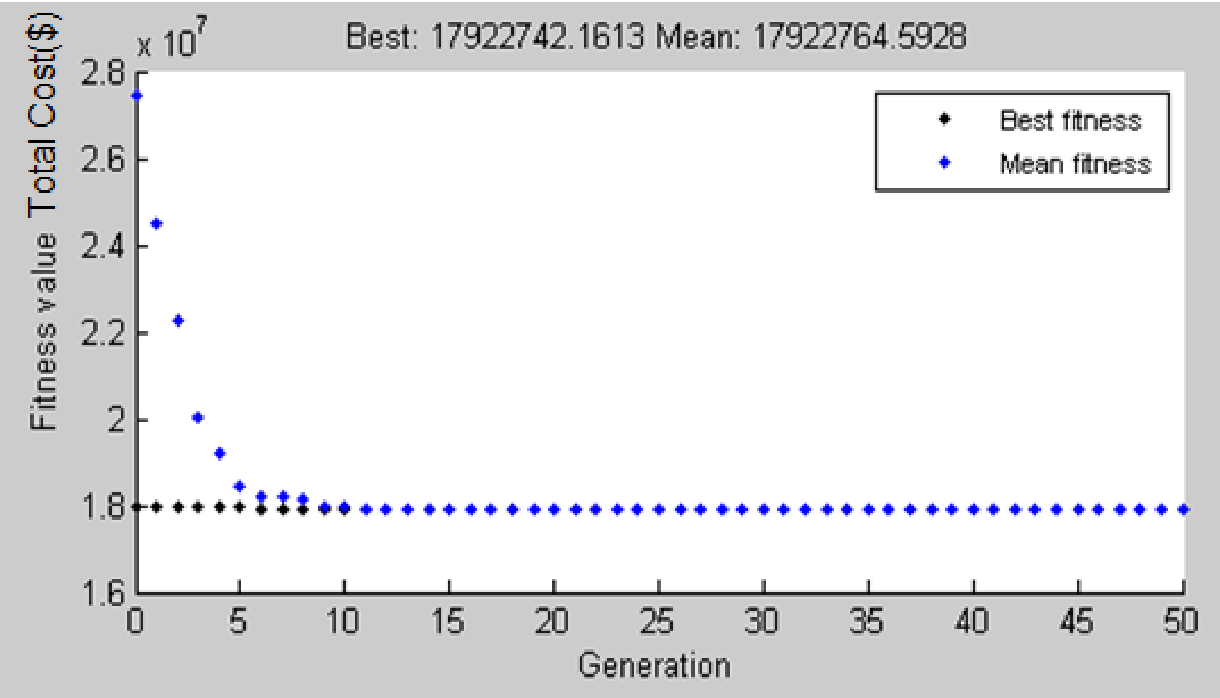

Figure 6-13: Fitness value versus Generation for Population size of 50

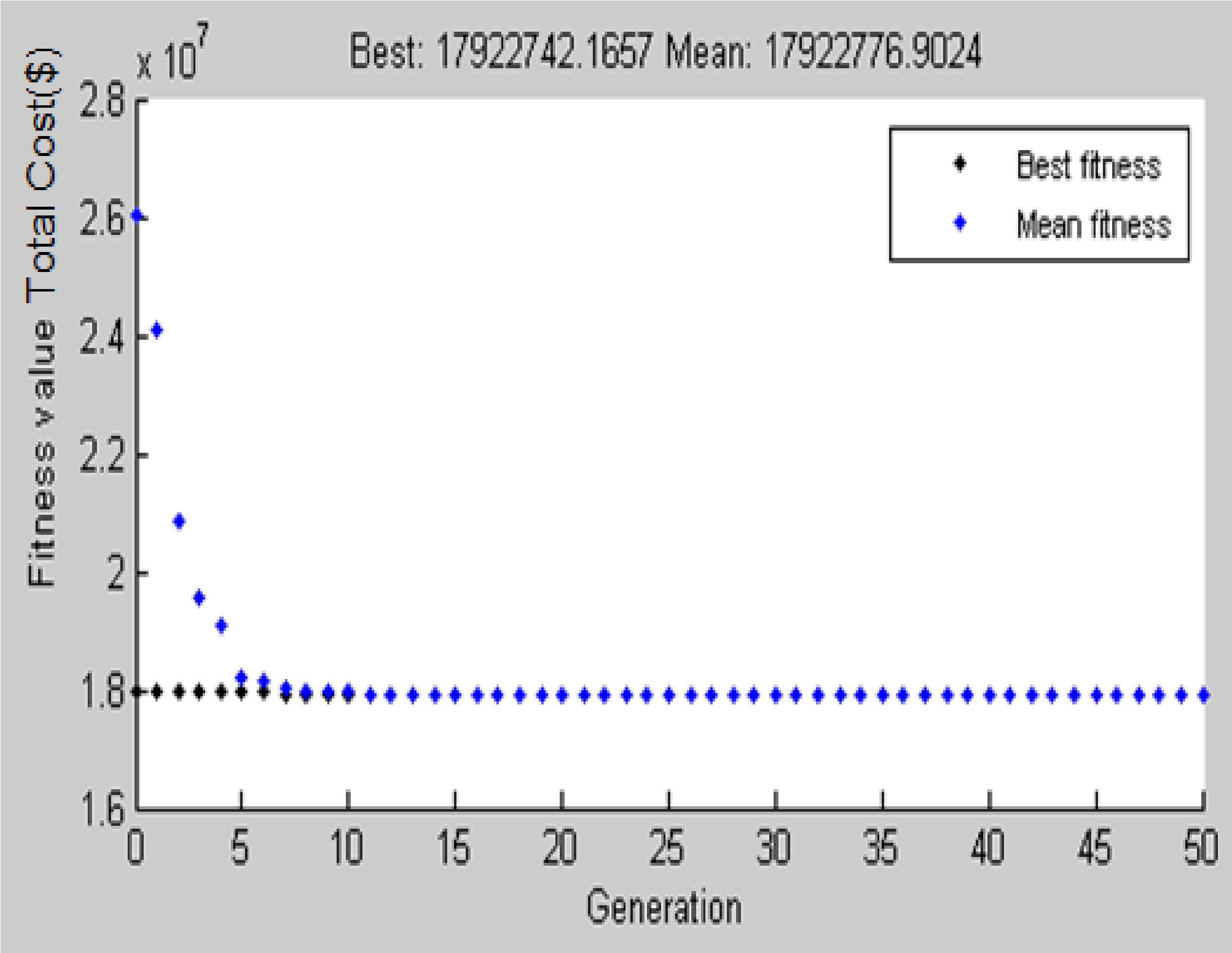

Figure 6-14: Fitness value versus Generation for Population size of 60

In all the runs, the convergence history shows a continuous line which is an indication of the existence of a smooth design space. This is because of using a GA operator termed elitism which guarantees the copying of the fittest individual into the next generation so as to ensure it is not lost as the generations progresses.

Figure 6-15 shows the graph total cost versus the drive power for the pipeline compressor. The lowest total cost of about $17.9 million is obtained when a drive power of 34MW is used. Ironically, as the drive power reduces, the total cost increases, this is a trend that may not be expected but this is depicted on Figure 6-16. For higher available drive power the fewer the number of compressor stations along the pipeline and consequently the less the capital cost, the less fuel and maintenance cost. Drive power is one of the important parameters on which the discharge pressure depends. And from Figure 6-17 shows the pressure profile along the pipeline, and that lower inlet pressure drops to the minimum pressure over a shorter distance and a compressor station is required to boost the pressure to the pressure of the natural gas in order to be able to deliver at the required discharger pressure at the city gate. In Figure 6-17, compressor stations are shown as Z_1 for 34 MW drive power, Y_1, Y_2 and Y_3 for a 24 MW drive power. For a 20 MW drive power compressor station, the station are situated at positions X_1 to X_5 as indicated on the graph.

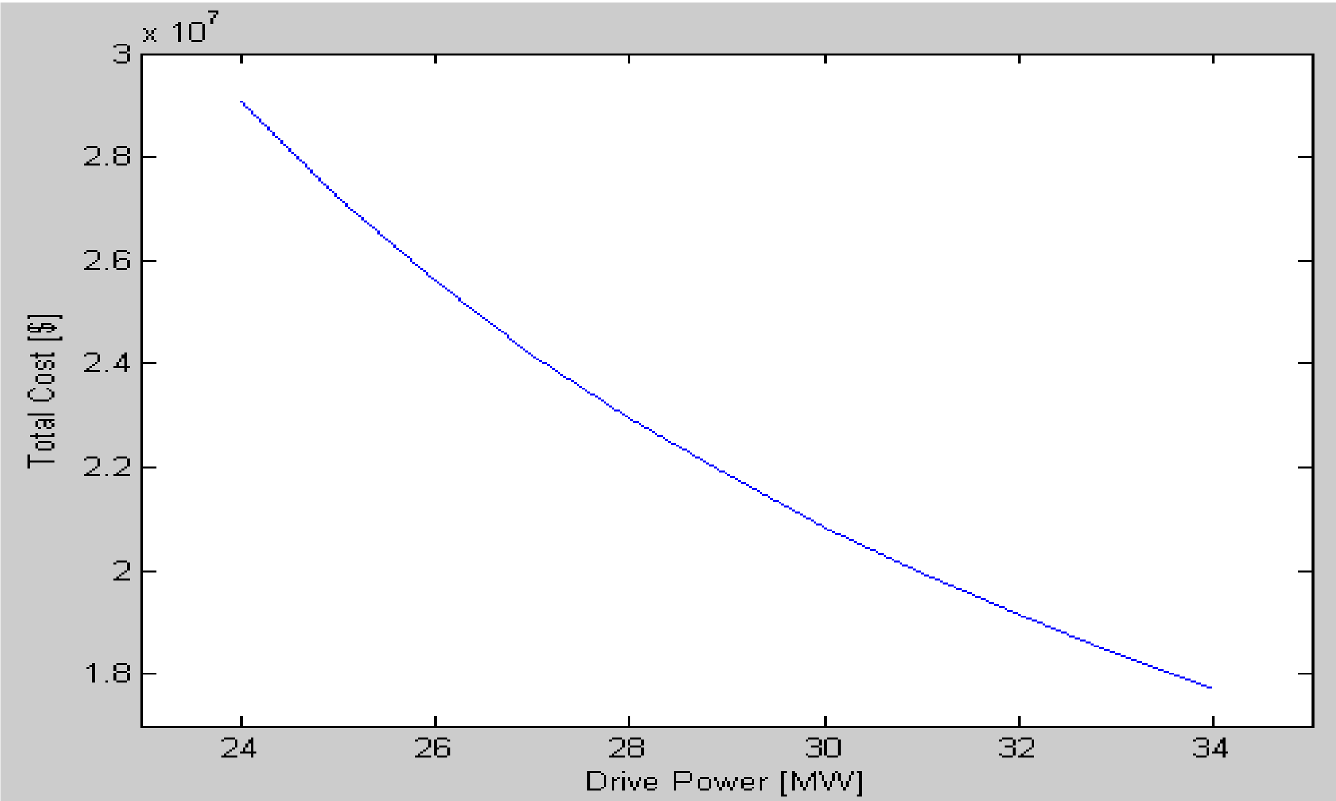

Figure 6-15: Total Cost versus Drive power

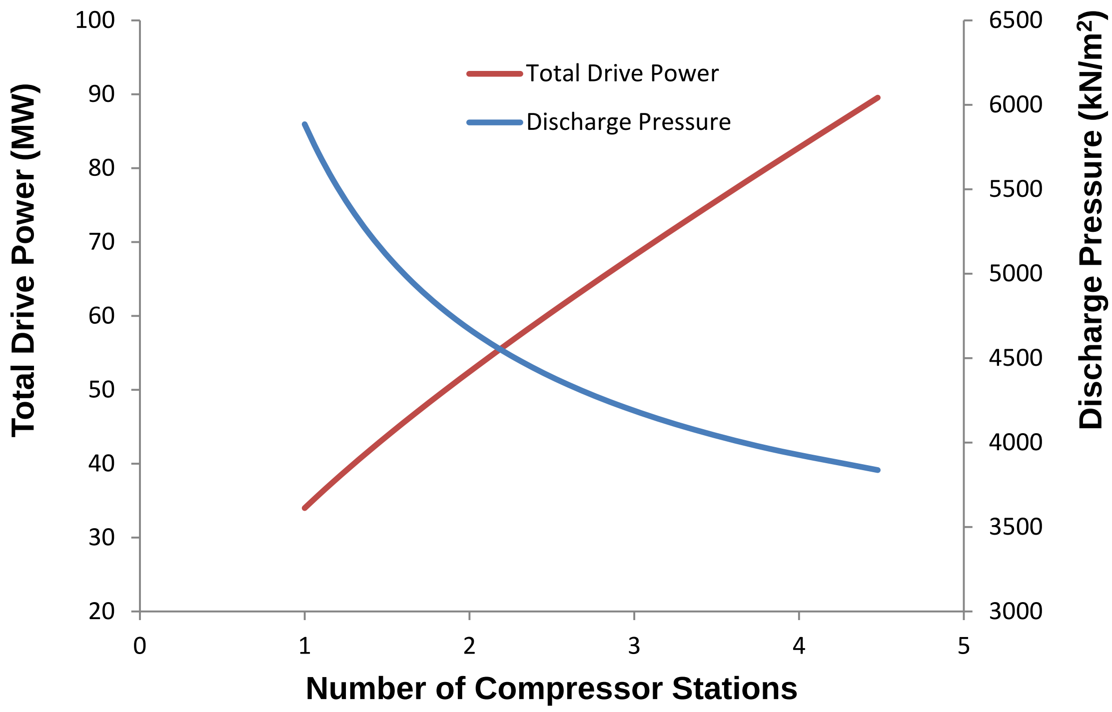

Figure 6-16: Total Drive Power and Discharge Pressure versus NCS

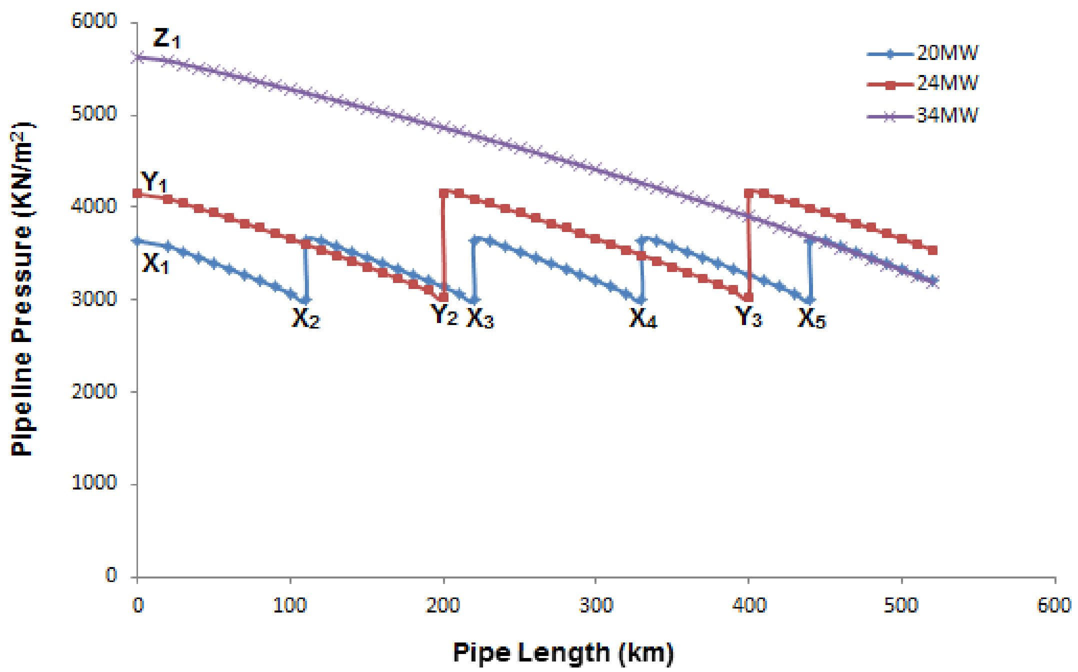

Figure 6-17: Pipeline pressure profile & compressor location for varying drive power

The optimum compressor station location along the 512 km pipeline is shown in Figure 6-17. For a drive power of 20 MW which produces a compressor station discharge pressure (pipeline inlet pressure) of 3635.2 kN/m^2 (36.4 bar) will require the first booster station of 20 MW at a position 110 km for the first compressor station to maintain hydraulic balance. For a 24 MW compressor station, the discharge pressure which is also the pipeline inlet pressure is 4144.9 kN/m^2 (41.4 bar) and employing Weymouth model for natural gas flow in pipelines, the flow pressure would drop to 3000 kN/m^2 (30 bar) at a position 200 km from the first compressor station. Another 24 MW compressor station is needed to boost the pressure from 3000 kN/m^2 to 4144.9 kN/$m^{2.}$ The discharge pressure from a 34 MW compressor station is 5626.2 kN/m^2 (56.3 bar). The pressure at the end of the entire length of the pipeline, a distance of 512 km, is 3188.7 kN/m^2 (31.9 bar). This obviously does not require another compressor station since the discharge is approximately the required discharge at the city gate. A pressure relief may be required at the city gate to drop the pressure of the natural gas to 3000 kN/m^2.

For a 34 MW, one compression station is required and for a 20 MW, five compressor stations are required in order to maintain 3000kN/m^2 (30 bar) in the pipeline otherwise an identical compressor station equipment cannot be used and this will affect the economics and stock inventory of parts. From the optimization of power plant selection based on least total cost, it is more profitable to use one large power plant than many smaller ones. Should the pipe length in Figure 6-18 be 1000 km, another compressor station will be required for a 34 MW plant to be able to deliver the gas at 30 bar.

Throughout the study, hydraulic balance was maintained and this makes it possible for all the compression equipment to be identical. This will reduce inventory of spare parts and minimize maintenance.

7 CONCLUSION AND RECOMMENDATION

This research presents a tool which can be used to establish the profitability or otherwise of a natural gas pipeline project. This chapter summarises the methods adopted and results obtained. This chapter also present some recommendation for future work in gas turbine applications in pipelines.

7.1 Summary of Methods

The methods employed in the development of all the modules that make up the TERA methodology for natural gas pipeline involves the use of TURBOMATCH, FORTRAN coding and MATLAB. The design and off-design simulation of the gas turbines was carried out by developing TURBOMATCH input files in order to establish the performance parameters of the gas turbines under varying conditions, and to note the performance parameters which are fed as inputs to other modules. Pipeline module developed in FORTRAN codes has input files specifying the designed throughput and delivery pressure employing Weymouth model equation. The output parameters from the pipeline module, such as the pipe size, gas properties and inlet pressure into the pipeline, are written to pipeline output files. Compressor module, which is a subroutine of the pipeline module, receives data from the pipeline output and calculates the compressor head. It selects a compressor based on the head and volumetric flow. The final drive power required by the compressor gives TURBOMATCH the instruction to run a particular gas turbine simulation to meet the requirement. The parameters such as fuel flow and shaft power at varying operating conditions are fed into the economic module. Gas turbine emission calculations are based on chemical equilibrium of combustion equations. This is linked with the fuel flow results from the performance simulation output. The economic module uses net present valuation methodology to establish the profitability of the pipeline system, and also computes the life cycle cost. It also presents the financial risk, based on discount rate and production life of the pipeline project. Genetic algorithm in MATLAB was used to optimize the total cost of using gas turbine as the compressor driver. Optimum compressor station position and number of compressor stations along the pipeline were also obtained based on hydraulic balance theory. The completely integrated TERA modules can serve as a tool for rapidly assessing a pipeline project.

7.2 Summary of Results

The result from the pipeline module generally shows that an increase in gas turbine shaft power is required for a reduced pipe size and increase in throughput of the natural gas. In order to maintain a specified discharge pressure of 30 bar at the city gate for the baseline pipeline system, a gas turbine which can produce a shaft power of 34 MW at the least favourable ambient temperature of 45^{o}C is required. The inlet pressure required to deliver gas at a fixed pressure reduces with increase pipe size, and consequently requiring less gas turbine shaft power. The compressor power required increases with throughput to maintain the delivery pressure, and this ultimately guides the selection of gas turbine. From the gas turbine emission calculations, 40.7 MW SCTS engine emits 1.65 x 10^5 ton of CO_2 annually which amount to \$2.6 million in emission tax.

Case Study I considered gas turbine as the pipeline compressor prime mover for the integrated TERA methodology. The operating cost of the gas turbine is seen to increase with increased throughput of natural gas. There is a saving in material cost when pipe sizes are reduced, but this savings is far less than gas turbine operating cost as a result of the reduced pipe size, and this makes pipe under-sizing economically unprofitable. The future value of the pipeline project with gas turbine as the prime mover assessed by net present valuation methodology shows an increase in NPV with increased throughput. This implies project expansion but with a fixed pipe size, but requiring higher gas turbine size and consequently more fuel and associated cost. The net present values in this scenario are still highly positive, which is a result of the economies of scale of pipeline project. The cost breakdown shows fuel cost to have the highest percentage and the life cycle cost of the baseline plant is \$1.498 billion. The natural gas transportation cost analysis which guides the selection of an economic pipe size shows 304.8 mm (12”) pipe size to be the economic pipe size for 0.5 Mm^3/day throughput, with a transportation cost of \$0.15/m^3 which is equivalent to \$3.95/GJ. Although this is the economic pipe size but the cost of transportation shows that it is uneconomical to transport 0.5 Mm^3/day of natural gas over long distance or in interstate pipelines which transport from a well-head or processing plant to a city gate where the gas is passed to distribution lines. The economic pipe size for 4.5Mm^3/day is 609.6 mm (24”) with a transportation cost of \$0.052/m^3 which is equivalent to \$1.37/GJ. This is in line with the cost of transportation in literature and from the confidential data from Nigerian National Petroleum Corporation (NNPC), which shows an average transportation cost of \$1.40/GJ over a distance of 500 km.

The results of Case Study II which considered electric motor drive as prime mover for the pipeline compressor has the same trend but slight difference in the economics compared to gas turbine drive. The economic pipe size for 0.5 Mm^3/day is 304.8 mm (12”), with a transportation cost of \$0.13/m^3 which is equivalent to \$3.4/GJ. This result equally shows that it is not economical to transport just 0.5 Mm^3/day over an interstate pipelines. The economic pipe size for a 4.5 Mm^3/day is 609.6 mm (24”) with a transportation cost of \$0.0432/m^3 which is equivalent to \$1.14/GJ. No off-site emission cost was calculated for the electric motor drive, but this is a subject for further research.

The single objective cost function optimization based on drive power available shows a convergence just before 20th generations, with the 34 MW being the best individual with the lowest cost.

7.3 Recommendation

The following are suggested areas for further research;

I. Power plant optimization with varying pipe sizes and throughput. A detailed multi-objective optimization with more variables such as pipe sizes, thickness and throughput should be carried out as this was not done in this research because of time and its complexity. The objective functions may be conflicting cost functions such as minimizing total cost and maximizing profit.

II. Comprehensive study of failures of compressors and compressor station equipment should be looked into. Detailed risk modelling of compressor station considering all the main equipment and incorporating with the economic model.

III. Detail study of the off-site emissions associated with the use of electric drive for pipeline compressor. Carrying out emission analysis of different fuels used in power generation based on the percentage of power generated from each source as it relates to the power requirements of an electric drive used for pipeline compressors.

IV. More plant simulation and increase plant library size to accommodate power plant requirements for gas gathering and distribution networks. Gas distribution networks uses pipe size range different from interstate gas transmission and would require an elaborate modelling to cater for several injections and deliveries which is associated with the system.

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Appendices

Appendix A : Research Achievements

i. American Society of Mechanical Engineers-IGTI award for outstanding PhD research in gas turbine applications in pipeline (2012)
ii. **A. Nasir, P. Pilidis, S. Ogaji & W. Mohamed (2013)** 'Some Economic Implications of Deploying Gas Turbine in Natural Gas Pipeline Networks', International Journal of Engineering and Technology, Vol. 5, No. 1, February 2013
iii. **A. Nasir, P. Pilidis, S. Ogaji & W. Mohamed (2012)** "A Study of the Effect of Gas Turbine Emissions on the Economics of Natural Gas Pipeline Transportation" Proceedings of the 20th ASME POWER International Conference on Nuclear Engineering ICONE-ASME20, July 30 - August 3, 2012, Anaheim, California, USA.
iv. **A. Nasir, P. Pilidis, S. Ogaji & W. Mohamed '** Electric drive for Pipeline compression: Techno-economic analysis' being prepared for publication
v. **A. Nasir, P. Pilidis, S. Ogaji & W. Mohamed** 'Compressor Station Optimization: Location and Cost Objective function' being prepared for publication

Appendix B : Input files

B.1 General wrapper input file

```
1           !GT/MOTOR SWITCH (1 FOR GT, 2 FOR MOTOR)
4540000.0   !Required Flow rate m^3/day-------------------------
512.0       !length of pipeline (km)
750.0       !PIPEUNIT_COST ($/ton)
110.0       !land cost (£/m^2)
```

```
7.0             !Gas Price (£/m^3)
0.9             !Compressibility factor (Z)
19.5            !Molecular weight
858.0           !Inlet pressure into 1st compressor(p1) (kN/m^2)
2312.0                  !Discharge pressure from 1st compressor P2 (kN/m^2)
2250.0                  !Inlet pressure to the 2nd compressor P3 (kN/m^2)
5900.0                  !Discharge pressure from 2nd compressor P4 (kN/m^2)
1.4             !POLYTROPIC EXPONENT n
278.15                  !Inlet gas temperature T1 (K)
0.84            !Efficiency 1st Compressor (Eff1)
0.8             !Efficiency 2nd Compressor (Eff2)
9.52            !Pipe Thickness t (m)
15.0            !Cost of Coating, wrapping and delivery per m ($/m) CWD_Cost
198000                  !Unit Installation cost per km
0.0             !E_Cost_Factor (0.01-0.05) depending on number of roads,
highway and rivers to cross
1000000.0       !Comp1_cost ($)
1000000.0       !comp2_cost ($)
0.95            !Pipeline efficiency E_P
288.0           !Base temperature (K)
101.0           !Base pressure (kPa)
1.08            !Pipe elevation factor
0.65            !Gas gravity
293.0           !Gas flowing temperature (K)
5900.0                  !P1p pipeline inlet pressure
3000.0                  !P2p pipeline discharge pressure
10              !COMPSTATION_FACTOR (REPRESENTING THE COST OF
OTHER EQUIPMENTS IN THE COMPRESSOR STATION
350             !Cost of GT per kW GT_COST
2.6457                !Fuel Flow kg/s 1.982-------4.8918, 1.4697, -----------------------
43000           !Calorific value of Fuel(kJ/m3)
1.022640        !Volume correction factor
0.9             !Fuel Density F_Density, kg/m
8760            !Operating hour per year
5.0             !Prize of Gas $/GJ--------------------------------------
30              !GT Maintenance factor Mtce_Factor$/0.746kWh/year
117.4           !Cost of electric motor per $ kW (Motcost_kw)
34000           ! Electric Motor Power rating P_rating (kW)
0.05            !Cost of electricity $ per kWh (E_cost_kwh)
6600            !Voltage V
2882            !Current
0.1             !DISCOUNT RATE, D_RATE-------------------------------------
100000          !Cost per Main valve station(MVS) $
1340            !COMPRESSOR STATION COST $ PER kW(COMP STATION
        COST    FACTOR)(1340)
0.98            !Electric motor efficiency at 100% Load (FULL LOAD) MTOR_EFF
300000.0        !Cost per MSR ($) Cost_Per_MVR
20              !Production Life (PRD_LIFE)
```

15 !Electric Motor maintenance factor ($/0.746kW/year)

DEGRADATION FACTOR

YEAR	DEGRADATION FACTOR
1	1.005
2	1.016055
3	1.0313
4	1.0488
5	1.0698
6	1.091
7	1.113
8	1.133
9	1.158
10	1.181
11	1.204
12	1.228
13	1.253
14	1.278
15	1.304
16	1.330
17	1.356
18	1.383
19	1.411
20	1.439
21	1.467
22	1.494
23	1.522
24	1.547
25	1.576
26	1.605
27	1.634
28	1.662
29	1.691
30	1.720

B.2 TURBOMATCH input files

Below is the FORTRAN programmed instructions for TURBOMATCH which simulates the design and off-design thermodynamic performance of gas turbines and the results are written to a separated TMR file.

```
! INDUSTRIAL GAS TURBINE SIMULATION
GENERAL ELECTRIC LM 6
MODELLED BY ABDULKARIM NASIR, ////
OD SI KE CT FP
-1
-1
INTAKE  S1-2      D1-4            R300
```

```
COMPRE  S2-3      D5-10            R301   V5  V6
PREMAS  S3-4,13   D12-15
COMPRE  S4-5      D16-21           R302   V16 V17
PREMAS  S5-6,14   D23-26
BURNER  S6-7      D27-29           R303
MIXEES  S7,14,8
TURBIN  S8-9      D30-37,302              V31
MIXEES  S9,13,10
TURBIN  S10-11    D39-46,301,47           V39 V40
NOZCON  S11-12,1  D48              R304
PERFOR  S1,0,0    D39,49-51,304,300,303,0,0,0,0,0,0,0,0,0
CODEND
DATA ITEMS
////
1 0.0         !INTAKE ALTITUDE
2 0.0         !ISA DEVIATION
3 0.0         !MACH NUMBER
4 0.9951      !PRESSURE RECOVERY
!LP COMPRESSOR
5 0.8         !Z PARAMETER
6 0.85        !ROTATIONAL SPEED N
7 2.45        !PRESSURE RATIO
8 0.875       !ISENTROPIC EFFICIENCY
9 1.0         !ERROR SELECTION
10 5.0        !MAP NUMBER
!PREMAS
12 1.0        !BYPASS RATIO
13 0.0        !MASS FLOW LOSS
14 1.0        !PRESSURE RECOVERY
15 0.0        !PRESSURE LOSS
!HP COMPRESSOR
16 0.8        !SURGE MARGIN
17 -1.0       !SPOOL SPEED
18 12.25      !PRESSURE RATIO
19 0.875      !EFFICIENCY
20 1.0        !ERROR SELECTOR
21 5.0        !COMPRESSOR MAP NUMBER
!PREMAS
23 0.85       !BY PASS RATIO
24 1.80       !MASS FLOW LOSS
25 1.0        !PRESSURE FACTOR
26 0.0        !PRESSURE LOSS
!BURNER
27 0.075      !FRACTUAL PRESSURE LOSS
28 1.0        !COMBUSTION EFFICIENCY
29 -1.0       !FUEL FLOW
!HP TURBINE
30 0.0        !AUXILIARY POWER REQUIRED
```

```
31 0.8        !NON-DIMENSIONAL MASSFLOW
32 0.6         !NON-DIMENSIONAL SPEED
33 0.885       !EFFICIENCY
34 -1.0       !COMPRESSOR TURBINE
35 2.0        !COMPRESSOR NUMBER
36 5.0        !TURBINE MAP NUMBER
37 1000       !POWER LOW INDEX
!IP TURBINE
39 40725000.00 !AUXILIARY POWER REQUIRED
40 0.85       !NON-DIMENSIONAL MASS FLOW
41 -1.0       !NON-DIMENSIONAL SPEED
42 0.885       !EFFICIENCY
43 -1.0       !COMPRESSOR TURBINE
44 1.0        !COMPRESSOR NUMBER
45 5.0        !TURBINE MAP NUMBER
46 -1.0       !POWER LOW INDEX
47 0.0        !
!CONVERGENT NOZZLE
48 -1.0       !AIR FIXED
!PERFORMANCE
49 1.00       !PROPELLER EFFICIENCY
50 0.0        !SCALING INDEX (0=NO SCALING)
51 0.0        !REQUIRED THRUST
-1
1 2 126.6     !INLET MASS FLOW
7 6 1540.00    !COMBUSTION OUTLET TEMPERATURE
-1
2  -5.0
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
```

```
7  6  1670.0         ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  5.0
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0         ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0         ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0         ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0         ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0         ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0         ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  6.0
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0         ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0         ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0         ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0         ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0         ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0         ! OD Calculation; DT=0;  TET = 1500.0K
-1
```

```
2  7.0
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  9.0
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  10.0
-1
```

```
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2    11.0   ! --New OD Calculation; DT=-5.0; TET = 1500.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  13.0  ! --New OD Calculation; DT=-10; TET = 1000.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
```

```
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  14.0   ! --New OD Calculation; DT=5;  TET = 1500.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  15.0   ! --New OD Calculation; DT=5;  TET = 1500.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
```

```
7  6  1520.0         ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0         ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0         ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0         ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0         ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0         ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  17.0   ! --New OD Calculation; DT=5;  TET = 1500.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0         ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0         ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0         ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0         ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0         ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0         ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  19.0   ! --New OD Calculation; DT=5;  TET = 1500.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0         ! OD Calculation; DT=0; TET = 1100.0K
-1
```

```
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  21.0   ! --New OD Calculation; DT=10;  TET = 1000.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  23.0   ! --New OD Calculation; DT=10;  TET = 1000.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
```

```
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  24.0   ! --New OD Calculation; DT=10;  TET = 1000.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  25.0   ! --New OD Calculation; DT=10;  TET = 1000.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
```

```
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  26.0   ! --New OD Calculation; DT=10;  TET = 1000.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  27.0   ! --New OD Calculation; DT=10;  TET = 1000.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
```

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-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  30.0   ! --New OD Calculation; DT=10;  TET = 1000.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  34.0   ! --New OD Calculation; DT=10;  TET = 1000.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
```

```
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  38.0   ! --New OD Calculation; DT=10;  TET = 1000.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  39.0   ! --New OD Calculation; DT=10;  TET = 1000.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
```

```
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1670.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
2  45.0   ! --New OD Calculation; DT=10;  TET = 1000.0K
-1
7  6  1480.0 ! --New OD Calculation; DT=TAMB(STANDARD)-
TAMB(ACTUAL)=0;  TET = 1000.0K
-1
-1
7  6  1520.0        ! OD Calculation; DT=0; TET = 1100.0K
-1
-1
7  6  1550.0        ! OD Calculation; DT=0; TET = 1200.0K
-1
-1
7  6  1580.0        ! OD Calculation; DT=0; TET = 1300.0K
-1
-1
7  6  1610.0        ! OD Calculation; DT=0; TET = 1400.0K
-1
-1
7  6  1640.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
7  6  1710.0        ! OD Calculation; DT=0;  TET = 1500.0K
-1
-1
-3
```

B.3 Pipe data

Pipe Dimensions Imperial/Metric Pipe Chart

Nominal Pipe Size Inches	Nominal Pipe Size mm	OD Inches	OD mm	Schedule Designations ANSI/ASME	Wall Thickness Inches	Wall Thickness mm	Lbs/Ft	Kg/M
12	300	12.75	323.9	5S	0.156	3.96	20.980	31.22
12	300	12.75	323.9	10S	0.180	4.57	24.200	36.02
12	300	12.75	323.9	20	0.250	6.35	33.380	49.68
12	300	12.75	323.9	30	0.330	8.38	43.770	65.14
12	300	12.75	323.9	STD/40S	0.375	9.53	49.560	73.76
12	300	12.75	323.9	40	0.406	10.31	53.520	79.65
12	300	12.75	323.9	XS/80S	0.500	12.70	65.420	97.36
12	300	12.75	323.9	60	0.562	14.27	73.150	108.87
12	300	12.75	323.9	80	0.688	17.48	88.630	131.90
12	300	12.75	323.9	100	0.844	21.44	107.320	159.72
12	300	12.75	323.9	120/XX	1.000	25.40	125.490	186.76
12	300	12.75	323.9	140	1.125	28.58	139.670	207.86
12	300	12.75	323.9	160	1.312	33.32	160.270	238.52
14	350	14	355.6	10S	0.188	4.78	27.73	41.27
14	350	14	355.6	10	0.250	6.35	36.71	54.63
14	350	14	355.6	20	0.312	7.92	45.61	67.88
14	350	14	355.6	STD/30/40S	0.375	9.53	54.57	81.21
14	350	14	355.6	40	0.438	11.13	63.44	94.41
14	350	14	355.6	XS/80S	0.500	12.70	72.09	107.29
14	350	14	355.6	60	0.594	15.09	85.05	126.58
14	350	14	355.6	80	0.750	19.05	106.13	157.95
14	350	14	355.6	100	0.938	23.83	130.85	194.74
14	350	14	355.6	120	1.094	27.79	150.90	224.58
14	350	14	355.6	140	1.250	31.75	170.21	253.32
14	350	14	355.6	160	1.406	35.71	189.10	281.43
16	400	16	406.4	10S	0.188	4.78	31.75	47.25
16	400	16	406.4	10	0.250	6.35	42.05	62.58

Nominal Pipe Size Inches	Nominal Pipe Size mm	OD Inches	OD mm	Schedule Designations ANSI/ASME	Wall Thickness Inches	Wall Thickness mm	Lbs/Ft	Kg/M
16	400	16	406.4	STD/30/40S	0.375	9.53	62.58	93.13
16	400	16	406.4	XS/40/80S	0.500	12.70	82.77	123.18
16	400	16	406.4	60	0.656	16.66	107.50	159.99
16	400	16	406.4	80	0.844	21.44	136.61	203.31
16	400	16	406.4	100	1.031	26.20	164.82	245.29
16	400	16	406.4	120	1.219	30.96	192.43	286.38
16	400	16	406.4	140	1.438	36.53	223.64	332.83
16	400	16	406.4	160	1.594	40.49	245.25	364.99
18	450	18	457.2	10S	0.188	4.78	35.76	53.22
18	450	18	457.2	10	0.250	6.35	47.39	70.53
18	450	18	457.2	20	0.312	7.92	58.94	87.72
18	450	18	457.2	STD/40S	0.375	9.53	70.59	105.06
18	450	18	457.2	30	0.438	11.13	82.15	122.26
18	450	18	457.2	XS/80S	0.500	12.70	93.45	139.08
18	450	18	457.2	40	0.562	14.27	104.67	155.78
18	450	18	457.2	60	0.750	19.05	138.17	205.63
18	450	18	457.2	80	0.938	23.83	170.92	254.37
18	450	18	457.2	100	1.156	29.36	207.96	309.50
18	450	18	457.2	120	1.375	34.93	244.14	363.34
18	450	18	457.2	140	1.562	39.67	274.22	408.11
18	450	18	457.2	160	1.781	45.24	308.5	459.13
20	500	20	508	10S	0.218	5.54	46.06	68.55
20	500	20	508	10	0.250	6.35	52.73	78.48
20	500	20	508	STD/20/40S	0.375	9.53	78.60	116.98
20	500	20	508	XS/30/80S	0.500	12.70	104.13	154.97
20	500	20	508	40	0.594	15.09	123.11	183.22
20	500	20	508	60	0.812	20.62	166.40	247.65
20	500	20	508	80	1.031	26.19	208.87	310.85
20	500	20	508	100	1.281	32.54	256.10	381.14
20	500	20	508	120	1.500	38.10	296.37	441.07
20	500	20	508	140	1.750	44.45	341.09	507.63

Nominal Pipe Size Inches	Nominal Pipe Size mm	OD Inches	OD mm	Schedule Designations ANSI/ASME	Wall Thickness Inches	Wall Thickness mm	Lbs/Ft	Kg/M
24	600	24	609.6	10/10S	0.250	6.35	63.41	94.37
24	600	24	609.6	STD/20/40S	0.375	9.53	94.62	140.82
24	600	24	609.6	XS/80S	0.500	12.70	125.49	186.76
24	600	24	609.6	30	0.562	14.27	140.68	209.37
24	600	24	609.6	40	0.688	17.48	171.29	254.92
24	600	24	609.6	60	0.969	24.61	238.35	354.72
24	600	24	609.6	80	1.219	30.96	296.58	441.39
24	600	24	609.6	100	1.531	38.89	367.39	546.77
24	600	24	609.6	120	1.812	46.02	429.39	639.04
24	600	24	609.6	140	2.062	52.37	483.10	718.97
24	600	24	609.6	160	2.344	59.54	542.13	806.83
30	750	30	762	10	0.312	7.92	98.93	147.23
30	750	30	762	STD/40S	0.375	9.53	118.65	176.58
30	750	30	762	XS/20/80S	0.500	12.70	157.53	234.44
30	750	30	762	30	0.625	15.88	196.08	291.82
36	900	36	914.4	10	0.312	7.92	118.92	176.98
36	900	36	914.4	STD/40S	0.375	9.53	142.68	212.34
36	900	36	914.4	XS/80S	0.500	12.70	189.57	282.13

Appendix C : FORTRAN and MATLAB codes

C.1 FORTRAN codes

```
Program Pipeline
Implicit None
!PIPELINE TECHNO-ECONOMIC MODEL WITH SUBROUTINE HANDLING
COMPRESSOR STATIONS AND ELECTRIC MOTORS
!*******************************************************************************************
****
!DECLARATION OF VARIABLES
REAL,DIMENSION(:), ALLOCATABLE :: Table_a (:,:),DEG_A(:)
REAL,DIMENSION(:), ALLOCATABLE :: Table_b (:,:),Q_Flow(:)
REAL,DIMENSION(:), ALLOCATABLE :: Table_M(:,:)
Real::
PIPELINE_SYS_CAPCOST,Cost_Per_MSR,MSR,MSR_Cost,TOTAL_GT_CAP
COST,PRD_LIFE,EM_MTCE_Factor,Q_Flow
```

```
Real:: Q,D,Pi, L, Area, Volume, PIPEUNIT_COST, COST_LAND,
COST_GAS,FF, M_WEIGHT, t, Pipe_weight, Total_Pipeweight, pipe_cost
Real:: P1, P2, P3, P4, n, W1, W2, T1, Eff1, Eff2, Z, COMP1_WORK,
COMP1_POWER, COMP2_WORK, COMP2_POWER,
Total_Power,DRIVER_POWER
Real:: Total_Pipecost,
Pipeline_Installation_Cost,Pipeline_Capitalcost,CWD_Cost,Extra_cost,E_Cost_
Factor,ROW,CompStatncost
Real:: Total_compcost, Comp1_cost,
comp2_cost,Tb,e,G,Tf,PB,P1p,P2p,P_DIFF,DENOM1,P_DENOM,De,UNIT_IN
STALATIONCOST,F_Density,Rev_Total
Real::
DENOM11,T_B,T_B_E,E_P,Din,COMPSTATION_FACTOR,TOTAL_COMPST
ATION_COST,GTCOST_PKW,GT_CAPCOST,M_Fuel,CV_Fuel,VC_Factor
Real::
O_Time,GTEnergyCost_PA,Energy_PA,Gas_Price,F_Energy,F_Vol,Energy_To
tal,GTTotal_E_Cost,Annual_Mtce_Cost,Motcost_kw
Real:: P_rating,
I,V,M_OPhourPA,M_Ophour,Total_Projectcost_GT,M_Total_opcost,REV_PA,G
AS_E_SOLD,Q_E,D_RATE,Disct_F,LOAN,EQUITY
Real::
Annual_RningcostGT,COST_PER_MVS,MVS_Cost,MVS,CSCF,TOTAL_COM
PST_COST,SCADA_COST,TOTALCAP_COST_GT,TOTALCAP_COST_EM,c
apcost
Integer::YR,j,GT_MOT_SWITCH,YEAR,DTR,k,s
Real::
Total_Projectcost_Emotor,EMotor_TotcostPA,Cost_Emotor,M_MntceCostPA,M
_Op_cost_PA,TOTAL_E_COST,DSCNT_OP_COST_MOTOR,MTOR_Total
Real::
E_ENERGY_PA,E_ENERGY,GT_TOTAL_COST,GTMTCE_PA,P_TOTAL_CO
ST,E_COST_KWH,MTCE_FACTOR,TOTAL_REV,DSCNT_OP_COST_GT,GT
_Total
Real::
GT_PV_YR,GT_NET_CFL,GT_CASH_OF,CASH_INF,OP_COST_MOTOR,GA
S_REV_PA,GT_NPV,GT_OPCOST_PA,TOTAL_CAPCOST,M_PV_YR,M_NET
_CFL,M_CASH_OF
Real::
M_CASH_OF,M_NPV,LOAN_P,GT_NPV,CASH_INF_T,DT,GT_CASH_OF_T,
GAS_ENERGY_PA,TAX,GT_NET_CFL_ATAX,GT_CAPCOST,MTOR_CASH_
OF_T,MTOR_NPV
Real::
MTOR_PV_YR,MTOR_NET_CFL_ATAX,MTOR_NET_CFL,MTOR_CASH_OF,
MTOR_CAPCOST,GAS_E,MTOR_EFF,MTOR_OPCOST_PA,MTOR_E_PA,Y,
P2D,Y1,Y2,Y3,Y4
Real:: QT,QT1,DT1,Mass_Fuel, kg_CO2, CO2_cost_YR, CO2_tax,
GT_Op_Cost_30, GT_Mtce_cost_30, GT_CO2_Tax_30, GT_OP_COST_30_T
Real:: GT_Mtce_cost_30_T, GT_CO2_T, GT_CO2_COST_YR_T,
GT_Op_Tax_30_T,  GT_CO2_TAX_YR_T, GT_CO2_TAX_30_T
```

```
Real:: LL, AA, AB, AC, AA_T, AB_T, AC_T, AAM,ABM,AAM_T,ABM_T
CHARACTER*256 :: DUMMY
Pi=3.142
!****************************************************************************************
****
!OPENING THE I/O FILES

OPEN (UNIT=11, FILE= 'Input.dat',STATUS= 'OLD', ACTION= 'READ')
OPEN (UNIT=12, FILE= 'Pipe_Output.dat',STATUS= 'REPLACE', ACTION=
'WRITE')
OPEN (UNIT=13, FILE= 'engine_tabular.dat',STATUS= 'OLD', ACTION=
'READ')
OPEN (UNIT=14, FILE= 'Comp_Out.dat',STATUS= 'OLD', ACTION= 'READ')

!****************************************************************************************
****
!READING VARIABLES FROM MAIN INPUT FILES

Read (11,*) DUMMY
READ (11,*) GT_MOT_SWITCH
READ (11,*) Q
Read (11,*) L
Read (11,*) PIPEUNIT_COST
Read (11,*) COST_LAND
Read (11,*) COST_GAS
Read (11,*) Z
Read (11,*) M_WEIGHT
Read (11,*) P1
Read (11,*) P2
Read (11,*) P3
Read (11,*) P4
Read (11,*) n
Read (11,*) T1
Read (11,*) Eff1
Read (11,*) Eff2
Read (11,*) t
Read (11,*) CWD_Cost
Read (11,*) UNIT_INSTALATIONCOST
Read (11,*) E_Cost_Factor
Read (11,*) Comp1_cost
Read (11,*) comp2_cost
Read (11,*) E_P
Read (11,*) Tb
Read (11,*) Pb
Read (11,*) e
Read (11,*) G
Read (11,*) Tf
Read (11,*) P1p
```

```
Read (11,*) P2p
Read (11,*) COMPSTATION_FACTOR
Read (11,*) GTCOST_PKW
Read (11,*) M_Fuel
Read (11,*) CV_Fuel
Read (11,*) VC_Factor
Read (11,*) F_Density
Read (11,*) O_Time
Read (11,*) Gas_Price
Read (11,*) Mtce_Factor
Read (11,*) Motcost_kw
Read (11,*) P_rating
Read (11,*) E_cost_kwh
Read (11,*) V
ReaD (11,*) I
READ (11,*) D_RATE
READ (11,*) COST_PER_MVS
READ (11,*) CSCF
READ (11,*) MTOR_EFF
READ (11,*) Cost_Per_MSR
READ (11,*) PRD_LIFE
READ (11,*) EM_MTCE_Factor
ALLOCATE(DEG_A(1:30))

READ (11,*)DUMMY
  DO j=1,30
    READ (11,*) DUMMY, DEG_A(j)
   ! write (*,*) DEG_A(j),'     DEG_A(j)'
  END DO
!ALLOCATE(Q_Flow(1:10))
!READ (11,*)DUMMY
 ! DO k=1,10
  !  READ (11,*) DUMMY, Q_Flow(k)
  write (*,*) COMPSTATION_FACTOR,'     COMPSTATION_FACTOR'
  !END DO
DO s=1,10
  Q=Q_Flow(s)
write (12,*)'This table is for Q=',Q
!*******************************************************************************************
********THE big LOOP

!*******************************************************************************************
***
!PIPELINE CALCULATIONS
P_DIFF=(P1p**2)-(e*(P2p**2))
DENOM1=(G*Tf*Z*L)
P_DENOM=SQRT(P_DIFF/DENOM1)
T_B=Tb/Pb
```

```
T_B_E=T_B*E_P
DENOM11=0.0037435*T_B_E
De=Q/(DENOM11*P_DENOM)
D=De**0.375
write (*,*) D,'      Pipe Diameter  mm'
Din=D/25.4
write (*,*) Din,'      Pipe Diameter  inches'
IF (Din >=9.5 .AND. Din<=10.0)Write (12,*) 'Din inch= 10 INCH', Din
IF (Din >10.0 .AND. Din<=12.0)Write (12,*) 'Din inch= 12 INCH', Din
IF (Din >12.0 .AND. Din<=14.0)Write (12,*) 'Din inch= 14 INCH', Din
IF (Din >14.0 .AND. Din<=16.0)Write (12,*) 'Din inch= 16 INCH', Din
IF (Din >16.0 .AND. Din<=18.0)Write (12,*) 'Din inch= 18 INCH', Din
IF (Din >18.0 .AND. Din<=20.0)Write (12,*) 'Din inch= 20 INCH', Din
IF (Din >20.0 .AND. Din<=24.0)Write (12,*) 'Din inch= 24 INCH', Din
IF (Din >24.0 .AND. Din<=30.0)Write (12,*) 'Din inch= 30 INCH', Din
IF (Din >30.0 .AND. Din<=36.0)Write (12,*) 'Din inch= 36 INCH', Din
IF (Din >36.0 .AND. Din<=40.0)Write (12,*) 'Din inch= 40 INCH', Din
IF (Din >40.0 .AND. Din<=42.0)Write (12,*) 'Din inch= 42 INCH', Din
IF (Din >42.0 .AND. Din<=44.0)Write (12,*) 'Din inch= 44 INCH', Din
IF (Din >44.0 .AND. Din<=48.0)Write (12,*) 'Din inch= 48 INCH', Din

IF (Din >=9.5 .AND. Din<=10.0)Write (*,*) 'Din inch= 10 INCH', Din
IF (Din >10.0 .AND. Din<=12.0)Write (*,*) 'Din inch= 12 INCH', Din
IF (Din >12.0 .AND. Din<=14.0)Write (*,*) 'Din inch= 14 INCH', Din
IF (Din >14.0 .AND. Din<=16.0)Write (*,*) 'Din inch= 16 INCH', Din
IF (Din >16.0 .AND. Din<=18.0)Write (*,*) 'Din inch= 18 INCH', Din
IF (Din >18.0 .AND. Din<=20.0)Write (*,*) 'Din inch= 20 INCH', Din
IF (Din >20.0 .AND. Din<=24.0)Write (*,*) 'Din inch= 24 INCH', Din
IF (Din >24.0 .AND. Din<=30.0)Write (*,*) 'Din inch= 30 INCH', Din
IF (Din >30.0 .AND. Din<=36.0)Write (*,*) 'Din inch= 36 INCH', Din
IF (Din >36.0 .AND. Din<=40.0)Write (*,*) 'Din inch= 40 INCH', Din
IF (Din >40.0 .AND. Din<=42.0)Write (*,*) 'Din inch= 42 INCH', Din
IF (Din >42.0 .AND. Din<=44.0)Write (*,*) 'Din inch= 44 INCH', Din
IF (Din >44.0 .AND. Din<=48.0)Write (*,*) 'Din inch= 48 INCH', Din
        write (*,*) Din, '     Din'
        write (*,*) 'Q=45400000 m3/day, discharge pressure fixed at 3000kN/m2'
        Write (12,*) 'Q=45400000 m3/day, discharge pressure fixed at 
3000kN/m2'
        write (*,*)''
        write (12,*) '        DT','        Pipe Inlet Pressure (kN/m2)'

        write (*,*) '         DT', '          Pipe Inlet Pressure (kN/m2)'
DO QT=5,85,5
        QT1=QT*100000
        write (*,*) QT1,'     throughput'
        write (12,*) QT1, '    throughput'
DO DT=100,1200,50
   Din=DT*0.03937
```

```
    Y1=(DT**2.667)
    Y2=DENOM11*Y1
    Y=(Q/Y2)**2
    Y3=e*(P2p**2)
    Y4=Y*DENOM1
    P2D=(Y4+Y3)**0.5
    write (*,*) DT,  P2D
   write (12,*) DT,  P2D
end do
          write (*,*)''
          write (12,*) ''
end do
          write (*,*)''
          write (12,*)''
          write (*,*) 'DT1= 304.8 mm or 12"'
          write (*,*) '       QT1','        Pipe Inlet Pressure (kN/m2)'
          write (12,*) '       QT1','        Pipe Inlet Pressure (kN/m2)'
DO        QT=5,85,5
          QT1=QT*100000
    DT1=304.8
    Y1=(DT1**2.667)
        Y2=DENOM11*Y1
    Y=(QT1/Y2)**2
    Y3=e*(P2p**2)
    Y4=Y*DENOM1
    P2D=(Y4+Y3)**0.5
    write (*,*) QT1, P2D
    write (12,*) QT1, P2D
END DO
          write (*,*)''
          write (12,*)''
          write (*,*) 'DT1= 609.6 mm or 24"'
          write (*,*) '       QT1','         Pipe Inlet Pressure (kN/m2)'

          write (12,*) 'DT1= 609.6 mm or 24"'
          write (12,*) '       QT1','         Pipe Inlet Pressure (kN/m2)'
DO        QT=5,85,5
          QT1=QT*100000
         DT1=609.6
        Y1=(DT1**2.667)
       Y2=DENOM11*Y1
      Y=(QT1/Y2)**2
    Y3=e*(P2p**2)
    Y4=Y*DENOM1
    P2D=(Y4+Y3)**0.5
    !write (*,*) QT1, P2D
    write (12,*) QT1, P2D
    end do
```

```
        write (*,*)''
        write (12,*)''
        write (*,*) 'DT1= 914.4 mm or 36"'
        write (*,*) '      QT1',' 	Pipe Inlet Pressure (kN/m2)'
        write (12,*) '      QT1',' 	Pipe Inlet Pressure (kN/m2)'
DO      QT=5,85,5
        QT1=QT*100000
   DT1=914.4
   Y1=(DT1**2.667)

   Y2=DENOM11*Y1
   Y=(QT1/Y2)**2
   Y3=e*(P2p**2)
   Y4=Y*DENOM1
   P2D=(Y4+Y3)**0.5
   !write (*,*) QT1, P2D
   write (12,*) QT1, P2D
END DO
        write (*,*)''
        write (12,*)''
        write (*,*) 'DT1= 1 219.2  mm or 48"'
        write (*,*) '      QT1',' 	Pipe Inlet Pressure (kN/m2)'
        write (12,*) '      QT1',' 	Pipe Inlet Pressure (kN/m2)'
DO      QT=5,85,5
        QT1=QT*100000
   DT1=1219.2
   Y1=(DT1**2.667)

   Y2=DENOM11*Y1
   Y=(QT1/Y2)**2
   Y3=e*(P2p**2)
   Y4=Y*DENOM1
   P2D=(Y4+Y3)**0.5
   !write (*,*) QT1, P2D
   write (12,*) QT1, P2D
END DO
        write (*,*)''
        write (12,*)''
        write (*,*) 'DT1= 1 422.4  mm or 56"'
        write (*,*) '      QT1',' 	Pipe Inlet Pressure (kN/m2)'
        write (12,*) 'DT1= 1 422.4  mm or 56"'
        write (12,*) '      QT1',' 	Pipe Inlet Pressure (kN/m2)'
DO      QT=5,85,5
        QT1=QT*100000
   DT1=1422.4
   Y1=(DT1**2.667)

   Y2=DENOM11*Y1
```

```
   Y=(QT1/Y2)**2
   Y3=e*(P2p**2)
   Y4=Y*DENOM1
   P2D=(Y4+Y3)**0.5
   !write (*,*) QT1, P2D
   write (12,*) QT1, P2D
END DO

!***************************************************************
!CALCULATING TOTAL PIPE WEIGHT AND COST
!***************************************************************
Pipe_weight=0.0246*t*(D-t)! kg/m
!write (*,*) Pipe_weight,'       Pipe_weight'
Total_Pipeweight=Pipe_weight*L
Pipe_cost=PIPEUNIT_COST*Total_Pipeweight
write (*,*)Pipe_cost,'      PipeMaterial_cost'
Area=(Pi*D**2)/4
!_____________________________________________________________________________________________
! PIPE EXTERNALLY COATED, WRAPPED AND DELIVERED TO FIELD AT
EXTRA COST OF $15 PER METTER
!****************************************************************************************

Total_Pipecost= Pipe_cost+(CWD_Cost*L)
!Unit Installation cost at $164 per meter
Pipeline_Installation_Cost=UNIT_INSTALATIONCOST*L
!Extra_cost owing to crossing of roads, rivers and highways
Extra_cost=E_Cost_Factor*Pipeline_Installation_Cost
Pipeline_Capitalcost=Total_Pipecost+Pipeline_Installation_Cost+Extra_cost
!write (*,*) Pipeline_Capitalcost,'       Pipeline_Capitalcost'
!____________________________
!Pipeline Maintenance Cost
!************************
Annual_Mtce_Cost= 0.05*Total_Pipecost
P_Total_Cost=Annual_Mtce_Cost+Pipeline_Capitalcost
!_______________________________________
!COMPRESSOR STATION TECHNO-ECONOMICS
!**********************************
         w1=Q*1.89/84600
         w2=Q*1.84/84600
         COMP1_WORK=(8.314/M_WEIGHT)*(Z*T1)*(n/(n-1))*(((P2/P1)**((n-
1)/n))-1)
         COMP1_POWER=W1*COMP1_WORK/EFF1
         COMP2_WORK=(8.314/M_WEIGHT)*(Z*T1)*(n/(n-1))*(((p4/p3)**((n-
1)/n))-1)
         COMP2_POWER=W2*(COMP2_WORK)/EFF2

         Total_Power= COMP1_POWER+COMP2_POWER
```

```
        DRIVER_POWER=Total_Power*1.15
write (*,*) DRIVER_POWER, '    DRIVER_POWER'
GT_CAPCOST=DRIVER_POWER*GTCOST_PKW
Total_compcost=Comp1_cost+comp2_cost
TOTAL_COMPST_COST=CSCF*Total_Power
WRITE (*,*) TOTAL_COMPST_COST, '    TOTAL_COMPST_COST'
!________________________________________
!MAIN VALVE STATIONS AT EVERY 32 km
!*********************************
        MVS=L/32
        MVS_Cost=nint(MVS)*COST_PER_MVS
        Write (*,*) MVS, '     MVS'
        write (*,*) nint(MVS), '     nint(MVS)'
        Write (*,*) MVS_Cost, '      MVS_Cost'
!__________________________________
!METER STATIONS AND REGULATORS
!****************************
        MSR=L/25

        MSR_Cost=nint(MSR)*Cost_Per_MSR
        write (*,*) MSR, '     MSR'
        write (*,*) nint(MSR), '     nint(MSR)'
        Write (*,*) MSR_Cost, '      MSR_Cost'
!_____________________________________________
!NATURAL GAS PIPELINE SYSTEM CAPITAL COST
!**************************************
        CAPCOST=TOTAL_COMPST_COST
________________________________________
!SCADA AND TELECOMMUNICATION SYSTEM
!***********************************
        SCADA_COST=0.03*capcost
!___________________
!ROW-RIGHT OF WAY
!****************
        CompStatncost=Total_compcost+(Total_compcost*COMPSTATION_FA
        CTOR)
        ROW=0.06*Capcost
                PIPELINE_SYS_CAPCOST=Pipeline_Capitalcost+MSR_Cost+R
        OW+SCADA_COST+MVS_Cost

        TOTAL_CAPCOST=PIPELINE_SYS_CAPCOST+CAPCOST
        WRITE (*,*) PIPELINE_SYS_CAPCOST,'
PIPELINE_SYS_CAPCOST'
        write (*,*) TOTAL_CAPCOST,'       TOTAL_CAPCOST'
IF (GT_MOT_SWITCH == 1)THEN
!____________________________
!ECONOMICS OF GAS TURBINE
!************************
```

```
        GT_CAPCOST= DRIVER_POWER*GTCOST_PKW
        write (*,*) GT_CAPCOST, '    GT_CAPCOST'
        !write (*,*) TOTAL_GT_CAPCOST,'      TOTAL_GT_CAPCOST'
        TOTALCAP_COST_GT=PIPELINE_SYS_CAPCOST+TOTAL_COMPST
_COST+GT_CAPCOST
        WRITE (*,*) TOTALCAP_COST_GT, '      TOTALCAP_COST_GT'
!______________________________
!GT OPERATING COST (FUEL COST)
!*****************************
        !READING FROM THE PERFORMANCE OUTPUT
        Read (13,'(8/)')
        READ (13,*) dummy, dummy, FF !FF IS FUEL FLOW kg/s
        FF=Mass_Fuel
        write (*,*) Mass_Fuel, '    Mass of fuel'
        F_Vol=(Mass_Fuel/F_Density)
        write (*,*) F_Vol, '     F_Vol'
        F_Energy=F_Vol*CV_Fuel*VC_Factor
        write (*,*) F_Energy, '     F_Energy'
        write (12,*) F_Energy, '     F_Energy'
        Energy_PA=F_Energy*O_Time*3600
        write (*,*) Energy_PA, '     Energy_PA'
        write (12,*) Energy_PA, '     Energy_PA'
!_________
!FUEL COST
!*********
        GT_OPCOST_PA=(Energy_PA/1000000)*Gas_Price
        write (*,*) GT_OPCOST_PA,'  GT_OPCOST_PA'
        write (12,*) GT_OPCOST_PA,'  GT_OPCO_PA'
!____________________________
!GAS TURBINE MAINTENANCE COST
!***************************
        GTMtce_PA=(Mtce_Factor*DRIVER_POWER)/0.746
        write (*,*) GTMtce_PA, '    GTMtce_PA'
        write (12,*) GTMtce_PA, '    GTMtce_PA'
        GT_Total_Cost=GT_CAPCOST+GTEnergyCost_PA
        write (12,*) GT_Total_Cost, '    GT_Total_Cost'
        write (*,*) GT_Total_Cost, '    GT_Total_Cost'
        Annual_RningcostGT=GTMtce_PA+GTEnergyCost_PA
        total_Projectcost_GT=P_Total_Cost+TOTAL_COMPSTATION_COST+G
T_Total_Cost
!**********************************************
!REVENUES FROM SALES OF NATURAL GAS
!**********************************************
        Q_E=Q/(24*3600)
        write (*,*) Q_E, '  Q_E'
  write (12,*) Q_E, '  Q_E'
        GAS_E_SOLD=(Q_E*CV_Fuel*VC_Factor*3600*O_Time)
        REV_PA=(GAS_E_SOLD*Gas_Price)
```

```
        WRITE (12,*) REV_PA,  '    REV_PA $)'
        WRITE (12,*) TOTAL_REV,  '   TOTAL_REV_PA $)'
!CASH FLOW CALCULATIONS
!*********************
!CALCULATING CASH INFLOW
        LOAN=0.8*TOTALCAP_COST_GT
        EQUITY=0.2*TOTALCAP_COST_GT
   !CALCULATION OF NPV
        ALLOCATE(Table_a(0:33,1:6))
!ALLOCATE(Table_b(0:10,1:6))
        WRITE (*,*) ''
        WRITE (*,*) '      YEAR', '       Disct_F', '    GAS_REV_PA', '
GT_PV_YR',' GT_NET_CFL', '     CASH_INF'
        WRITE (12,*) '       YEAR', '          Disct_F', '      GAS_REV_PA', '
GT_PV_YR','      GT_NET_CFL', '      CASH_INF'
        WRITE (*,*) ''
        WRITE (*,*) ''
        CO2_tax=16
        GT_Total=0
        Rev_Total=0
        GT_NPV=0
        CASH_INF_T=0
        GT_CASH_OF_T=0
        AA_T=0
        AB_T=0
        AC_T=0
        LOAN_P=LOAN+(0.06*LOAN)
DO YR=1,33,1

        YEAR=YR+2012
        Disct_F=1/((1+D_RATE)**YR)
        GAS_E_SOLD=(Q_E*CV_Fuel*VC_Factor)
        GAS_ENERGY_PA=GAS_E_SOLD*3600*O_Time
        GAS_REV_PA=((GAS_ENERGY_PA/1000000)*Gas_Price)
        write (*,*) GAS_REV_PA, '  GAS_REV_PA'
IF (YR==1)THEN
   CASH_INF=LOAN+EQUITY
   ELSE IF (YR .GE. 2 .AND. YR .LE.3)THEN
     CASH_INF=0
        ELSE IF (YR .GE.4 .AND. YR.LE.33) THEN
       CASH_INF=GAS_REV_PA*DEG_A(YR-3)
END IF
IF (YR==1)THEN
   GT_CASH_OF=0.3*TOTAL_CAPCOST
   AA=0
   AB=0
   AC=0
   ELSE IF (YR==2)THEN
```

```
    GT_CASH_OF=0.3*TOTAL_CAPCOST
    AA=0
    AB=0
    AC=0
   ELSE IF (YR==3)THEN
    AA=0
    AB=0
    AC=0
            GT_CASH_OF=0.4*TOTAL_CAPCOST
            LL= (O_Time*3600*44)/16
          kg_CO2 = Mass_Fuel * LL
            write (*,*) kg_CO2,'   kg_CO2'
        CO2_cost_YR = (kg_CO2 * CO2_tax)/1000
        write (*,*) CO2_cost_YR,'   CO2_cost_YR'
    ELSE IF (YR.GE.4 .AND. YR.LE.33)THEN

        AA=GT_OPCOST_PA*DEG_A(YR-3)
         !write (*,*) AA,'       AA'
          AB=GTMtce_PA*DEG_A(YR-3)
          AC=CO2_cost_YR*DEG_A(YR-3)
                     WRITE (*,*) AA, AB, AC
          GT_CASH_OF=AA+AB+AC+(0.05*LOAN_P)

END IF

        GT_NET_CFL=CASH_INF-GT_CASH_OF
        TAX=0.3*GT_NET_CFL
        GT_NET_CFL_ATAX=GT_NET_CFL-TAX
        GT_PV_YR=GT_NET_CFL_ATAX*Disct_F
        GT_NPV=GT_NPV+GT_PV_YR
        AA_T=AA_T+AA
        AB_T=AB_T+AB
        AC_T=AC_T+AC
        CASH_INF_T=CASH_INF_T+CASH_INF
        GT_CASH_OF_T=GT_CASH_OF_T+GT_CASH_OF

        Table_a (YR,1)= Year
        Table_a (YR,2)= Disct_F
        Table_a (YR,3)= GAS_REV_PA
        Table_a (YR,4)= GT_PV_YR
        Table_a (YR,5)= GT_NET_CFL
        Table_a (YR,6)= CASH_INF



        WRITE (*,*) Table_a (YR,1),Table_a (YR,2),Table_a (YR,3),Table_a
        (YR,4),Table_a (YR,5),Table_a (YR,6)
```

```
        WRITE (12,*) Table_a (YR,1),Table_a (YR,2),Table_a (YR,3),Table_a
        (YR,4),Table_a (YR,5),Table_a (YR,6)

END DO
        write (*,*)CASH_INF, '     cash in flow'
        write (*,*) GT_CASH_OF,'       GT_CASH_OF'
        WRITE (*,*) GT_NPV, '    GT_NPV'
        WRITE (12,*) GT_NPV, '    GT_NPV'
        WRITE (*,*) AA_T, '    GT_Op_Cost_30_T'
        WRITE (*,*) AB_T, '    GT_Mtce_cost_30_T'
        WRITE (*,*) AC_T, '    GT_CO2_Tax_30_T'
        WRITE (*,*) GT_CASH_OF_T,'     GT_CASH_OF_T'

!END OF GAS TURBINE CODE
!******************************************************************************
!******************************************************************************
!******************************************************************************

!ELECTRIC DRIVE CODE BEGINS

ELSE IF (GT_MOT_SWITCH == 2)THEN
!_________________________________
!ELECTRIC MOTOR TECHNO-ECONOMICS
!************************
!ELECTRIC MOTORCAPITAL COST
        Cost_Emotor=DRIVER_POWER*Motcost_kw
        WRITE (*,*) Cost_Emotor, '       Cost_Emotor'
!MAINTENANCE COST
        M_MntceCostPA= EM_MTCE_Factor*DRIVER_POWER
        WRITE (*,*) M_MntceCostPA,'     M_MntceCostPA'
   !OPERATING COST
        MTOR_E_PA=(DRIVER_POWER/MTOR_EFF)*O_TIME
        WRITE (*,*) MTOR_E_PA,'      MTOR_E_PA'
        MTOR_OPCOST_PA=MTOR_E_PA*E_cost_kwh
        WRITE (*,*) MTOR_OPCOST_PA,'      MTOR_OPCOST_PA'
        WRITE (*,*) E_cost_kwh,'     E_cost_kwh'
        write (12,*) E_ENERGY, '    E_ENERGY'
        M_OPhourPA=M_Ophour*365

        write (12,*) M_OPhourPA,   '    M_OPhourPA'
        E_ENERGY_PA=E_ENERGY*M_OPhourPA
        M_Op_cost_PA=E_ENERGY_PA/1000*E_cost_kwh

        MTOR_CAPCOST=Cost_Emotor+PIPELINE_SYS_CAPCOST+TOTAL_
        COMPST_COST
        EMotor_TotcostPA=Cost_Emotor+M_MntceCostPA+M_Op_cost_PA
        Total_Projectcost_Emotor=P_Total_Cost+TOTAL_COMPSTATION_CO
        ST+EMotor_TotcostPA
```

```
        TOTALCAP_COST_EM=Pipeline_Capitalcost+Total_compcost+MVS_C
        ost+SCADA_COST+ROW+Cost_Emotor
        Write (12,*) EMotor_TotcostPA, '    EMotor_TotcostPA'
        Write (12,*) M_Op_cost_PA, '   Op_cost_PA'
        MTOR_Total=MTOR_Total+DSCNT_OP_COST_MOTOR
!_______________________________________
!REVENUES FROM SALES OF NATURAL GAS
!*********************************
        Q_E=Q/24
        GAS_E_SOLD=(Q_E*CV_Fuel*VC_Factor*O_Time)
        REV_PA=(GAS_E_SOLD/1000000*Gas_Price)
        TOTAL_REV=REV_PA*30

        WRITE (12,*) REV_PA,  '    REV_PA $)'
        WRITE (12,*) TOTAL_REV,  '   TOTAL_REV_PA $)'
  !CASH FLOW CALCULATIONS
!*********************


!___________________________
!CALCULATING CASH INFLOW
!************************
        LOAN=0.8*MTOR_CAPCOST
        EQUITY=0.2*MTOR_CAPCOST
!CALCULATION OF NPV
        ALLOCATE(Table_M(0:30,1:6))
        WRITE (*,*) ''
        WRITE (*,*) '      YEAR', '    Disct_F', '    GAS_E_SOLD', '    REV_PA','
DSCNT_OP_COST_MOTOR', ' DSCNT_OP_COST_GT'
        WRITE (12,*) '    YEAR', '    Disct_F', '    GAS_E_SOLD', '    REV_PA','
DSCNT_OP_COST_MOTOR', ' DSCNT_OP_COST_GT'
        WRITE (*,*) ''
        WRITE (*,*) ''
        MTOR_Total=0
        Rev_Total=0
        MTOR_NPV=0
        CASH_INF_T=0
        MTOR_CASH_OF_T=0
        AAM_T=0
        ABM_T=0
        LOAN_P=LOAN+(0.06*LOAN)
DO YR=1,23,1
        YEAR=YR+2012
        Disct_F=1/((1+D_RATE)**YR)
        GAS_E=Q_E*CV_Fuel*VC_Factor
        GAS_E_SOLD=(GAS_E*O_time)
        GAS_REV_PA=(GAS_E_SOLD/1000000*Gas_Price)
  IF (YR==1)THEN
    CASH_INF=LOAN+EQUITY
```

```
   AAM=0
   ABM=0
   ELSE IF (YR .GE. 2 .AND. YR .LE.3)THEN
     CASH_INF=0
   AAM=0
   ABM=0

  ELSE IF (YR .GE.4 .AND. YR.LE.23) THEN
    AAM=0
    ABM=0
       CASH_INF=GAS_REV_PA
END IF
  IF (YR==1)THEN
   MTOR_CASH_OF=0.3*MTOR_CAPCOST
       ELSE IF (YR==2)THEN
     MTOR_CASH_OF=0.3*MTOR_CAPCOST
         ELSE IF (YR==3)THEN
   MTOR_CASH_OF=0.4*MTOR_CAPCOST
      ELSE IF (YR.GE.4 .AND. YR.LE.23)THEN
    AAM=MTOR_OPCOST_PA*DEG_A(YR-3)
   !write (*,*) AA,'       AA'
   ABM=M_MntceCostPA*DEG_A(YR-3)

MTOR_CASH_OF=MTOR_OPCOST_PA+M_MntceCostPA+0.05*LOAN_P

END IF
          MTOR_NET_CFL=CASH_INF-MTOR_CASH_OF
          TAX=0.3*MTOR_NET_CFL
          MTOR_NET_CFL_ATAX=MTOR_NET_CFL-TAX
          MTOR_PV_YR=MTOR_NET_CFL_ATAX*Disct_F
          MTOR_NPV=MTOR_NPV+MTOR_PV_YR
          CASH_INF_T=CASH_INF_T+CASH_INF
          MTOR_CASH_OF_T=MTOR_CASH_OF_T+MTOR_CASH_OF
          AAM_T=AAM_T+AAM
          ABM_T=ABM_T+ABM

          Table_a (YR,1)= Year
          Table_a (YR,2)= Disct_F
          Table_a (YR,3)= GAS_E_SOLD
          Table_a (YR,4)= REV_PA
          Table_a (YR,5)= MTOR_NET_CFL
          Table_a (YR,6)= MTOR_PV_YR
          WRITE (*,*) Table_a (YR,1),Table_a (YR,2),Table_a (YR,3),Table_a
          (YR,4),Table_a (YR,5),Table_a (YR,6)
          WRITE (12,*) Table_a (YR,1),Table_a (YR,2),Table_a (YR,3),Table_a
          (YR,4),Table_a (YR,5),Table_a (YR,6)
  END DO !(DO NO 1)
          WRITE (*,*) AAM_T,'        Motor Total Operating cost'
```

```
        WRITE (*,*) ABM_T,'       Motor Total Maintenance cost'
        WRITE (*,*) Cost_Emotor,'         Cost_Emotor'
        WRITE (*,*) MTOR_CASH_OF_T,'         MTOR_CASH_OF_T'
        WRITE (*,*) MTOR_NPV, '        MTOR_NPV'
        WRITE (12,*) MTOR_NPV, '        MTOR_NPV'
END IF

        DEALLOCATE(Table_a)
        WRITE (12,*) '' !table gaps
!**************************************************************************************
********THE big LOOP ENDS
End Do
        write (*,*) Rev_Total, '     Rev_Total'
        write (*,*) GT_Total, '     GT_Total'
        write (*,*) MTOR_Total, '    MTOR_Total'
  !______________________________________________________________
!WRITING RESULTS TO MAIN OUTPUT FILES
!*****************************************************
        Write (12,*) Total_Projectcost_GT, '     Total_Projectcost_GT'
        Write (12,*) Total_Projectcost_Emotor, '    Total_Projectcost_Emotor'
        Write (12,*) M_Total_opcost, '    M_Total_opcost'
        Write (*,*) Total_Projectcost_GT, '     Total_Projectcost_GT'
        Write (*,*) Total_Projectcost_Emotor, '    Total_Projectcost_Emotor'
        Write (*,*) M_Total_opcost, '    M_Total_opcost'
        Write (12,*) Area, 'Area m^2'
        Write (12,*) Area, 'Area m^2'
        Write (12,*) Volume, 'Volume m^3'
        Write (12,*) Pipe_weight, '   Pipe_weight kg/m)'
        Write (12,*) COMP1_WORK, '   COMP1_WORK (J)'
        Write (12,*) COMP1_POWER, '   COMP1_POWER kW'
        Write (12,*) COMP2_WORK, '   COMP2_WORK (J'
        Write (12,*) COMP2_POWER, '   COMP2_POWER kW'
        Write (12,*) Total_Power, '   Total_Power kW'
        Write (12,*) DRIVER_POWER, '   DRIVER_POWER kW'
        Write (12,*) Pipe_cost, '   Pipe_cost $'
        Write (12,*) Total_Pipecost, '   Total_Pipecost $'
        Write (12,*) Pipeline_Installation_Cost, '   Pipeline_Installation_Cost $'
        Write (12,*) Pipeline_Capitalcost, '   Pipeline_Capitalcost $'
        Write (12,*) Total_compcost, '   Total_compcost $'
        Write (12,*) ROW, '   ROW $'
        Write (12,*) Total_Projectcost, '   Total_Projectcost $'
        Write (12,*) D, '   D mm ='
        Write (12,*) T_B, '   T_B mm ='

        Write (12,*) DENOM11, '   DENOM11 mm'
        Write (12,*) T_B_E, '   T_B_E mm '
        Write (12,*) TOTAL_COMPSTATION_COST, '
        TOTAL_COMPSTATION_COST $'
```

```
WRITE (12,*) GT_CAPCOST, '   GT_CAPCOST'
Write (12,*) Energy_Total, '  Energy_Total'
Write (12,*) GTEnergyCost_PA, '      GTEnergyCost_PA'
Write (12,*) GTTotal_E_Cost, '  GTTotal_E_Cost'


!************************************************************************************************
****
!END OF CODE AND CHECKINGS
End Program Pipeline
```

C.2 MATLAB codes for Genetic Algorithm Optimization

```
function optimum_GTcost = gtoptimum(DRIVE_POWER)

n=1.4;
EFF2=0.8;
w1=189;
Z=0.9;
Q=4540000;
G=0.6;
Gas_Price=5;
E=0.95;
Pb=101;
Tb=288.15;
Tf=332.007;
T1=278.15;
D=609.6;
O_Time=8760;
L=512;
CV_Fuel=43000;
Mtce_Factor=30;
GTCOST_PKW=350;
CSCF=1340;
VC_Factor=1.022640;
F_Density=0.9;
M_WEIGHT=19.5;
CO2_tax=16;
P2p=3000;
P1=1870;
n1=n/(n-1);
 n2=(n-1)/n;
 %savefile='DRIVE_POWER.mat';
%DRIVE_POWER=34;
 %for DRIVE_POWER=20:1:40
   % save(savefile, 'DRIVE_POWER')
   NE12=EFF2*DRIVE_POWER*1000;
   NE=NE12/w1;
   NE011=8.314*Z*T1;
  NE1=M_WEIGHT/NE011; %!  8.314*Z*T1
  Ne01=NE*n2;
  NE02=Ne01*NE1;
   NE03=(1+NE02);
   NE04=NE03^n1;
   P_d=P1*NE04;
    PDiff=P_d^2-P2p^2;
```

```
    Conts=G*Tf*Z;
    NE11=PDiff/Conts;
    NE2=3.7435*0.001*E*Tb;
    NE3=D^2.667;
    NE4=NE3*NE2;
    NE5=Q*Pb;
    NE6=NE5/NE4;
    NE7=NE6^2;
    NE8=1/NE7;
    L_pipe=NE11*NE8;
 %savefile='DRIVE_POWERNCSGT_OPCOST_PA.mat';
    NCS=L/L_pipe;
    Mass_Fuel=DRIVE_POWER*0.057; %%! DRIVE_POWER is in MW here
    F_Vol=Mass_Fuel/F_Density;

F_Energy=F_Vol*CV_Fuel*VC_Factor;

Energy_PA=F_Energy*O_Time*3600;
kg_CO2 = Mass_Fuel * O_Time*3600*(44/16);

CO2_cost_YR = kg_CO2 * CO2_tax/1000;

GTMtce_PA=(Mtce_Factor*DRIVE_POWER*1000)/0.746
GT_CAPCOST= DRIVE_POWER*GTCOST_PKW*1000
TOTAL_COMPST_COST=CSCF*DRIVE_POWER*1000;

%!FUEL COST

GT_OPCOST_PA=(Energy_PA/1000000)*Gas_Price
EX=GT_OPCOST_PA;
EX1=GTMtce_PA;
 GT_OPTM_COST=EX+EX1+(GT_CAPCOST)+(CO2_cost_YR);   % obj function
    %save(savefile, 'GT_OPCOST_PA', 'NCS', 'DRIVE_POWER');

%end

optimum_GTcost = GT_OPTM_COST;
```

```
close all
clear all
clc
%GT_COST = @(var) fitnessfun(var);
optimum_GTcost = @(DRIVE_POWER)gtoptimum(DRIVE_POWER);
lb = 20;ub = 34;
opts =
gaoptimset('populationSize',50,'Generations',50,'PlotFcns',{@gaplotbes
tf,@gaplotbestindiv,@gaplotscores,@gaplotrange,@gaplotselection,@gaplo
tdistance},'Display','iter'); % an optimization structure setting
initpop as the initial population
[varga fga flga oga population scores] =
ga(optimum_GTcost,1,[],[],[],[],lb,ub,[],opts) % calls ga
@gaplotscorediversity,@gaplotgenealogy,@gaplotstopping
opts

clear all
clc
%close all
```

```
n=1.4;
EFF2=0.8;
w1=189;
Z=0.9;
Q=4540000;
G=0.6;
Gas_Price=5;
E=0.95;
Pb=101;
Tb=288.15;
Tf=332.007;
T1=278.15;
D=609.6; %609.6;
O_Time=8760;
L=512;
CV_Fuel=43000;
Mtce_Factor=30;
GTCOST_PKW=350;
CSCF=1340;
VC_Factor=1.022640;
F_Density=0.9;
M_WEIGHT=19.6;
P2p=3000;
P1=1870;
n1=n/(n-1);
 n2=(n-1)/n;
 CO2_tax=16;
 savefile='DRIVE_POWER.mat';
%DRIVE_POWER=20
power = []; NCS_A = []; GT_cost = []; CSPosition = []; Discharge_P =
[];
Emission_CO2 = []; Mass_of_Fuel = []; slip = []; Torque = [];
Pressure_diff = [];
Total_GT_Power = []; GT_COST_TOTAL = []; Emission_kg = [];
Mass_of_Fuel =[];
%CSP=0;
%Comp_Work=(Z/1000)(8314/M_Weight)*T1*(n/n-1)*(((P_d/P_s)^(n-1/n))-1
%Drive_Power=mass flow*CW/comp eff
Total_Power = 0;
GT_OPTM_COST_T=0;
% for DRIVE_POWER=24:1:34;
     DRIVE_POWER=34;
    save(savefile, 'DRIVE_POWER')
   NE12=EFF2*DRIVE_POWER*1000;
   NE=NE12/w1;
   NE011=8.314*Z*T1;
  NE1=M_WEIGHT/NE011; %!  8.314*Z*T1
  Ne01=NE*n2;
  NE02=Ne01*NE1;
   NE03=(1+NE02);
   NE04=NE03^n1;
   P_d=P1*NE04
    PDiff=P_d^2-P2p^2;
    Conts=G*Tf*Z;
    NE11=PDiff/Conts;
    NE2=3.7435*0.001*E*Tb;
    NE3=D^2.667;
    NE4=NE3*NE2;
    NE5=Q*Pb;
```

```
    NE6=NE5/NE4;
    NE7=NE6^2;
    NE8=1/NE7;
    L_pipe=NE11*NE8
    %CSP=CSP+L_pipe;
 savefile='DRIVE_POWERNCSGT_OPCOST_PA.mat';
    NCS=L/L_pipe;
   % if NCS>1
   Total_Power=Total_Power+(NCS*DRIVE_POWER);

    Mass_Fuel=DRIVE_POWER*0.057 %%! DRIVE_POWER is in MW here

    F_Vol=Mass_Fuel/F_Density;

F_Energy=F_Vol*CV_Fuel*VC_Factor;

Energy_PA=F_Energy*O_Time*3600;
GTMtce_PA=(Mtce_Factor*DRIVE_POWER*1000)/0.746;
GT_CAPCOST= DRIVE_POWER*GTCOST_PKW*1000;
TOTAL_COMPST_COST=CSCF*DRIVE_POWER*1000;


%!FUEL COST
%for Gas_Price=1:1:10
GT_OPCOST_PA=(Energy_PA/1000000)*Gas_Price;
%
***********************************************************************
*****************************************************
%!CALCULATING CO2 EMISSION AND CO2 COST
%!*****************************************************************=***
*********************************************************

kg_CO2 = Mass_Fuel * O_Time*3600*(44/16)

CO2_cost_YR = kg_CO2 * CO2_tax/1000

EX=GT_OPCOST_PA
EX1=GTMtce_PA
Em_cost_T= CO2_cost_YR
GTcapcost= GT_CAPCOST*NCS
 GT_OPTM_COST=EX+EX1+(GT_CAPCOST*NCS)+(CO2_cost_YR)    % obj function
    %save(savefile, 'GT_OPCOST_PA', 'NCS', 'DRIVE_POWER');
    GT_OPTM_COST_T=GT_OPTM_COST_T+GT_OPTM_COST

    TOTAL_COST_PA=GT_CAPCOST+GTMtce_PA+GT_OPCOST_PA+Em_cost_T


 power = [power; DRIVE_POWER];
 NCS_A = [NCS_A; NCS];
 GT_cost = [GT_cost; GT_OPTM_COST];
 CSPosition = [CSPosition; L_pipe];
 Discharge_P = [Discharge_P; P_d];
 Emission_kg= [Emission_kg; kg_CO2];
 Emission_CO2= [Emission_CO2; CO2_cost_YR];
 Mass_of_Fuel= [Mass_of_Fuel; Mass_Fuel];
 Pressure_diff = [Pressure_diff; PDiff];
```

```
 Total_GT_Power = [Total_GT_Power; Total_Power];
 GT_COST_TOTAL = [GT_COST_TOTAL; GT_OPTM_COST_T];
 Mass_of_Fuel = [Mass_of_Fuel; Mass_Fuel];

    %else break
   % end

% end
 %end
power
NCS_A
GT_cost
CSPosition
Discharge_P
Emission_kg
Emission_CO2
Mass_of_Fuel
Pressure_diff
Total_GT_Power
GT_COST_TOTAL


figure (1)
plot(power, CSPosition)
axis ([19 35 100 550])
xlabel ('Drive Power [MW]'); ylabel('Optimum Compressor Station
Position (km)')
figure (2)
plot(power, NCS_A)
axis ([23 35 0.5 3.0])
xlabel ('Drive Power [MW]'); ylabel('Optimum Number of Compressor
Stations')
figure (3)
%clf
subplot(4,1,1)
plot(power, NCS_A)
xlabel ('Drive Power [MW]'); ylabel('Number of Compressor Stations')
subplot (4,1,2)
plot (power, GT_cost)
xlabel ('Drive Power [MW]'); ylabel('GT Cost [$]')
subplot (4,1,3)
plot (NCS_A, GT_cost)
xlabel ('Number of Compressor Station'); ylabel('GT Cost [$]')
subplot(4,1,4)
plot(power, CSPosition)
xlabel ('Drive Power [MW]'); ylabel('Compressor Station Positions')
%figure (4)
%plot (power, GT_cost)
%xlabel ('Drive Power [MW]'); ylabel('GT Cost [$]')
figure (4)
plot (power, Discharge_P)
axis ([23 35 4000 6000])
xlabel ('Drive Power [MW]'); ylabel('Discharge Pressure (KN/m2)')
figure (5)
plot (power, GT_cost)
axis ([23 35 1.7e7 3e7]);
xlabel ('Drive Power [MW]'); ylabel('Total Cost [$]')
figure (6)
```

```
plot (Emission_CO2, Mass_of_Fuel)
xlabel ('Emission Cost ($)'); ylabel('Mass of Fuel [kg/s]')
figure (7)
plot (power, Mass_of_Fuel)
axis ([23 35 1.3 2.0])
xlabel ('Drive Output [MW]'); ylabel('Mass of Fuel [kg/s]')
Rr=3.16;
Er=240;
Xr=2.14;
for s=-1:0.1:1 % DRIVE_POWER=24:1:42
    Torq=(Rr*Er)/(Rr^2+(s^2*Xr^2));
     slip = [slip; s];
     Torque = [Torque; Torq];
end
```

C.3 PERFORMANCE DATA AND TURBOMATCH RESULTS

Some TMR outputs are shown below. This shows some of the desing point results of the engine simulations and few off-design point rsults just to illustrate because the entire off-design point result is too large to print.

```
 1

  TURBOMATCH SCHEME - Windows NT version (October 1999)

 LIMITS:100 Codewords, 800 Brick Data Items, 50 Station Vector
 15 BD Items printable by any call of:-
 OUTPUT, OUTPBD, OUTPSV, PLOTIT, PLOTBD or PLOTSV

  Input "Program" follows

 ! INDUSTRIAL GAS TURBINE SIMULATION


 Programmed by ABDULKARIM NASIR,

 OD SI KE CT FP
  -1
  -1
 INTAKE  S1-2      D1-4             R100
 COMPRE  S2-3      D5-10            R101      V5   V6
 PREMAS  S3,12,4   D11-14
 PREMAS  S4,13,5   D15-18
 BURNER  S5-6      D19-21           R102
 MIXEES  S6,13,7
 TURBIN  S7-8      D22-29,101                 V23
 MIXEES  S8,12,9
 TURBIN  S9-10     D30-38                     V30 V31
 NOZCON  S10-11,1  D39              R107
 PERFOR  S1,0,0    D30,40-42,107,100,102,0,0,0,0,0,0
 CODEND
```

```
DATA ITEMS
1 0.0           ! INTAKE ALTITUDE
2 0.0           ! ISA DEVIATION
3 0.0           ! MACH NO
4 0.9951        ! PRESSURE RECOVERY
!COMPRESSOR
5 -1.0          ! Z PARAMETER
6 -1.0          ! ROTATIONAL SPEED N
7 25.0         ! PRESSURE RATIO
8 0.8825         ! ISENTROPIC EFFICIENCY
9 0.0           ! ERROR SELECTION
10 4.0          ! MAP NUMBER
!PREMAS
11 0.025        ! BLEED AIR
12 0.00         ! FLOW LOSS
13 1.0          ! PRESSURE RECOVERY
14 0.0          ! PRESSURE DROP
!PREMAS
15 0.075        ! BLEED AIR
16 0.0          ! FLOW LOSS
17 1.0          ! PRESSURE RECOVERY
18 0.0          ! PRESSURE DROP
!BURNER
19 0.075        ! FRACTIONAL PRESSURE LOSS DP/P
20 1.0          ! COMBUSTION EFFICIENCY
21 -1.0         ! FUEL FLOW
!HP TURBINE
22 0.0          ! AUXILIARY WORK
23 -1.0         ! NDMF
24 -1.0         ! NDSPEED CN
25 0.895        ! ISENTROPIC EFFICIENCY
26 -1.0         ! PCN
27 1.0          ! COMPRESSOR NUMBER
28 4.0          ! TURBINE MAP NUMBER
29 -1.0         ! POWER LOW INDEX
!POWER TURBINE
30 33679000.0   ! AUXILIARY WORK
31 -1.0         ! NDMF
32 -1.0         ! NDSPEED CN
33 0.885        ! ISENTROPIC EFFICIENCY
34 -1.0         ! PCN
35 0.0          ! COMPRESSOR NUMBER
36 4.0          ! MAP NUMBER
37 1.5          ! POWER LOW INDEX
38 -1.          ! COMWORK
!NOZCON
39 -1.          ! THROAT AREA
!PERFOR
40 1.00         ! PROPELLER EFFICIENCY
41 0.0          ! SCALING INDEX
42 0.0          ! REQUIRED THRUST
-1
1 2 90.0        ! INLET MASS FLOW
6 6 1575.0      ! COMBUSTION OUTLET TEMPERATURE
-1
                                                  Time Now 18:30:42

***********************************************
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```
                 The Units for this Run are as follows:-

 Temperature = K   Pressure = Atmospheres   Length = metres

 Area = sq metres  Mass Flow = kg/sec       Velocity = metres/sec

 Force = Newtons   s.f.c.(Thrust) =  mg/N sec   s.f.c.(Power) = mug/J

 Sp. Thrust =    N/kg/sec     Power =   Watts
1
      ***** DESIGN POINT ENGINE CALCULATIONS *****



 ***** AMBIENT AND INLET PARAMETERS *****
 Alt. =      0.0      I.S.A. Dev. =    0.000       Mach No. =  0.00
 Etar = 0.9951        Momentum Drag =      0.00

 ***** COMPRESSOR  1 PARAMETERS *****
 PRSF =  0.23529E+02     ETASF =  0.10633E+01     WASF =  0.50798E+00
 Z = 0.85000             PR =  25.000             ETA = 0.88250
 PCN =   1.0000          CN = 1.00000             COMWK =  0.44246E+08

 ***** COMBUSTION CHAMBER PARAMETERS *****
 ETASF =  0.10000E+01
 ETA = 1.00000           DLP = 1.8658             WFB =   1.9721

 ***** TURBINE  1 PARAMETERS *****
 CNSF =  0.80319E+02     ETASF =  0.10508E+01     TFSF =  0.27256E+01
 DHSF =  0.18969E+05
 TF =  414.346            ETA = 0.89500            CN =  2.060
 AUXWK =  0.00000E+00

 ***** TURBINE  2 PARAMETERS *****
 CNSF = -0.68748E-02     ETASF =  0.10391E+01     TFSF =  0.69818E+00
 DHSF =  0.19225E+05
 TF =  414.346            ETA = 0.88500            CN =  2.060
 AUXWK =  0.33679E+08

 Additional Free Turbine Parameters:-
 Speed = *****%          Power =  0.33679E+08

 ***** CONVERGENT NOZZLE  1 PARAMETERS *****
 NCOSF =  0.10000E+01
 Area =  1.0241          Exit Velocity =  199.36  Gross Thrust =
17833.24
 Nozzle Coeff. =  0.97259E+00



 Scale Factor on above Mass Flows, Areas, Thrusts & Powers =
1.0000
```

```
 Station   F.A.R. Mass Flow   Pstatic    Ptotal Tstatic   Ttotal     Vel
Area
     1   0.00000    90.000   1.00000   1.00000  288.15   288.15     0.0
******
     2   0.00000    90.000    ******   0.99510  ******   288.15  ******
******
     3   0.00000    90.000    ******  24.87752  ******   761.64  ******
******
     4   0.00000    87.750    ******  24.87752  ******   761.64  ******
******
     5   0.00000    81.169    ******  24.87752  ******   761.64  ******
******
     6   0.02430    83.141    ******  23.01170  ****** 1575.00  ******
******
     7   0.02247    89.722    ******  23.01170  ****** 1520.21  ******
******
     8   0.02247    89.722    ******   5.17190  ****** 1121.99  ******
******
     9   0.02191    91.972    ******   5.17190  ****** 1113.73  ******
******
    10   0.02191    91.972    ******   1.09232  ******   801.23  ******
******
    11   0.02191    91.972   1.00000   1.09232  783.69   801.23   199.4
1.0241
    12   0.00000     2.250    ******  24.87752  ******   761.64  ******
******
    13   0.00000     6.581    ******  24.87752  ******   761.64  ******
******


   Shaft Power =  33679000.00
    Net Thrust = 17833.24
  Equiv. Power =  34828832.00
     Fuel Flow =   1.9721
        S.F.C. =  58.5565
      E.S.F.C. =  56.6233
 Sp. Sh. Power =    374211.09
 Sp. Eq. Power =    386987.03
 Sh. Th. Effy. =   0.3960
          Time Now 18:30:42

 ************************************************
 2    -5.0 !
 -1
 -1
                                                   Time Now 18:30:42

 ************************************************
 BERR( 1) =   -0.50427E-02
 BERR( 2) =    0.13658E-02
 BERR( 3) =   -0.27871E-02
 BERR( 4) =    0.23557E-01
 BERR( 5) =    0.62242E-01

 Loop   1
 BERR( 1) =    0.20135E-05
 BERR( 2) =    0.13447E-05
 BERR( 3) =   -0.18549E-04
 BERR( 4) =    0.74977E-03
```

```
 BERR( 5) =    0.12829E-02
1

      ***** OFF DESIGN ENGINE CALCULATIONS. Converged after   1 Loops
*****



 ***** AMBIENT AND INLET PARAMETERS *****
 Alt. =      0.0      I.S.A. Dev. =   -5.000       Mach No. =  0.00
 Etar = 0.9951        Momentum Drag =      0.00

 ***** COMPRESSOR  1 PARAMETERS *****
 PRSF =  0.23529E+02     ETASF =  0.10633E+01     WASF =  0.50798E+00
 Z = 0.85291             PR =  25.567             ETA = 0.87582
 PCN =   1.0005          CN = 1.00930             COMWK =  0.45324E+08

 ***** COMBUSTION CHAMBER PARAMETERS *****
 ETASF =  0.10000E+01
 ETA = 1.00000           DLP = 1.8999             WFB =   2.0275

 ***** TURBINE  1 PARAMETERS *****
 CNSF =  0.80319E+02     ETASF =  0.10508E+01     TFSF =  0.27256E+01
 DHSF =  0.18969E+05
 TF =  414.434            ETA = 0.89503            CN =  2.061
 AUXWK =  0.00000E+00

 ***** TURBINE  2 PARAMETERS *****
 CNSF = -0.68748E-02     ETASF =  0.10391E+01     TFSF =  0.69818E+00
 DHSF =  0.19225E+05
 TF =  415.136            ETA = 0.88468            CN =  2.061
 AUXWK =  0.34744E+08

 Additional Free Turbine Parameters:-
 Speed = *****%          Power =  0.34744E+08

 ***** CONVERGENT NOZZLE  1 PARAMETERS *****
 NCOSF =  0.10000E+01
 Area =  1.0241          Exit Velocity =  203.01  Gross Thrust =
18584.40
 Nozzle Coeff. =  0.97266E+00



 Scale Factor on above Mass Flows, Areas, Thrusts & Powers =
1.0000



 Station  F.A.R. Mass Flow   Pstatic    Ptotal Tstatic  Ttotal     Vel
Area
     1   0.00000    92.092   1.00000   1.00000  283.15  283.15     0.0
******
     2   0.00000    92.092    ******   0.99510  ******  283.15  ******
******
     3   0.00000    92.092    ******  25.44218  ******  757.52  ******
******
     4   0.00000    89.790    ******  25.44218  ******  757.52  ******
******
```

```
     5   0.00000    83.056    ******  25.44218  ******  757.52  ******
******
     6   0.02441    85.083    ******  23.54229  ****** 1575.00  ******
******
     7   0.02258    91.817    ******  23.54229  ****** 1519.97  ******
******
     8   0.02258    91.817    ******   5.28095  ****** 1121.41  ******
******
     9   0.02202    94.120    ******   5.28095  ****** 1113.08  ******
******
    10   0.02202    94.120    ******   1.09632  ******  797.97  ******
******
    11   0.02202    94.120   1.00000   1.09632  779.80  797.97   203.0
1.0241
    12   0.00000     2.302    ******  25.44218  ******  757.52  ******
******
    13   0.00000     6.734    ******  25.44218  ******  757.52  ******
******


   Shaft Power =  34744476.00
    Net Thrust = 18584.40
  Equiv. Power =  35942740.00
     Fuel Flow =   2.0275
        S.F.C. =  58.3552
      E.S.F.C. =  56.4097
 Sp. Sh. Power =    377279.28
 Sp. Eq. Power =    390290.88
 Sh. Th. Effy. =   0.3974
          Time Now 18:30:42

 ***********************************************
 2    5.0   !
 -1
 -1
                                                  Time Now 18:30:42

 ***********************************************
 BERR( 1) =    0.10670E-01
 BERR( 2) =   -0.89491E-02
 BERR( 3) =   -0.41643E-02
 BERR( 4) =   -0.43465E-01
 BERR( 5) =   -0.12058E+00

 Loop   1
 BERR( 1) =   -0.66096E-04
 BERR( 2) =    0.12702E-03
 BERR( 3) =   -0.11864E-03
 BERR( 4) =    0.35973E-02
 BERR( 5) =    0.69028E-02

 Loop   2
 BERR( 1) =    0.13859E-06
 BERR( 2) =   -0.11574E-05
 BERR( 3) =    0.22282E-05
 BERR( 4) =   -0.46621E-03
 BERR( 5) =   -0.74232E-03
1
```

```
      ***** OFF DESIGN ENGINE CALCULATIONS. Converged after   2 Loops
*****



 ***** AMBIENT AND INLET PARAMETERS *****
 Alt. =      0.0      I.S.A. Dev. =    5.000       Mach No. =  0.00
 Etar = 0.9951        Momentum Drag =      0.00

 ***** COMPRESSOR  1 PARAMETERS *****
 PRSF =  0.23529E+02     ETASF =  0.10633E+01     WASF =  0.50798E+00
 Z = 0.84931             PR =  24.214             ETA = 0.88658
 PCN =   0.9893          CN = 0.98082             COMWK =  0.42710E+08

 ***** COMBUSTION CHAMBER PARAMETERS *****
 ETASF =  0.10000E+01
 ETA = 1.00000           DLP = 1.8132             WFB =   1.9020

 ***** TURBINE  1 PARAMETERS *****
 CNSF =  0.80319E+02     ETASF =  0.10508E+01     TFSF =  0.27256E+01
 DHSF =  0.18969E+05
 TF =  414.255            ETA = 0.89374            CN =  2.038
 AUXWK =  0.00000E+00

 ***** TURBINE  2 PARAMETERS *****
 CNSF = -0.68748E-02     ETASF =  0.10391E+01     TFSF =  0.69818E+00
 DHSF =  0.19225E+05
 TF =  413.348            ETA = 0.88538            CN =  2.059
 AUXWK =  0.32265E+08

 Additional Free Turbine Parameters:-
 Speed = *****%          Power =  0.32265E+08

 ***** CONVERGENT NOZZLE  1 PARAMETERS *****
 NCOSF =  0.10000E+01
 Area =  1.0241          Exit Velocity =  194.37  Gross Thrust =
16828.93
 Nozzle Coeff. =  0.97247E+00



 Scale Factor on above Mass Flows, Areas, Thrusts & Powers =
1.0000



 Station  F.A.R. Mass Flow   Pstatic    Ptotal Tstatic  Ttotal     Vel
Area
     1   0.00000    87.129   1.00000   1.00000  293.15  293.15     0.0
******
     2   0.00000    87.129    ******   0.99510  ******  293.15  ******
******
     3   0.00000    87.129    ****** 24.09544  ******  764.93  ******
******
     4   0.00000    84.951    ****** 24.09544  ******  764.93  ******
******
     5   0.00000    78.580    ****** 24.09544  ******  764.93  ******
******
```

```
     6   0.02420    80.482    ******  22.28220  ****** 1575.00  ******
******
     7   0.02239    86.853    ******  22.28220  ****** 1520.41  ******
******
     8   0.02239    86.853    ******   5.02166  ****** 1123.29  ******
******
     9   0.02183    89.031    ******   5.02166  ****** 1115.07  ******
******
    10   0.02183    89.031    ******   1.08493  ******  805.96  ******
******
    11   0.02183    89.031   1.00000   1.08493  789.32  805.96   194.4
1.0241
    12   0.00000     2.178    ******  24.09544  ******  764.93  ******
******
    13   0.00000     6.371    ******  24.09544  ******  764.93  ******
******


   Shaft Power =  32264696.00
    Net Thrust = 16828.93
  Equiv. Power =  33349774.00
     Fuel Flow =   1.9020
        S.F.C. =  58.9495
      E.S.F.C. =  57.0315
 Sp. Sh. Power =    370308.50
 Sp. Eq. Power =    382762.16
 Sh. Th. Effy. =   0.3934
          Time Now 18:30:43

 ***********************************************
 2  10.0  ! -
 -1
 -1
                                                   Time Now 18:30:43

 ***********************************************
 BERR( 1) =    0.52996E-02
 BERR( 2) =   -0.78435E-02
 BERR( 3) =   -0.75073E-02
 BERR( 4) =   -0.21648E-01
 BERR( 5) =   -0.63675E-01

 Loop   1
 BERR( 1) =   -0.58567E-04
 BERR( 2) =    0.77874E-04
 BERR( 3) =   -0.27658E-04
 BERR( 4) =    0.10363E-02
 BERR( 5) =    0.20961E-02
1

      ***** OFF DESIGN ENGINE CALCULATIONS. Converged after   1 Loops
*****



 ***** AMBIENT AND INLET PARAMETERS *****
 Alt. =      0.0      I.S.A. Dev. =   10.000       Mach No. =  0.00
 Etar = 0.9951        Momentum Drag =      0.00
```

```
 ***** COMPRESSOR  1 PARAMETERS *****
 PRSF =  0.23529E+02     ETASF =  0.10633E+01     WASF =  0.50798E+00
 Z = 0.84881             PR =  23.460             ETA = 0.89052
 PCN =   0.9788          CN = 0.96229             COMWK =  0.41239E+08

 ***** COMBUSTION CHAMBER PARAMETERS *****
 ETASF =  0.10000E+01
 ETA = 1.00000           DLP = 1.7628             WFB =   1.8350

 ***** TURBINE  1 PARAMETERS *****
 CNSF =  0.80319E+02     ETASF =  0.10508E+01     TFSF =  0.27256E+01
 DHSF =  0.18969E+05
 TF =  414.161            ETA = 0.89252            CN =  2.016
 AUXWK =  0.00000E+00

 ***** TURBINE  2 PARAMETERS *****
 CNSF = -0.68748E-02     ETASF =  0.10391E+01     TFSF =  0.69818E+00
 DHSF =  0.19225E+05
 TF =  412.295            ETA = 0.88578            CN =  2.058
 AUXWK =  0.30836E+08

 Additional Free Turbine Parameters:-
 Speed = *****%          Power =  0.30836E+08

 ***** CONVERGENT NOZZLE  1 PARAMETERS *****
 NCOSF =  0.10000E+01
 Area =  1.0241          Exit Velocity =  189.74  Gross Thrust =
15907.88
 Nozzle Coeff. =  0.97244E+00



 Scale Factor on above Mass Flows, Areas, Thrusts & Powers =
1.0000
```

Station	F.A.R.	Mass Flow	Pstatic	Ptotal	Tstatic	Ttotal	Vel	Area
1	0.00000	84.380	1.00000	1.00000	298.15	298.15	0.0	******
2	0.00000	84.380	******	0.99510	******	298.15	******	******
3	0.00000	84.380	******	23.34496	******	768.19	******	******
4	0.00000	82.270	******	23.34496	******	768.19	******	******
5	0.00000	76.100	******	23.34496	******	768.19	******	******
6	0.02411	77.935	******	21.58220	******	1575.00	******	******
7	0.02230	84.105	******	21.58220	******	1520.60	******	******
8	0.02230	84.105	******	4.87803	******	1124.61	******	******
9	0.02175	86.215	******	4.87803	******	1116.44	******	******

```
    10   0.02175    86.215    ******   1.08319  ******  811.53  ******
******
    11   0.02175    86.215   1.00000   1.08319  795.69  811.53   189.7
1.0241
    12   0.00000     2.109    ******  23.34496  ******  768.19  ******
******
    13   0.00000     6.170    ******  23.34496  ******  768.19  ******
******


   Shaft Power =  30835524.00
    Net Thrust = 15907.88
  Equiv. Power =  31861216.00
     Fuel Flow =   1.8350
        S.F.C. =  59.5096
      E.S.F.C. =  57.5939
 Sp. Sh. Power =    365438.47
 Sp. Eq. Power =    377594.19
 Sh. Th. Effy. =   0.3897
          Time Now 18:30:43

 ***********************************************
 2  15.0   ! -
 -1
 -1
                                                  Time Now 18:30:43

 ***********************************************
 BERR( 1) =    0.50154E-02
 BERR( 2) =   -0.75525E-02
 BERR( 3) =   -0.73933E-02
 BERR( 4) =   -0.19832E-01
 BERR( 5) =   -0.59387E-01

 Loop   1
 BERR( 1) =   -0.46141E-04
 BERR( 2) =   -0.61416E-04
 BERR( 3) =   -0.28407E-03
 BERR( 4) =    0.94965E-03
 BERR( 5) =    0.15030E-02
1

      ***** OFF DESIGN ENGINE CALCULATIONS. Converged after   1 Loops
*****



 ***** AMBIENT AND INLET PARAMETERS *****
 Alt. =      0.0      I.S.A. Dev. =   15.000       Mach No. =  0.00
 Etar = 0.9951        Momentum Drag =      0.00

 ***** COMPRESSOR  1 PARAMETERS *****
 PRSF =  0.23529E+02     ETASF =  0.10633E+01     WASF =  0.50798E+00
 Z = 0.84853             PR =  22.742             ETA = 0.89431
 PCN =   0.9688          CN = 0.94448             COMWK =  0.39841E+08

 ***** COMBUSTION CHAMBER PARAMETERS *****
 ETASF =  0.10000E+01
 ETA = 1.00000           DLP = 1.7144             WFB =   1.7711
```

```
***** TURBINE  1 PARAMETERS *****
CNSF =  0.80319E+02     ETASF =  0.10508E+01     TFSF =  0.27256E+01
DHSF =  0.18969E+05
TF =  414.067            ETA = 0.89120            CN =  1.995
AUXWK =  0.00000E+00

***** TURBINE  2 PARAMETERS *****
CNSF = -0.68748E-02     ETASF =  0.10391E+01     TFSF =  0.69818E+00
DHSF =  0.19225E+05
TF =  411.254            ETA = 0.88618            CN =  2.056
AUXWK =  0.29529E+08

Additional Free Turbine Parameters:-
Speed = *****%          Power =  0.29529E+08

***** CONVERGENT NOZZLE  1 PARAMETERS *****
NCOSF =  0.10000E+01
Area =  1.0241          Exit Velocity =  185.16  Gross Thrust =
15038.40
Nozzle Coeff. =  0.97236E+00



Scale Factor on above Mass Flows, Areas, Thrusts & Powers =
1.0000




Station  F.A.R. Mass Flow   Pstatic    Ptotal Tstatic  Ttotal     Vel
Area
     1   0.00000    81.756   1.00000   1.00000  303.15  303.15     0.0
******
     2   0.00000    81.756    ******   0.99510  ******  303.15  ******
******
     3   0.00000    81.756    ******  22.63010  ******  771.49  ******
******
     4   0.00000    79.712    ******  22.63010  ******  771.49  ******
******
     5   0.00000    73.733    ******  22.63010  ******  771.49  ******
******
     6   0.02402    75.504    ******  20.91565  ****** 1575.00  ******
******
     7   0.02222    81.483    ******  20.91565  ****** 1520.80  ******
******
     8   0.02222    81.483    ******   4.73952  ****** 1125.90  ******
******
     9   0.02166    83.527    ******   4.73952  ****** 1117.77  ******
******
    10   0.02166    83.527    ******   1.07801  ******  816.53  ******
******
    11   0.02166    83.527   1.00000   1.07801  801.46  816.53   185.2
1.0241
    12   0.00000     2.044    ******  22.63010  ******  771.49  ******
******
    13   0.00000     5.978    ******  22.63010  ******  771.49  ******
******
```

```
  Shaft Power =  29529098.00
   Net Thrust = 15038.40
 Equiv. Power =  30498728.00
    Fuel Flow =   1.7711
       S.F.C. =  59.9791
     E.S.F.C. =  58.0722
Sp. Sh. Power =    361187.62
Sp. Eq. Power =    373047.72
Sh. Th. Effy. =   0.3866
          Time Now 18:30:43

***********************************************
2  20.0   !
-1
-1
                                                  Time Now 18:30:43

***********************************************
BERR( 1) =    0.48069E-02
BERR( 2) =   -0.74972E-02
BERR( 3) =   -0.75246E-02
BERR( 4) =   -0.19583E-01
BERR( 5) =   -0.58468E-01

Loop   1
BERR( 1) =   -0.69743E-04
BERR( 2) =    0.12559E-03
BERR( 3) =   -0.56151E-04
BERR( 4) =    0.98036E-03
BERR( 5) =    0.16846E-02
1

     ***** OFF DESIGN ENGINE CALCULATIONS. Converged after   1 Loops
*****



***** AMBIENT AND INLET PARAMETERS *****
Alt. =      0.0      I.S.A. Dev. =   20.000       Mach No. =  0.00
Etar = 0.9951        Momentum Drag =      0.00

***** COMPRESSOR  1 PARAMETERS *****
PRSF =  0.23529E+02     ETASF =  0.10633E+01     WASF =  0.50798E+00
Z = 0.84880             PR =  22.009             ETA = 0.89824
PCN =   0.9576          CN = 0.92597             COMWK =  0.38381E+08

***** COMBUSTION CHAMBER PARAMETERS *****
ETASF =  0.10000E+01
ETA = 1.00000           DLP = 1.6637             WFB =   1.7079

***** TURBINE  1 PARAMETERS *****
CNSF =  0.80319E+02     ETASF =  0.10508E+01     TFSF =  0.27256E+01
DHSF =  0.18969E+05
TF =  413.969            ETA = 0.88919            CN =  1.972
AUXWK =  0.00000E+00

***** TURBINE  2 PARAMETERS *****
CNSF = -0.68748E-02     ETASF =  0.10391E+01     TFSF =  0.69818E+00
```

DHSF = 0.19225E+05
TF = 410.151 ETA = 0.88660 CN = 2.055
AUXWK = 0.28215E+08

Additional Free Turbine Parameters:-
Speed = *****% Power = 0.28215E+08

***** CONVERGENT NOZZLE 1 PARAMETERS *****
NCOSF = 0.10000E+01
Area = 1.0241 Exit Velocity = 180.55 Gross Thrust = 14183.65
Nozzle Coeff. = 0.97228E+00

Scale Factor on above Mass Flows, Areas, Thrusts & Powers = 1.0000

Station	F.A.R.	Mass Flow	Pstatic	Ptotal	Tstatic	Ttotal	Vel	Area
1	0.00000	79.090	1.00000	1.00000	308.15	308.15	0.0	******
2	0.00000	79.090	******	0.99510	******	308.15	******	******
3	0.00000	79.090	******	21.90137	******	774.22	******	******
4	0.00000	77.113	******	21.90137	******	774.22	******	******
5	0.00000	71.329	******	21.90137	******	774.22	******	******
6	0.02394	73.037	******	20.23765	******	1575.00	******	******
7	0.02215	78.820	******	20.23765	******	1520.96	******	******
8	0.02215	78.820	******	4.60176	******	1127.68	******	******
9	0.02159	80.798	******	4.60176	******	1119.57	******	******
10	0.02159	80.798	******	1.07377	******	822.20	******	******
11	0.02159	80.798	1.00000	1.07377	807.90	822.20	180.5	1.0241
12	0.00000	1.977	******	21.90137	******	774.22	******	******
13	0.00000	5.783	******	21.90137	******	774.22	******	******

Shaft Power = 28215066.00
Net Thrust = 14183.65
Equiv. Power = 29129584.00
Fuel Flow = 1.7079
S.F.C. = 60.5321
E.S.F.C. = 58.6317
Sp. Sh. Power = 356747.38
Sp. Eq. Power = 368310.44

```
 Sh. Th. Effy. =   0.3831
           Time Now 18:30:43

 ************************************************
 2  25.0   !
 -1
 -1
                                                   Time Now 18:30:43

 ************************************************
 BERR( 1) =    0.45593E-02
 BERR( 2) =   -0.71002E-02
 BERR( 3) =   -0.71479E-02
 BERR( 4) =   -0.19237E-01
 BERR( 5) =   -0.56799E-01

 Loop   1
 BERR( 1) =   -0.66648E-04
 BERR( 2) =    0.96845E-04
 BERR( 3) =   -0.40863E-04
 BERR( 4) =    0.93541E-03
 BERR( 5) =    0.15284E-02
1

      ***** OFF DESIGN ENGINE CALCULATIONS. Converged after   1 Loops
*****



 ***** AMBIENT AND INLET PARAMETERS *****
 Alt. =      0.0      I.S.A. Dev. =   25.000       Mach No. =  0.00
 Etar = 0.9951        Momentum Drag =      0.00

 ***** COMPRESSOR  1 PARAMETERS *****
 PRSF =  0.23529E+02     ETASF =  0.10633E+01     WASF =  0.50798E+00
 Z = 0.84923             PR =  21.317             ETA = 0.90198
 PCN =   0.9470          CN = 0.90838             COMWK =  0.37009E+08

 ***** COMBUSTION CHAMBER PARAMETERS *****
 ETASF =  0.10000E+01
 ETA = 1.00000           DLP = 1.6158             WFB =   1.6479

 ***** TURBINE  1 PARAMETERS *****
 CNSF =  0.80319E+02     ETASF =  0.10508E+01     TFSF =  0.27256E+01
 DHSF =  0.18969E+05
 TF =  413.860            ETA = 0.88723            CN =  1.950
 AUXWK =  0.00000E+00

 ***** TURBINE  2 PARAMETERS *****
 CNSF = -0.68748E-02     ETASF =  0.10391E+01     TFSF =  0.69818E+00
 DHSF =  0.19225E+05
 TF =  409.054            ETA = 0.88703            CN =  2.053
 AUXWK =  0.26975E+08

 Additional Free Turbine Parameters:-
 Speed = *****%          Power =  0.26975E+08

 ***** CONVERGENT NOZZLE  1 PARAMETERS *****
 NCOSF =  0.10000E+01
```

```
 Area =  1.0241          Exit Velocity =  176.13  Gross Thrust =
13392.66
 Nozzle Coeff. =  0.97221E+00



 Scale Factor on above Mass Flows, Areas, Thrusts & Powers =
1.0000


 Station  F.A.R. Mass Flow   Pstatic    Ptotal Tstatic  Ttotal     Vel
Area
     1   0.00000    76.565   1.00000   1.00000  313.15  313.15     0.0
******
     2   0.00000    76.565    ******   0.99510  ******  313.15  ******
******
     3   0.00000    76.565    ****** 21.21226  ******  777.07  ******
******
     4   0.00000    74.651    ****** 21.21226  ******  777.07  ******
******
     5   0.00000    69.052    ****** 21.21226  ******  777.07  ******
******
     6   0.02386    70.700    ****** 19.59651  ****** 1575.00  ******
******
     7   0.02207    76.299    ****** 19.59651  ****** 1521.13  ******
******
     8   0.02207    76.299    ******   4.46997  ****** 1129.37  ******
******
     9   0.02152    78.213    ******   4.46997  ****** 1121.28  ******
******
    10   0.02152    78.213    ******   1.06950  ******  827.75  ******
******
    11   0.02152    78.213   1.00000   1.06950  814.16  827.75   176.1
1.0241
    12   0.00000     1.914    ****** 21.21226  ******  777.07  ******
******
    13   0.00000     5.599    ****** 21.21226  ******  777.07  ******
******


   Shaft Power =  26975412.00
    Net Thrust = 13392.66
  Equiv. Power =  27838930.00
     Fuel Flow =   1.6479
        S.F.C. =  61.0885
      E.S.F.C. =  59.1937
 Sp. Sh. Power =    352320.44
 Sp. Eq. Power =    363598.66
 Sh. Th. Effy. =   0.3796
          Time Now 18:30:43

 ***********************************************
 2  30.0   !
 -1
 -1
                                                  Time Now 18:30:43

 ***********************************************
 BERR( 1) =    0.43447E-02
```

```
 BERR( 2) =   -0.69153E-02
 BERR( 3) =   -0.69867E-02
 BERR( 4) =   -0.18969E-01
 BERR( 5) =   -0.55507E-01

 Loop   1
 BERR( 1) =    0.19590E-02
 BERR( 2) =   -0.10949E-02
 BERR( 3) =    0.81850E-04
 BERR( 4) =    0.12754E-02
 BERR( 5) =    0.25787E-02
1

      ***** OFF DESIGN ENGINE CALCULATIONS. Converged after   1 Loops
*****



 ***** AMBIENT AND INLET PARAMETERS *****
 Alt. =      0.0      I.S.A. Dev. =   30.000       Mach No. =  0.00
 Etar = 0.9951        Momentum Drag =      0.00

 ***** COMPRESSOR  1 PARAMETERS *****
 PRSF =  0.23529E+02     ETASF =  0.10633E+01     WASF =  0.50798E+00
 Z = 0.84984             PR =  20.705             ETA = 0.90558
 PCN =   0.9367          CN = 0.89145             COMWK =  0.35769E+08

 ***** COMBUSTION CHAMBER PARAMETERS *****
 ETASF =  0.10000E+01
 ETA = 1.00000           DLP = 1.5691             WFB =   1.5907

 ***** TURBINE  1 PARAMETERS *****
 CNSF =  0.80319E+02     ETASF =  0.10508E+01     TFSF =  0.27256E+01
 DHSF =  0.18969E+05
 TF =  413.747            ETA = 0.88530            CN =  1.929
 AUXWK =  0.00000E+00

 ***** TURBINE  2 PARAMETERS *****
 CNSF = -0.68748E-02     ETASF =  0.10391E+01     TFSF =  0.69818E+00
 DHSF =  0.19225E+05
 TF =  407.953            ETA = 0.88748            CN =  2.052
 AUXWK =  0.25792E+08

 Additional Free Turbine Parameters:-
 Speed = *****%          Power =  0.25792E+08

 ***** CONVERGENT NOZZLE  1 PARAMETERS *****
 NCOSF =  0.10000E+01
 Area =  1.0241          Exit Velocity =  171.93  Gross Thrust =
12668.08
 Nozzle Coeff. =  0.97217E+00



 Scale Factor on above Mass Flows, Areas, Thrusts & Powers =
1.0000
```

```
 Station   F.A.R. Mass Flow   Pstatic    Ptotal Tstatic   Ttotal      Vel
Area
     1   0.00000    74.201   1.00000   1.00000  318.15  318.15      0.0
******
     2   0.00000    74.201    ******   0.99510  ******  318.15  ******
******
     3   0.00000    74.201    ****** 20.60394  ******  780.45  ******
******
     4   0.00000    72.346    ****** 20.60394  ******  780.45  ******
******
     5   0.00000    66.920    ****** 20.60394  ******  780.45  ******
******
     6   0.02377    68.511    ****** 19.03479  ****** 1575.00  ******
******
     7   0.02199    73.937    ****** 19.03479  ****** 1521.33  ******
******
     8   0.02199    73.937    ******   4.34627  ****** 1130.58  ******
******
     9   0.02144    75.792    ******   4.34627  ****** 1122.54  ******
******
    10   0.02144    75.792    ******   1.06687  ******  833.07  ******
******
    11   0.02144    75.792   1.00000   1.06687  820.13  833.07    171.9
1.0241
    12   0.00000     1.855    ****** 20.60394  ******  780.45  ******
******
    13   0.00000     5.426    ****** 20.60394  ******  780.45  ******
******


   Shaft Power =  25791794.00
    Net Thrust = 12668.08
  Equiv. Power =  26608594.00
     Fuel Flow =   1.5907
        S.F.C. =  61.6729
      E.S.F.C. =  59.7798
 Sp. Sh. Power =    347592.69
 Sp. Eq. Power =    358600.56
 Sh. Th. Effy. =   0.3760
          Time Now 18:30:43

 ***********************************************
 2  35.0   !
 -1
 -1
                                                  Time Now 18:30:43

 ***********************************************
 BERR( 1) =    0.78621E-02
 BERR( 2) =   -0.88880E-02
 BERR( 3) =   -0.66733E-02
 BERR( 4) =   -0.18077E-01
 BERR( 5) =   -0.52322E-01

 Loop   1
 BERR( 1) =   -0.76768E-04
 BERR( 2) =    0.22249E-03
 BERR( 3) =   -0.70046E-04
```

```
 BERR( 4) =    0.15074E-02
 BERR( 5) =    0.29030E-01

 Loop   2
 BERR( 1) =    0.31215E-04
 BERR( 2) =   -0.50003E-04
 BERR( 3) =   -0.44989E-05
 BERR( 4) =   -0.79330E-03
 BERR( 5) =   -0.76337E-02

 Loop   3
 BERR( 1) =   -0.71015E-05
 BERR( 2) =    0.12336E-04
 BERR( 3) =    0.28610E-05
 BERR( 4) =    0.19865E-03
 BERR( 5) =    0.18034E-02
1

      ***** OFF DESIGN ENGINE CALCULATIONS. Converged after   3 Loops
*****

 ***** AMBIENT AND INLET PARAMETERS *****
 Alt. =      0.0      I.S.A. Dev. =   35.000       Mach No. =  0.00
 Etar = 0.9951        Momentum Drag =      0.00

 ***** COMPRESSOR  1 PARAMETERS *****
 PRSF =  0.23529E+02     ETASF =  0.10633E+01     WASF =  0.50798E+00
 Z = 0.84769             PR =  20.010             ETA = 0.90963
 PCN =   0.9239          CN = 0.87242             COMWK =  0.34404E+08

 ***** COMBUSTION CHAMBER PARAMETERS *****
 ETASF =  0.10000E+01
 ETA = 1.00000           DLP = 1.5240             WFB =   1.5356

 ***** TURBINE  1 PARAMETERS *****
 CNSF =  0.80319E+02     ETASF =  0.10508E+01     TFSF =  0.27256E+01
 DHSF =  0.18969E+05
 TF =  413.608            ETA = 0.88282            CN =  1.902
 AUXWK =  0.00000E+00

 ***** TURBINE  2 PARAMETERS *****
 CNSF = -0.68748E-02     ETASF =  0.10391E+01     TFSF =  0.69818E+00
 DHSF =  0.19225E+05
 TF =  406.706            ETA = 0.88795            CN =  2.050
 AUXWK =  0.24629E+08

 Additional Free Turbine Parameters:-
 Speed = *****%          Power =  0.24629E+08

 ***** CONVERGENT NOZZLE  1 PARAMETERS *****
 NCOSF =  0.10000E+01
 Area =  1.0241          Exit Velocity =  167.72  Gross Thrust =
11956.57
 Nozzle Coeff. =  0.97212E+00

 Scale Factor on above Mass Flows, Areas, Thrusts & Powers =
1.0000
```

Station	F.A.R.	Mass Flow	Pstatic	Ptotal	Tstatic	Ttotal	Vel	Area
1	0.00000	71.799	1.00000	1.00000	323.15	323.15	0.0	******
2	0.00000	71.799	******	0.99510	******	323.15	******	******
3	0.00000	71.799	******	19.91246	******	782.41	******	******
4	0.00000	70.004	******	19.91246	******	782.41	******	******
5	0.00000	64.754	******	19.91246	******	782.41	******	******
6	0.02371	66.289	******	18.38844	******	1575.00	******	******
7	0.02194	71.540	******	18.38844	******	1521.45	******	******
8	0.02194	71.540	******	4.22253	******	1133.05	******	******
9	0.02139	73.335	******	4.22253	******	1125.00	******	******
10	0.02139	73.335	******	1.06431	******	839.53	******	******
11	0.02139	73.335	1.00000	1.06431	826.84	839.53	167.7	1.0241
12	0.00000	1.795	******	19.91246	******	782.41	******	******
13	0.00000	5.250	******	19.91246	******	782.41	******	******

```
   Shaft Power =  24629360.00
    Net Thrust = 11956.57
  Equiv. Power =  25400284.00
     Fuel Flow =   1.5356
        S.F.C. =  62.3486
      E.S.F.C. =  60.4563
 Sp. Sh. Power =    343031.22
 Sp. Eq. Power =    353768.44
 Sh. Th. Effy. =   0.3719
          Time Now 18:30:43
 **********************************************
 -3
```