Evaluation of a Combined Heat and Power (CHP) Retrofit For Supplemental on-site Power Generation at ENGEN oil Refinery

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This thesis work is dedicated to my late mother, Ramona Moodley, (passed on 31 May 2016) who has always inspired me to achieve academic excellence.

To my two toddlers whose births have been a source of inspiration, Dariuz Kriztiano Moodley and Krizelle Moodley, together with my supportive partner; Kristen Moodley, I have found the means to conclude this work amidst a demanding work-family-academic life.

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Abstract

This thesis is a critical evaluation of an opportunity project for on-site generation of electricity at the ENGEN refinery in Durban, South Africa. The key equipment discussed is a 2.5MW special purpose backpressure turbine which (prior to July of 2014), operated in continuous service as a compressor prime mover. The availability of the turbine since the plant decommissioning, has drawn business interest in a retrofit service application as a turbo-generator capable of electrical power production if re-engineered with a an optimal gearbox and electrical generator configuration.

The assessment method employed for the data extraction and calculations in this thesis is the “Plant performance triangle”. Historical and current process data are filtered for meaningful calculations and engineering analysis. Data segmentation methods are used to analyse the refinery operation at varying boiler loads where High Pressure (HP) steam at 40 barg is routed to the turbine and let down to 10 barg Medium Pressure (MP) header. The thesis evaluates the profitability of the devaluation of this steam by the isentropic steam expansion from thermal to mechanical to finally electrical energy, as opposed to isenthalpic (adiabatic) steam “let-down” (throttling) or pressure relief.

The design basis for the turbine operation is 42 tons/hr high-pressure (HP) steam to the turbine casing inlet. Calculations show that between 2.0 MW to 2.5 MW of electrical energy generation is possible with minimal additional consumption of HP steam from the refinery HP header. This is due to the steam load balancing of five onsite boilers between the high and medium pressure steam header mains. In essence, additional MP steam for power generation is “let-down” into the MP header resulting in the back-up of HP to MP “let-down” from parallel boilers into the MP header. By this, the refinery demand for steam at varying pressure headers is adjusted by automated boiler advanced control. The resultant economic value of electricity cost savings is approximately (conservatively – based on 2016 electricity prices) R9.9m per year. Two key parameters in the techno-economic assessment are fuel gas (combustible energy) and treated feed water cost. The cost of boiler feed water is assumed a fixed cost to the operation, however since the refinery steam headers require a mere 2.37 additional tons of HP steam to support the new turbine operation, added water costs do not pose a significant operating expense. Sensitivities are performed on varying water costs (R/kL), as this is a factor of the project profitability given the scarce water availability challenges in South Africa.

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1 Introduction

1.1 Background

Energy efficiency in large, complex industrial facilities is a growing area of focus for many businesses. South Africa, in recent years, has experienced compromised stability in the supply and demand of bulk electricity, creating a need for sustainable solutions to large industrial and residential consumers. Major industrial companies are generally investing in energy saving initiatives for longer-term profitability, reliability and growth.

The ENGEN crude oil refining complex in Durban, South Africa, has thereby undertaken to explore such energy saving initiatives in an effort to reduce imported electricity costs and its inherent environmental carbon footprint. ENGEN refinery (by design) is a 120 000 barrel/day integrated crude facility with onsite steam plants for various high, medium and low-pressure consumers. High pressure steam (HP) is supplied at 40 barg; superheated to 400°C, Medium pressure steam (MP) is drawn at 10 barg and 220°C (~40°C above saturation) and Low pressure (LP) steam is available at 140°C-150°C (~13-23°C above saturation).

In mid-2014, ENGEN acquired the full ownership of a joint venture lubes processing plant (SAFOR – South African Oil Refinery) situated on site at the Durban Refinery complex. The business deal concluded sole ownership of the process technology and equipment assets to ENGEN, part of which are currently unutilised. Among the assets acquired are two 40-barg HP steam boilers and two HP to MP backpressure turbines. The boilers are rated at MCR (maximum continuous rating) of 64 tons/hr each. The turbines are rated for a combined three MW shaft output with one larger 2.5 MW drive and a smaller 0.5 MW unit. Due to the decommissioning of some acquired process units at the SAFOR plant, the refinery steam balance has significantly changed in load demand and header let-down rates. The turbines are currently installed in a turbo-compressor configuration; however, the compressors are decommissioned and maintained under an equipment preservation plan.

An opportunity therefore exists to retrofit the existing unutilised turbine installation from a TC (turbo-compressor) configuration to a TG (Turbo-Generator) service enabling on onsite electricity generation capability. ENGEN facility is equipped with multiple HP boilers for HP steam on demand with capacity for surplus production. The operation of the turbines in TG service thereby requires careful analysis of the boiler loads and efficiencies and optimal let down rates to lower pressure headers. Due to additional steam demand required for TG service, careful design considerations are given to HP-MP header let down steam rates, boiler efficiencies, boiler feed water and fuel gas costs.

1.2 Investigative/Problem Statement

Two existing HP to MP steam turbines at the SAFOR sub-complex are currently decommissioned and being maintained under an equipment preservation plan. The 2.5 MW turbine had previously provided prime motive force for a propane compressor while the 0.5 MW turbine drive serviced an Inert gas compressor. Due to hydraulic constraints on the SAFOR to ENREF MP import/export line, a maximum of 50 tons/hr MP steam is transferable via the main MP control valve to the refinery from SAFOR and vice-versa. The design basis for the turbine steam flow is therefore set at 42 000 kg/hr to allow margin for good control valve operation (20% to 80% of valve operating range). The smaller steam turbine consumes approximately 15 000 kg/hr under normal operation. Due to MP steam export capacity constraints (a limitation to the full value obtainable for electricity generation), the smaller turbine is excluded from the envelope of this thesis.

The following key equipment design information is available:

- Purchase date: 08/11/1971 (November)
- Tag Name: C4002
- Type: API 612 Special Purpose Backpressure Multistage
- Description: Propane Refrigeration Compressor & Steam Turbine driver + Common Lube & Seal oil
- Design Horsepower: 2000kW
- Maximum Horsepower: 2500kW
- Design steam inlet pressure: 42 barg
- Design steam Inlet temperature: 388 °C
- Design steam Rate: 18.1 kg/kW/hr
- Max Allowable back pressure on casing: 12.0 barg

- Design steam Exhaust temperature: 270 °C
- Max Allowable temp at inlet Pressure: 400 °C
- Max Safe speed: 7820 rpm
- Normal operating steam flow: 36200 kg/hr at 18.1kg/kW/hr)

Since the decommissioning of the turbines in 2014, the two SAFOR boilers supplying motive steam to the C4002 turbine has operated at a turndown ratio of 2.37 (Ratio of Maximum operating load to current boiler load). This translates to a new average operating HP steam demand of 27 tons/hr per boiler, from previously 55tons/hr.

It is therefore proposed that C4002 turbine be retrofitted to a TG (Turbo Generator) set and uncoupled from the redundant compressor. For this re-configuration, all of ENREF's five boilers in operation are incorporated into the total refinery HP steam balance considering the available HP to MP export from SAFOR plant to the Refinery. The electrical shaft work derived is to thereby drive a generator and synchronise a 50Hz cycle to the refinery electrical grid main, resulting in a net reduction in ESKOM electricity import.

1.3 Aims and Objectives

This thesis is focussed on identifying and defining a design basis for a retrofit TC to TG process plant design.

The key aims of the study are to:

- Quantify the key operational differences with boiler and steam operation pre and post the mid-2014 SAFOR plant partial decommissioning.
- Design and develop the new process conditions required for the Turbine and Generator.
- Propose an optimal HP steam load distribution between the five available onsite boilers with integrated TG continuous service.
- Calculate the economic benefits associated with a continuous electrical supply to the grid main in TG service.
- Perform sensitivity analyses on various process and economic factors to identify optimal operation, profit and loss scenarios.

The key objectives of the study are to:

- Critically assess ENGEN refinery fuel gas molecular, combustion properties, LHV (Lower heating Value) and Energy density.
- Analyse and discuss fuel gas quality changes pre, and post mid-2014.
- Analyse pre and post individual boiler performances with respect to reduced load demands post mid-2014.
- Assess the viability of ENREF substation capacity for the new generator service.

1.4 Key Research Questions

In proposing a design basis for a retrofit TG configuration, various research questions are asked with intent to qualify the design as safe, profitable and sustainable. Key questions are summarised below:

- Is there sufficient historical and current process data of good quality for a definitive process plant study?
- Will mechanical and structural modifications be possible for a retrofit TG application? Has the refining industry previously attempted such retrofit service applications?
- Will ENGEN Refinery be able to satisfy and sustain the integrated steam network and several steam mains at a new operating point?
- How will the 19200MWh_e/month projected generation capacity tie into ENGEN electrical grid? Will this require additional work scope to the substation systems or upgraded hydraulic MP steam export infrastructure?
- Will the TG configuration justify capital expenditure as a business project? What will economic indicators of NPV and IRR reveal?

1.5 Limitations and Delimitations

In assessing the viability of a retrofit turbo-generator concept, operational and technical limitations are delimitations are considered below:

- A control valve exists at the SAFOR plant, which is manipulated to control the export MP steam from SAFOR MP header to the refinery main MP header. Control valve (8PV21) is designed for a maximum MP steam flow of 50 tons/hr. As a result, there is a clear limitation on the physical export capability of MP steam from SAFOR to the refinery MP main header. Since a turbo-generator service will devalue HP steam to MP steam, the HP steam load to the turbine is limited by the export valve capability at 50 tons/hr. For this reason, only the larger turbine (C4002) is considered for retrofit to a turbo-generator. The smaller 0.5kW turbine machine is capable of consuming 15 tons/hr HP steam that cannot be exported to the refinery MP main nor consumed at SAFOR plant. Due to this limitation, the net electricity generation potential is limited to a turbine HP steam load of 42 tons/hr, which sets the design basis for this study.

- During data collection and analysis, it was noted that the instrumentation around boiler 3 at North Complex is not reliable. Calculations around boiler efficiency, energy balances and fired duty did not provide sensible results. A limitation is thereby inherent in the cost of steam calculation; therefore, boiler 3 data was excluded from the averaging of the final steam cost.
- ENREF is limited by technical and operational skills in turbo-generator operation. The skills gap is seen as a limitation to the fast tracking of the project and preferably be addressed by operator/engineering training in line with the project progression.

1.6 Significance of the Study

Engen Petroleum as a company has adopted energy efficiency as part of the company's long-term strategic initiatives. This study as a result ties directly into the basket of projects driving the strategic intent of the business. The study is therefore significant as summarised below:

- Reduction in import of ESKOM supplied electricity implies that ENGEN will reduce its reliability on externally sourced power supply for at least 48 MWh/day. The net result is that the 2.0 to 2.4 MW on-site generation will be available on demand at the ENGEN grid, thereby providing an emergency electrical backup for critical equipment operation within the demand range of 48MWh/day.
- Project economics indicate that a payback period of 2.2 years for the capital expenditure required makes the project financially attractive. As a business opportunity, ENGEN would therefore be positioned to benefit from electricity generation throughout the annual production operation cycle. This is due to the added advantage that during planned production downtime, boilers are often in service for steam cleaning. Continuous operation will allow revenue generation by power purchase agreements with ESKOM.

- Energy companies in South Africa are being incentivised by the concept of a "Carbon Credit Economy", as defined "In the carbon economy, so-called "carbon-income" is basically derived from the trade in Certified Emission Reduction credits, more generally referred to as "carbon credits", which are yielded or produced by qualifying GHG mitigation projects. ", (Scholtz, 2011). Engen Petroleum will therefore seek to exploit these incentives as further cost savings in addition to contribute to a cleaner fuels production strategy.
- The data collection as used in this study will provide valuable insight into future development of energy optimisation at Engen Petr oleum. Data mining, conditioning and segregation as used for this study will be filed at ENGEN library for reference. The methodology of the Plant performance triangle approach, will serve to enrich future engineering studies around the refinery operation holistically.

2 Literature Review/Theory

2.1 Industrial CHP

Growing demand for distributed energy systems supplying heat and power is placing a premium on flexibility of design from manufacturers servicing an incredibly diverse market from mechanical drives, process heat and steam, and power generation.

Demand for industrial steam turbines supplying both heat and power is increasingly being driven by growth in decentralized energy applications.

Much of this growth may be attributed to the push for low-carbon energy, one of the biggest global drivers for this sector.

There are many other growth drivers though. For example, in Europe various environmental policies, such as the 2005 EU Directive prohibiting landfill of non-treated waste, have seen waste-to-energy activities rapidly intensify.

Steam turbines are one of the major technologies that can support this shift towards decarbonisation, a trend that is especially relevant for the power generation sector and across various industrial branches.

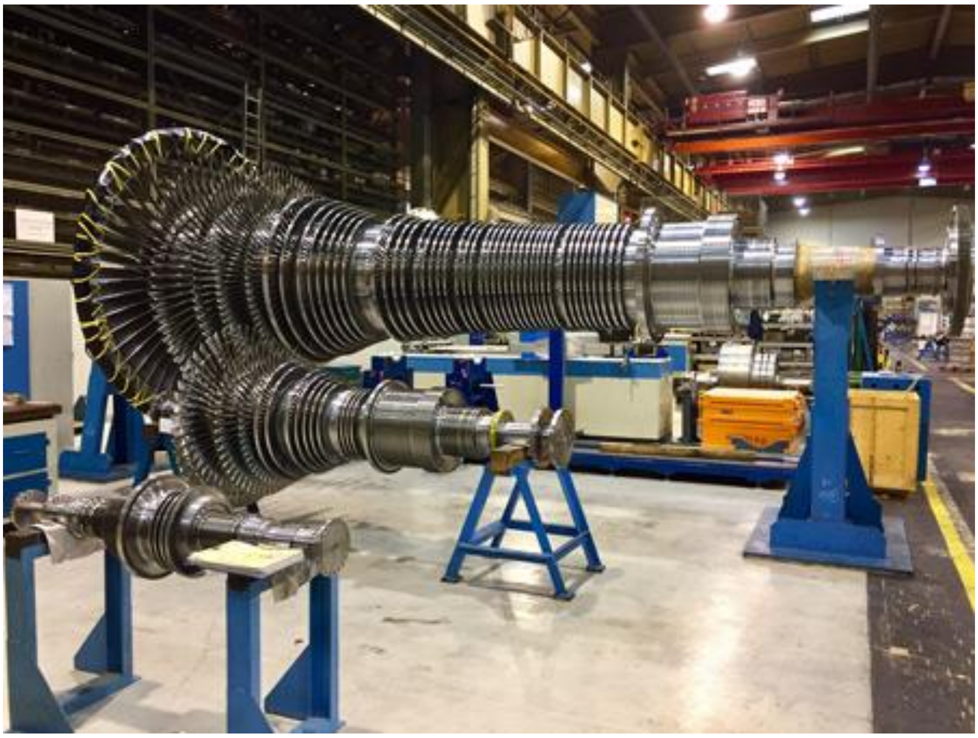
The steam turbine market for distributed heat and power is not only large and expanding, it is also extremely heterogeneous. Applications include the full range of power generation and mechanical drives, as well as the specific demands of various industrial processes:

A natural consequence of the diversity found in the small and medium-sized steam turbine sector is seen in the range of supply requirements. While a typical biomass plant may be 5 to 15 or even 20 MW, a waste-to-energy plant may start in the 10-15 MW range and reach 80 or 100 MW.

Similarly, a pulp and paper plant may need a turbine of more than 100 MW, serving several hundred tonnes per hour of steam at, say, 30 bar, 12 bar and 5 bar for different processes such as drying and finishing wood products, while a brewery may have completely different requirements. Each of these sectors not only has different demands, but those demands may also vary between seasons and across the day.

Responding to the need for uniquely optimised steam turbines, major manufacturers have looked to support this diverse market with reliable, flexible high efficiency steam turbines that are based on a modular design philosophy.

The modular platform allows a wide range of bespoke designs to be developed suitable for application in a variety of industries - given different requirements for process steam, power and heat and requiring a variety of solutions regarding steam offtake, condenser presents high-efficiency characteristics while supplying process steam and producing power across the load range for both outputs. Ultimately, the steam path - the number of stages, the angle of the blades, and the positions of the various bleeds and so on - is key to good performance (Siedel, 2017).



SCREENCAP 2-1 – STEAM TURBINE AND SHAFT

WORKSHOP ASSEMBLY – STEAM TURBINE CREDIT: MAN DIESEL & TURBO WEBSITE

2.2 Review of Mavainsa Report

Mavainsa is a company located in Valencia, Spain. According to the company services descriptor on their website: "MAVAINSA is mainly dedicated to the design, assembly, legalization and maintenance of industrial facilities for fluids in general: steam, hot and cold water, thermal oil, compressed air, fire-fighting networks, oil installations (fuel: fuel oil, gas-oil), water, gas and industrial cold treatments." (www.mavainsa.com)

This thesis references their November 2002 publication in "Chemical Engineering" entitled: Steam Balance Optimisation Strategies".

Mavainsa fully supports the concept of utilising high-pressure steam for backpressure turbine operation and thereby decreasing electricity consumption, as opposed to isenthalpic let-down via a PRV (Pressure reducing valve). Mavainsa mentions, *"the cost of electricity saved by operating a turbine to drive a pump or compressor instead of an electric motor is much higher than the cost of the fuel that would be required to superheat the turbine steam exhaust to the same temperature as the let-down valve outlet."*

This is clearer explained in the diagrammatic representation below:

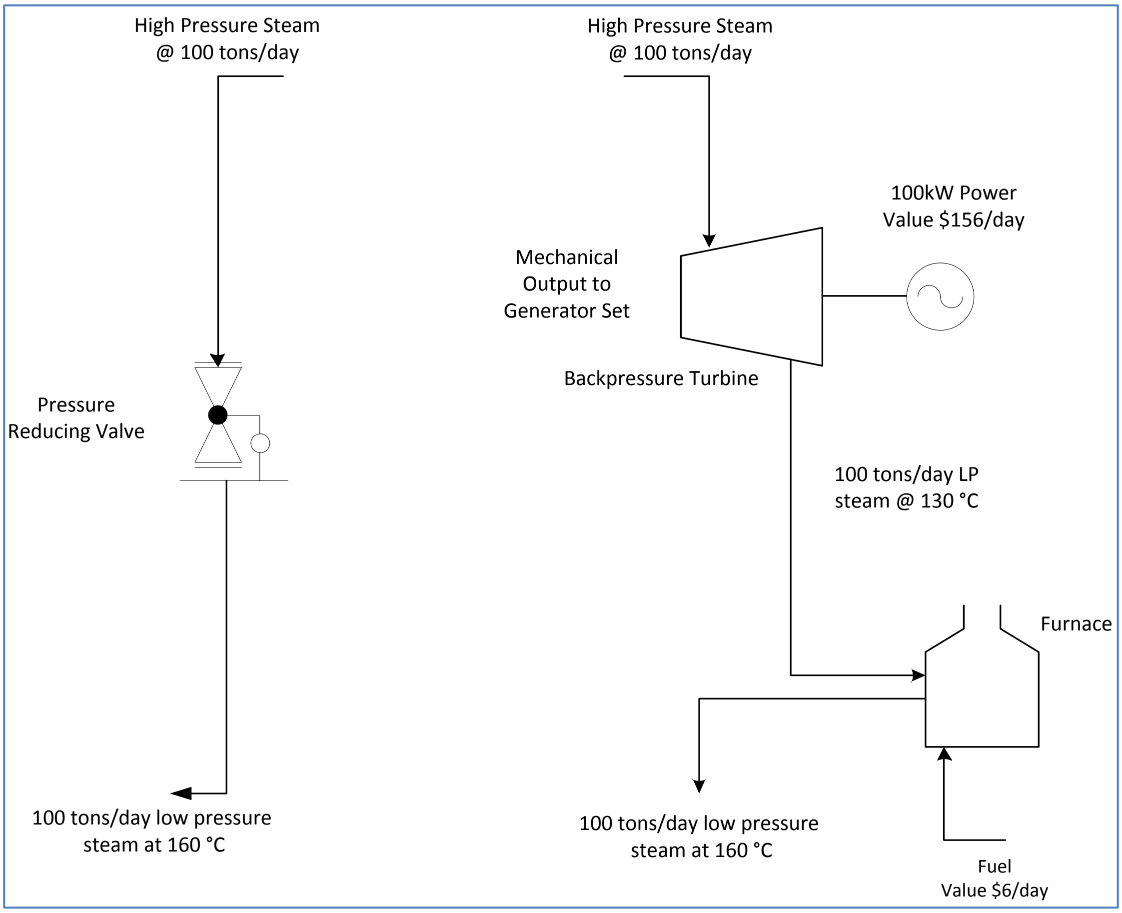


FIGURE 2-1 – COMPARISON OF HP STEAM PRESSURE REDUCTION SCENARIOS

– Adapted from Mavainsa Report (Mavainsa, November 2002), pp2.

In the example above (scheme on left figure) 100tpd (tons per day) HP steam is "let down" to LP steam at 160°C. The pressure reduction results in superheated LP steam at the same enthalpy as the HP steam (theoretically referred as isenthalpic Let-down). In option 2, by utilising a turbine to expand the steam, the exhaust temperature is reduced to 130°C however a mechanical shaft work output is generated by the steam enthalpy change equivalent to 100kW. It is estimated that this is valued at $156/day.

Should the exhaust LP steam be required for consumers at 160 °C by design (which is uncommon practice) a further 100kW of heat input to raise the temperature to 160°C will cost $6 per day. It is still calculated to be profitable by $150 per day by utilising a turbine generator concept over a PRV. The process control philosophy of steam header pressures integrated with electricity generation is explained by the concept of a "turbine by-pass".

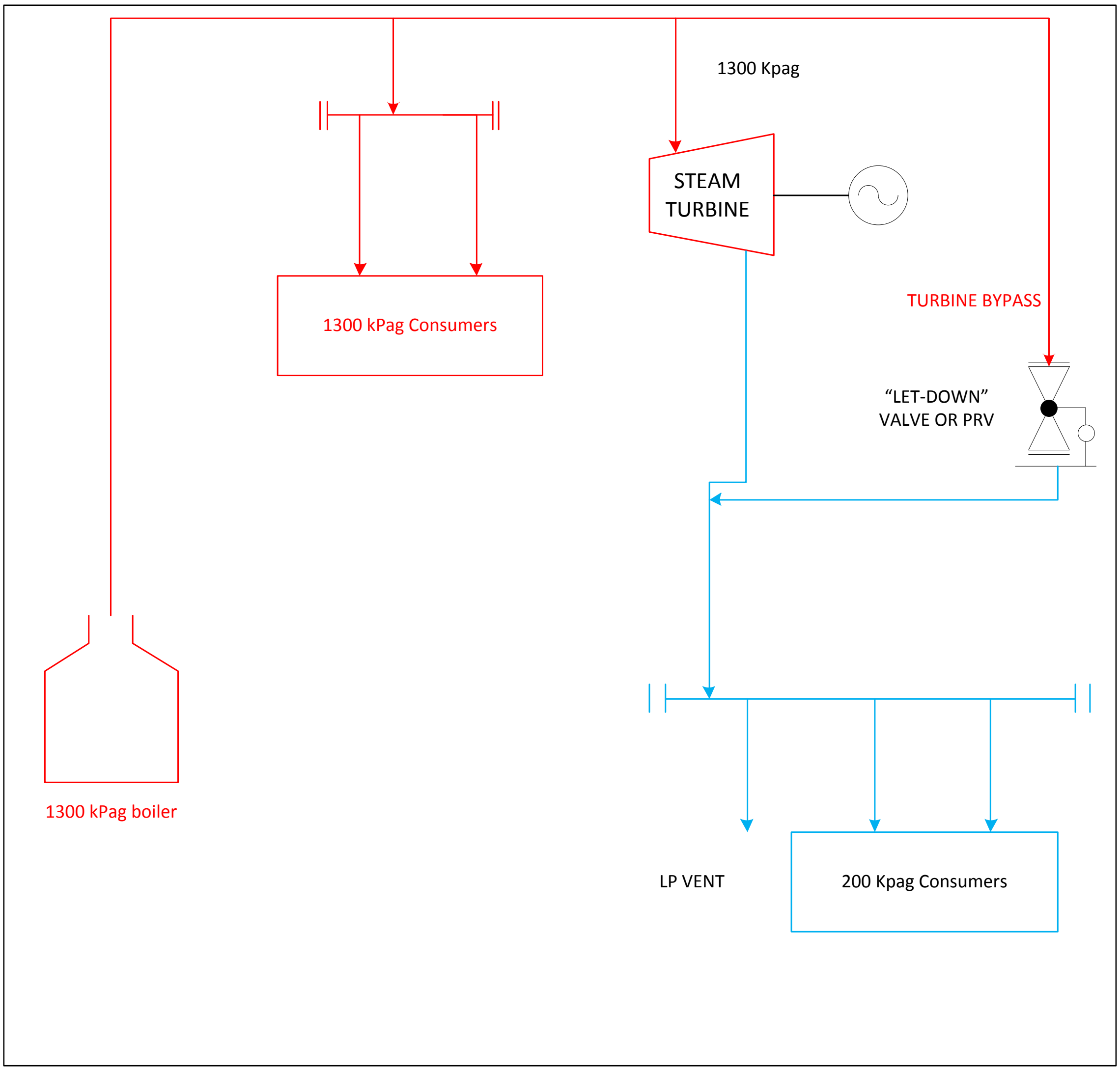


FIGURE 2-2 – TURBINE BY-PASS CONCEPT – ADAPTED FROM - (MAVAINSA, NOVEMBER 2002), PP 4

In the above operating scenario, a boiler produces steam at 1300 kPag for use by downstream consumers and a turbo-generator. A part of the steam is drawn by the turbine while an automated bypass across the turbine satisfied the LP steam demand. By utilisation of this control scheme, the turbine demand rate will remain optimised and stable. However, for steady operation, a small excess of produced steam must be available and simultaneously balanced with little or no LP steam venting.

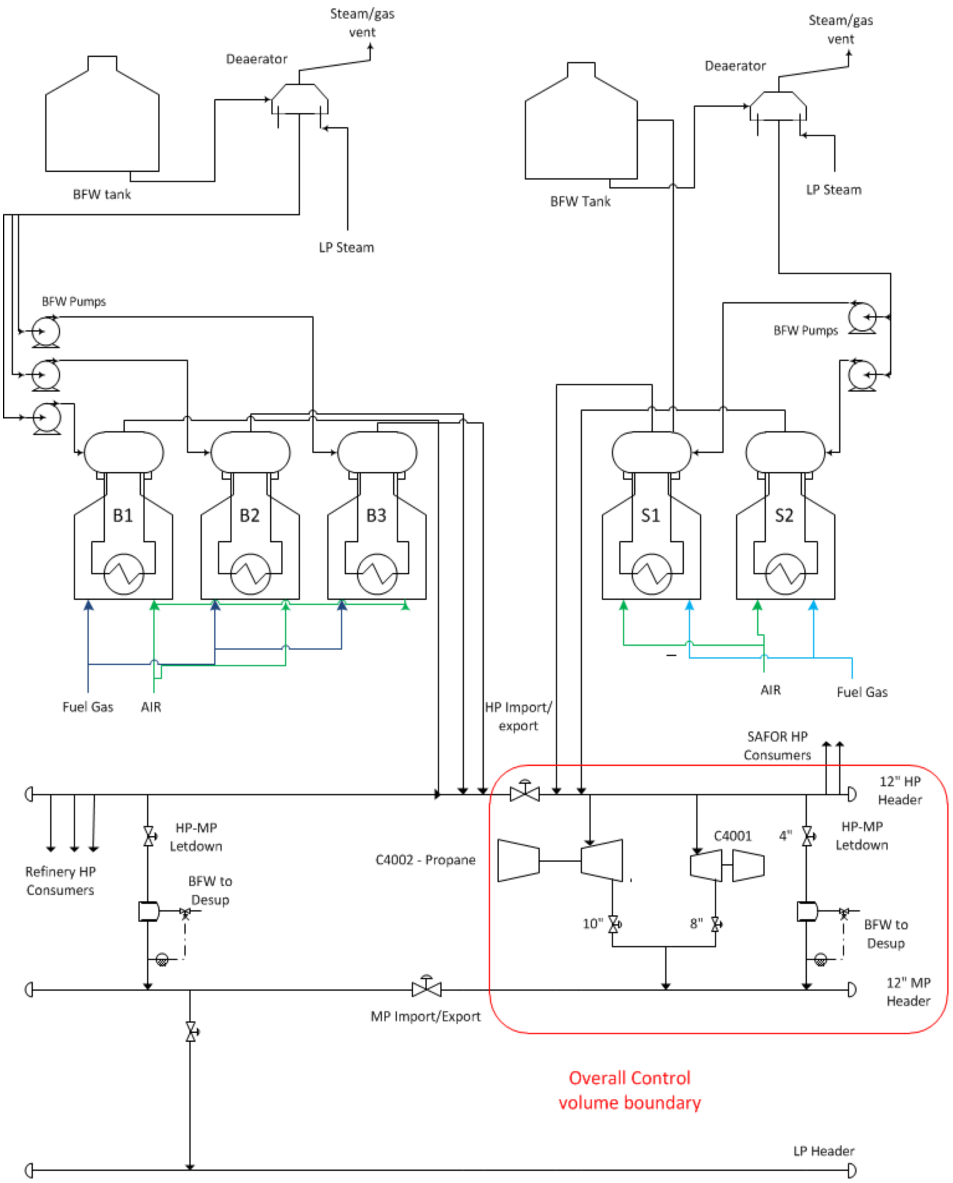
Later in the report, Mavainsa explains the advantages of utilising thermo-compressors as a means to further exploit energy savings by the concept of compressing available LP steam while simultaneously reducing high-pressure steam for process operation. The example given in the report is that of stripping steam used in refinery crude distillation towers. This example is a very viable and practical example since commercial crude distillation towers usually utilise stripping steam for improved crude distillation separation efficiency. This is to be further explored within ENREF as a means to compress and throttle LP vented steam to a higher pressure and combine with higher-pressure headers. The use and optimisation strategies of thermo-compressors are outside the scope of this thesis.

2.3 ENREF Water/Steam Process Overview

ENREF imports metro water supplemented by condensate return as feed water supply for five onsite boilers. Two boiler feed water (BFW) tanks store demineralised (softened) water, one each, at the North Complex site and SAFOR. De-aerators are installed on both plants for the supply of gas freed boiler feed water. The North Complex plant is equipped with three boilers supplying high-pressure steam to a header while SAFOR is equipped with two boilers. All boilers are fuel gas fired with air drawn from air compressors. Two pressure relief stations are available for let-down of HP to MP steam. The turbine prime driver that previously provided motive drive for a compressor is currently under preservation and shown in the figure coupled to a generator as proposed. Figure 1 also depicts 4 colour schemes; blue piping connections reflect cold water circuits, red denotes hot circuits, orange denotes LP steam. The rectangular double boxed light blue frame depicts the envelope for the new proposed turbo generator configuration. In this thesis, critical analyses and emphasis is placed on the equipment and processes within the design envelope. The key associated process equipment at SAFOR are:

- BFW pumps
- Boilers – F8001 & F8001s
- HP-MP steam turbine (C4002 considered only)
- IMPORT/EXPORT control valve

- Fuel gas to boilers
- Air to boilers
- Fail safe control systems
- A proposed new generator set



Legend:

B1 – North Complex Boilers (1 to 3)

S1 – SAFOR Boilers (1 to 2)

FIGURE 2-3 – ENREF HP, MP AND LP STEAM MAINS

Depicted above is a simplified process flow layout of ENREF water, steam and pressure header systems. This design of multiple steam mains systems is common for commercial crude refineries. The red single-bordered line depicts the specific water-steam and turbo-generation system applicable to the detail in this thesis.

Metro/corporation water is stored in large holding tanks, before being pumped to the North and SAFOR water treatment plants. Both North complex and SAFOR have demineralisation plants for softening of water. During this process, “hard” components in water (Ca and Mg) are removed by ionic exchange. The purpose of this process is to prevent downstream scale formation on piping. The resultant softened water is then stored in boiler feed water tanks before de-aeration. De-aerators are used to remove trapped gasses (mostly oxygen) in water, which have a tendency to reduce plant thermal efficiency. De-aerated water is thereafter fed to the boilers at a rate determined by the HP steam demand, from the specific boiler. This is an automated control system based on the header pressure main supply. HP steam from SAFOR boilers are consumed by HP consumers at SAFOR while excess HP steam is exported to the refinery header main. A similar infrastructure exists for the MP and LP header mains.

In the retrofit application, the HP steam consumer (C4002 turbine) will change service as a turbo compressor to a turbo generator. Since the SAFOR boilers are currently underutilised, a new operating point will be proposed for the new service. These are further discussed in later chapter

2.4 Thermodynamic Heat Cycle

What is Rankine Cycle?

Source - (www.differencebetween.com, n.d.)

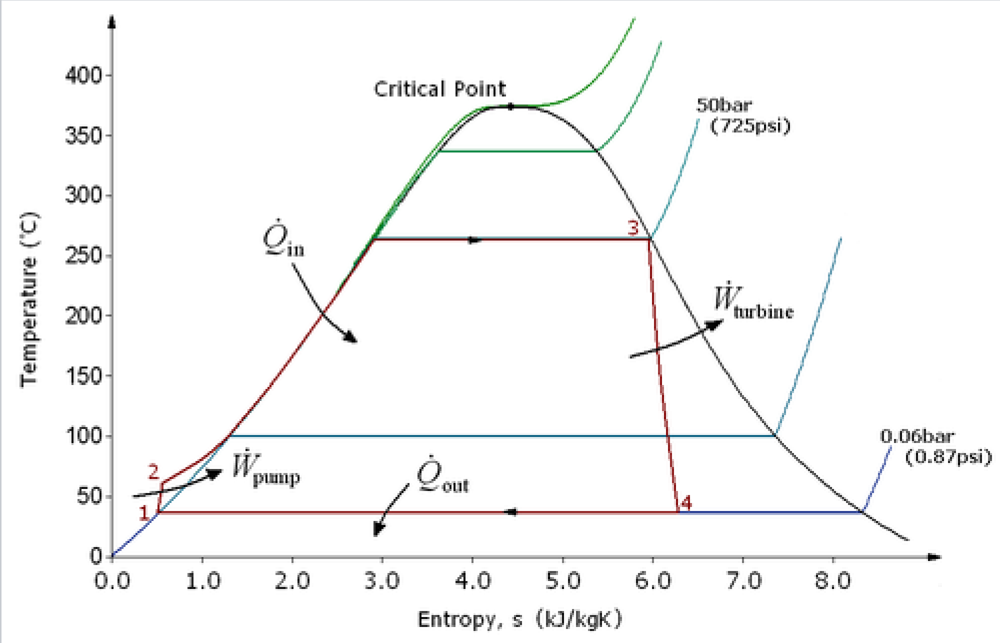
Rankine cycle is also a cycle, which converts heat into work. The Rankine cycle is a practically used cycle for systems consisting of a vapour turbine. There are four main processes in the Rankine cycle

1. The working of fluid into high pressure from a low pressure

2. The heating of the high-pressure fluid into a vapour

3. The vapour expands through a turbine turning the turbine, thereby generating power

4. The vapour is cooled back inside the condenser.



T–s diagram of a typical Rankine cycle operating between pressures of 0.06 bar and 50 bar. Left from the bell-shaped curve is liquid, right from it is gas, and under it is saturated liquid–vapour equilibrium.

FIGURE 2-4- T-S DIAGRAM – EXAMPLE RANKINE CYCLE (WWW.WIKIPEDIA.ORG, N.D.)

The four processes in the Rankine cycle are depicted graphically on a T-S diagram above.

Process 1–2: The working fluid is pumped from low to high pressure. As the fluid is a liquid at this stage, the pump requires little input energy.

Process 2–3: The high-pressure liquid enters a boiler, where it is heated at constant pressure by an external heat source to become a dry saturated vapour. The input energy required can be easily calculated graphically, using an enthalpy–entropy chart (h–s chart, or Mollier diagram), or numerically, using steam tables.

Process 3–4: The dry saturated vapour expands through a turbine, generating power. This decreases the temperature and pressure of the vapour, and some condensation may occur. The output in this process can be easily calculated using the chart or tables noted above.

Process 4–1: The wet vapour then enters a condenser, where it is condensed at a constant pressure to become a saturated liquid.

(www.wikipedia.org, n.d.)

2.5 Steam Turbine Model

Steam turbines are modelled by the principle of conservation of energy in a control volume. Energy is transferred to or from a system by work (W) and/or heat (Q). Work is done by a force moving through a distance (or its equivalent, as e.g. in the case of electrical work). Neither work nor heat is a property of the system (a state variable) so neither differential can be integrated without specifying a path. This is denoted by using δ rather than d in the expression (Moran, MJ., and Shapiro, H.N., 2008).

Reference is made to, path 3 – 4 from *Figure 2-4- T-S Diagram – Example Rankine Cycle* , page 14, for clarity of the state changes.

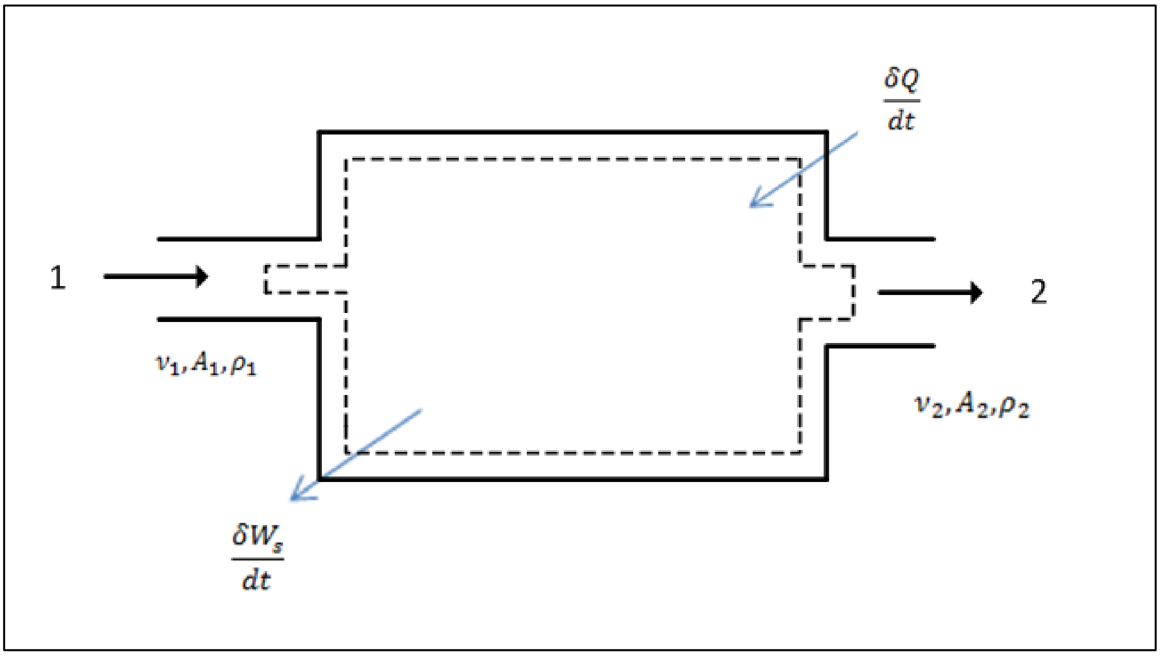


FIGURE 2-5- CONTROL VOLUME WITH ONE-DIMENSIONAL FLOW ACROSS BOUNDARIES

The illustration above has been adapted from (Moran, MJ., and Shapiro, H.N., 2008).

The full derivation of the integral expression in (James R.Welty, Charles E. Wicks, Robert E. Wilson, 1984) is however rigorous and academic for the purposes of this thesis, therefore a discretised and simplified form as derived by (Moran, MJ., and Shapiro, H.N., 2008), will be cited below.

$$\int_1^2 \boldsymbol{\delta W = W}$$ EQUATION 2-1

The rate of energy transfer by work is called *power*, denoted by $\dot{W}$, where in one dimension.

Work and Heat

$$\dot{\boldsymbol{W}} = \boldsymbol{FV}$$ EQUATION 2-2

Where F is Force (N) and V is velocity ($^m/_s$)

Similarly,

$$\int_1^2 \boldsymbol{\delta Q = Q}$$ EQUATION 2-3

The net *rate of energy transfer by heat is* $\dot{Q}$, and if it is known how $\dot{Q}$ varies with time, then

$$\boldsymbol{Q} = \int_1^2 \dot{\boldsymbol{Q}}\boldsymbol{dt}$$ EQUATION 2-4

The net rate of energy transfer as heat is related to the *heat flux* $\dot{q}$, the rate of heat transfer per unit area, by

$$\boldsymbol{Q} = \int_A \boldsymbol{q\dot{d}A}$$ EQUATION 2-5

Energy Rate Balance

The closed system energy balance is not the familiar $\Delta U = Q - W$, but

$$\boldsymbol{\Delta E = \Delta U + \Delta EK + \Delta PE}$$ EQUATION 2-6

$$= Q - W$$

Where EK and PE are the terms for kinetic and potential energy. The differential form is

$$\boldsymbol{dE = \delta Q - \delta W}$$ EQUATION 2-7

In addition, the instantaneous time rate form of the energy balance

$$\frac{dE}{dt} = Q - W$$

EQUATION 2-8

Or

$$\frac{dE}{dt} = \frac{dKE}{dt} + \frac{dPE}{dt} + \frac{dU}{dt} = \dot{Q} - \dot{W}$$ EQUATION 2-9

For one inlet (*i*), one outlet (*e*), 1D flow across a control volume, then

$$\frac{dE_{cv}}{dt} = \dot{Q} - \dot{W} + \dot{m}_i\left(u_i + \frac{v_i^2}{2} + gz_i\right) - \dot{m}_e\left(u_e + \frac{v_e^2}{2} + gz_e\right)$$ EQUATION 2-10

Where E_{cv} is the energy across the CV at time *t*, $\dot{Q}$ *and* $\dot{W}$, are the net rates of energy transfer as heat and work across the boundary of the CV at time t, *u* is the specific internal energy, *g* is the acceleration due to gravity and z is the elevation of the CV. If there is no mass flow the equation reduces to equation (2.2.6).

2.5.1 Evaluating Work for a CV

It is convenient to separate the net rate of energy transfer as work into or out of a CV ($\dot{W}$) into two parts. One is the rate of work done by the fluid pressure at the inlet and outlet as mass transported in or out. The other, called $\dot{W}_{CV}$, is the rate of *all other work*, such as done by rotating shafts, electrical work, etc.

The rate of energy transfer by work is force (N) x velocity ($^{m}/_{s}$), equation (2.2.1), so at the outlet, say,

$$\dot{W} = (p_e A_e)V_e$$ EQUATION 2-11

Where p_e = Pressure exiting the CV (Pa)

A_e = Area of the exit nozzle (m^2)

V_e = Velocity at the exit ($^{m}/_{s}$)

And similarly, for the inlet, so the work rate term for equation (2.5.8) is

$$\dot{W} = \dot{W}_{CV} + (p_e A_e)V_e + (p_i A_i)V_i \qquad \text{Equation 2-12}$$

And because $AV = \dot{m}v$

$$\dot{W} = \dot{W}_{CV} + \dot{m}_e(p_e v_e) - \dot{m}_i(p_i v_i) \qquad \text{Equation 2-13}$$

The terms $\dot{m}_e(p_e v_e)$ and $\dot{m}_i(p_i v_i)$ account for the work associated with the pressure at the outlet and inlet and are called *flow work.*

2.5.2 The Energy rate balance

Inserting this relation into equation 2.2.8 gives Equation 2-14 (below)

$$\frac{dE_{cv}}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \dot{m}_i\left(u_i + p_i v_i + \frac{v_i^2}{2} + gz_i\right) - \dot{m}_e\left(u_e + p_e v_e + \frac{v_e^2}{2} + gz_e\right)$$

Equation 2-15 (above)

Subscript CV is added to $\dot{Q}$ to emphasise that this is the rate of heat transfer over the surface of the CV. And because $h = u + pv$ where h is specific enthalpy ($^{kJ}/_{kg}$), this becomes

$$\boldsymbol{\frac{dE_{cv}}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \dot{m}_i\left(h_i + \frac{v_i^2}{2} + gz_i\right) - \dot{m}_e\left(h_e + \frac{v_e^2}{2} + gz_e\right)}$$

Equation 2-16 (above)

This is the master 1D, one inlet, and one outlet form of the energy balance for a CV. It only remains to relate $\dot{Q}_{cv}$ to entropy.

2.5.3 Steady state form of the Energy Balance

When $\dot{m}_i = \dot{m}_e$ and $dm_{cv}/dt = 0$, equation (2.2.13) becomes

$$\mathbf{0} = \frac{\dot{Q}_{cv}}{\dot{m}} - \frac{\dot{W}_{cv}}{\dot{m}} + (\boldsymbol{h_i} - \boldsymbol{h_e}) + \frac{v_i^2 - v_e^2}{2} + \boldsymbol{g}(\boldsymbol{z_i} - \boldsymbol{z_e}) \qquad \text{EQUATION 2-17}$$

2.5.4 The Entropy Balance

Entropy Balance for closed systems

The focus is on the *balance*, which means there is an explicit term σ representing the entropy difference between the real process and the process carried out reversibly, i.e., the amount of entropy produced in the system by *irreversibilities.*

Thus

$$\boldsymbol{S_2} - \boldsymbol{S_1} = \int_1^2 \left(\frac{\delta Q}{T}\right)_b + \boldsymbol{\sigma} \qquad \text{EQUATION 2-18}$$

In words this is

Change in entropy in the system during some time interval	=	Amount of entropy transferred into the system during the time interval	+	Entropy produced in the system during the interval

And in the differential form

$$\boldsymbol{dS} = \left(\frac{\delta Q}{T}\right)_{int\,rev} + \boldsymbol{\delta\sigma} \qquad \text{EQUATION 2-19}$$

A distinction is made between internal irreversibilities, those taking place in the system, and external irreversibilities, those taking place in the environment. Engineering design thus focusses on identifying the sources of the irreversibilities and reducing them. Common sources are (Moran, MJ., and Shapiro, H.N., 2008) pp.220:

1. Heat transfer due to ΔT
2. Unrestrained expansion of a fluid

3. Spontaneous chemical reaction (including phase changes)
4. Spontaneous mixing
5. Friction; sliding as well as within fluids
6. Current flow through a resistance
7. Magnetization or polarization with hysteresis
8. Inelastic deformation

All actual processes are irreversible, i.e., they contain irreversibilities and hence produce entropy.

If temperature is constant, equation 2-20 becomes

$$\boldsymbol{s_2 - s_1 = \frac{Q}{T_b} + \sigma} \qquad \text{EQUATION 2-21}$$

Where ${Q}/{T_b}$ represents the amount of entropy transferred through a portion of the system boundary at temperature T_b. Similarly, ${\dot{Q}}/{T_j}$ represents the *time rate* of entropy transfer through a portion of the boundary whose instantaneous temperature is T_j. The closed system entropy balance is then

$$\boldsymbol{\frac{dS}{dT}} = \sum_j \boldsymbol{\frac{\dot{Q}}{T_j} + \dot{\sigma}} \qquad \text{EQUATION 2-22}$$

Time rate of change of entropy in the system	=	(Sum of) time rate of entropy transfer through the portion(s) of the boundary	+	Time rate of entropy production due to irreversibilities in the system

2.5.5 Entropy balance for control volumes

Entropy is an extensive property, so it can be transferred in or out of systems by streams of matter. Modifying equation (2.2.18) gives

$$\frac{dS_{cv}}{dt} = \sum_j \frac{\dot{Q}_j}{T_j} + \sum_i \dot{m}_i s_i - \sum_e \dot{m}_e s_e + \dot{\sigma}_{cv} \qquad \text{Equation 2-23}$$

Where dS_{cv}/dt represents the time rate of change of entropy within the CV, $\dot{Q}_j$ represents the time rate of heat transfer at the point on the boundary where the instantaneous temperature is T_j, $\frac{\dot{Q}_j}{T_j}$ accounts for the accompanying rate of entropy transfer, $\dot{m}_i s_i$ and $\dot{m}_e s_e$ account for rates of *entropy transfer accompanying mass flow*, into and out of the CV and $\dot{\sigma}_{cv}$ denotes the time rate of entropy production due to irreversibilities within the CV.

2.5.6 Rate Balance for Control Volumes at Steady State

The steady state form of (2.2.19) is obtained by setting $dS_{cv}/dt = 0$. The one inlet, one outlet form is then

$$0 = \sum_j \frac{\dot{Q}_j}{T_j} + \dot{m}(s_i - s_e) + \dot{\sigma}_{cv} \qquad \text{Equation 2-24}$$

Or

$$(s_i - s_e) = \frac{1}{\dot{m}}\left(\sum_j \frac{\dot{Q}_j}{T_j}\right) + \frac{\dot{\sigma}_{cv}}{\dot{m}} \qquad \text{Equation 2-25}$$

The two terms on the right are now per unit mass flowing through the CV. If there is no heat transfer,

$$(s_i - s_e) = \frac{\dot{\sigma}_{cv}}{\dot{m}} \qquad \text{Equation 2-26}$$

So when there are irreversibilities within the CV, unit mass entropy increases as it passes from inlet to outlet, and when no irreversibilities are present $\dot{\sigma}_{cv} = 0$, $s_1 - s_2$, and the unit mass passes through isentropically.

2.5.7 Isentropic Turbine Efficiency

For no loss of heat, velocity, or potential energy in a turbine, equation (2.2.14) shows that the mass and energy rate balance becomes

$$\frac{\dot{\boldsymbol{W}}}{\dot{\boldsymbol{m}}} = \boldsymbol{h_i} - \boldsymbol{h_e} \qquad \text{EQUATION 2-27}$$

For a fixed inlet state, the work per unit mass flowing through the turbine depends only on h_e, and increases as h_e is reduced. The smallest allowed value of h_e will evidently give the maximum possible work output. Because there is no heat loss, equation (2.5.22) shows that this is the state having $\dot{\sigma}_{cv}$ = 0, and $s_i - s_e$., an isentropic process. The only outlet states that can actually be attained are those having $s_e > s_i$.

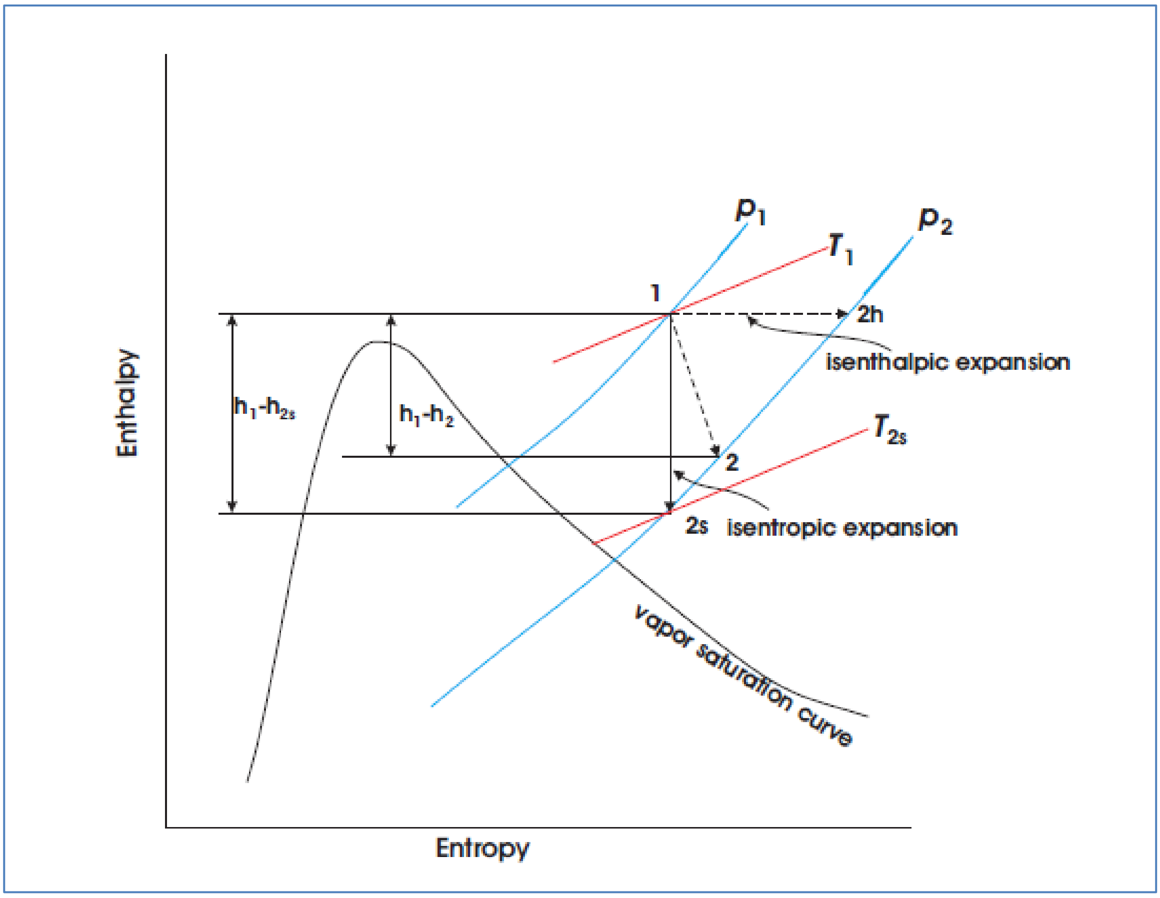


FIGURE 2-6 – SCHEMATIC MOLLIER OR H-S DIAGRAM TO ILLUSTRATE TURBINE EFFICIENCY

The diagram above was taken from (Moran, MJ., and Shapiro, H.N., 2008). Isobars are blue, isotherms are red. The isotherm through state 2 is not shown for clarity. The change from state 1 to state 2h is isenthalpic and irreversible. The change from state 1 to state 2s is isentropic and reversible. The dashed lines 1→ 2 and 1→ 2s represent disequilibrium states which cannot be represented on the diagram.

2.6 Refinery Fuel Gas Theory

Engen refinery utilises fuel gas as combustion fuel for boilers and furnaces. Fuel gas is a mixture of hydrogen and lighter hydrocarbons in the range of methane (CH_3) to butane and its isomers (C_4H_8). The gas mixtures are sourced from various light end distillation and stripping sub-processes. As a result, some hydrogen sulphides and other organic sulphur species are carried with the gas stream but generally in trace quantities. Inert gases such as nitrogen and CO_2 are also present but do not contribute to the net heat value when the fuel gas is combusted. These gases form rather NO_x (Nitrous oxides) and SO_x (Sulphur oxides) which bear an environmental hazard.

The significance of the fuel gas quality (Average Molecular weight) will be further explained later; however, the calculation of the heat value is significant in this section for determination of the heat input into the Rankine cycle.

Fuel gas heat of combustion heating value is characterised as either being lower or higher.

The lower heat value is defined as:

The lower heating value (also known as net calorific value) of a fuel is defined as the amount of heat released by combusting a specified quantity (initially at 25°C) and returning the temperature of the combustion products to 150°C, which assumes the latent heat of vaporization of water in the reaction products is not recovered. (http://hydrogen.pnl.gov/tools/lower-and-higher-heating-values-fuels, n.d.)

The higher heat value is defined as:

The higher heating value (also known gross calorific value or gross energy) of a fuel is defined as the amount of heat released by a specified quantity (initially at 25°C) once it is combusted and the products have returned to a temperature of 25°C, which takes into account the latent heat of vaporization of water in the combustion products. (http://hydrogen.pnl.gov/tools/lower-and-higher-heating-values-fuels, n.d.)

Fuel gas quality is determined by three characteristic properties viz. Average molecular weight, Lower heat value and energy density. Since the combusted fuel gas exits the furnace stack above 150 °C, the physical property of interest is the lower heating value of combustion. The condensing heat available from 150°C to ambient temperature is not utilised and is considered lost heat.

The equation used to determine the average molecular weight of the fuel gas mixture is given as:

$$MW_{mix}^{ave} = \left\{ \frac{\left(\sum_{i=1}^{n} MolFrac_i^{FG}\ x\ MW_i\right)}{\left(\sum_{i=1}^{n} MolFrac_i^{FG}\ \right)} \right\} + \frac{PPM_{H_2S}}{1\,x 10^6}\ x\ MW_{H_2S} \qquad \text{Equation 2-28}$$

Where:

MW_{mix}^{ave}	- Average molecular weight of fuel gas mixture [kg/kmol]
$MolFrac_i^{FG}$	- Mole fraction of species *i* in mixture [vol%/vol%]
MW_i	- Molecular weight of species *i.*
PPM_{H_2S}	- Parts per million of H_2S in the fuel gas [ppmv]
MW_{H_2S}	- Molecular weight of H_2S [kg/kmol]

This equation is applied at ENREF since only H_2S as a contaminant is measured in the fuel gas mixture. No analysers are available for measurement of nitrous compounds. It is accepted however that these elements are apparent in trace quantity (parts per million by volume – ppmv).

The Lower heat value of the fuel gas mixture is given by:

$$LHV_{Corr}^{FG} = \left[\!\left[\sum_{i=1}^{n}\left[Molfrac_i\ x\ MW_i\ \left({}^{kg}/_{kmol}\right)\ x\ \Delta H_i^{comb}\ \left({}^{kJ}/_{kg}\right)\right]\right]\!\right] x\ (1 - \%Inert\ Gases)$$

EQUATION 2-29

Where:

LHV_{Corr}^{FG} - Corrected Lower Heat Value [kJ/kg]

$\%Inert\ Gases$ - Volume percent O_2, CO_2 and/or N_2

Energy density is defined by the equation:

$$Normal\ Energy\ Density\left({}^{kJ}/_{nm^3}\right) = LHV_{corr}^{FG}\left({}^{kJ}/_{kg}\right)\ x\ \rho_{FG}^{stp}\ \left({}^{kg}/_{nm^3}\right)$$

EQUATION 2-30

Where:

ρ_{FG}^{stp} - Standard Normal density of gas mixture [kg/nm^3]

The standard normal density above was derived from the ideal gas law relationship at a pressure of 101.325 kPa and 273.15 °K. Refer Fuel Gas Calculations, page 85.

2.7 Industry Standards – CHP Performance Metrics

A set of industry accepted terms of reference and performance metrics are necessary to assess the CHP operation. A complete datasheet development is subject to a vendor-reviewed design and model of the system and typically developed at a later stage in the project.

2.7.1 Boiler Efficiency

The fired duty for the boiler is calculated as

$$Q_b^f(kW) = \varepsilon\left({}^{kJ}/_{nm^3}\right) xF\left({}^{nm^3}/_{s}\right)$$ EQUATION 2-31

Where

$Q_b^f = Boiler\ fired\ duty - From\ Fuel\ Gas\ firing\ (kW)$

ε = Energy density, calculated by product of LHV($^{kJ}/_{kg}$) and STD Normal fuel gas density, $^{kg}/_{nm^3}$

$F = Fuel\ gas\ volumetric\ flow\ rate\ \left(^{nm^3}/_s\right)$

The absorbed duty for the boiler is calculated as:

$Q_b^a(kW) = M\left(^{kg}/_s\right) x\ (H_2 - H_1)$ EQUATION 2-32

Where:

$Q_b^a =\ Boiler\ absorbed\ duty\ (kW)$

$M = Mass\ flow\ rate\ of\ steam\ through\ boiler\ (load)$ - $\left(^{kg}/_s\right)$

$(H_2 - H_1) = Enthalpy\ change: State\ 2 - State\ 1$ as per Part 2b – Determining State 2 – Boiler exhaust steam condition, page 89 - ($^{kJ}/_{kg}$).

Finally to calculate boiler efficiency:

Using Direct Method

$\varepsilon = \frac{\varphi_{out}}{\varphi_{in}}$ EQUATION 2-33

Where:

$\varphi_{in} = Absorbed\ Duty -\ (kW)$

$\varphi_{in} = Fired\ Duty -\ (kW)$

2.7.2 Isentropic Efficiency:

In an ideal expansion across a turbine, no entropy is generated and all energy is completely converted from thermal to shaft work. This is not a real world scenario and is merely used as a reference measure against the turbine efficiency or isentropic efficiency.

The efficiency of the turbine as a performance metric is thereby calculated as a percentage of the isentropic enthalpy change.

The following equations apply – Refer State 3 to State 4 – page 42:

$H_{state\ 3-state\ 4}$ = Actual Enthaply Change from Process $\left(\frac{kJ}{kg}\right)$

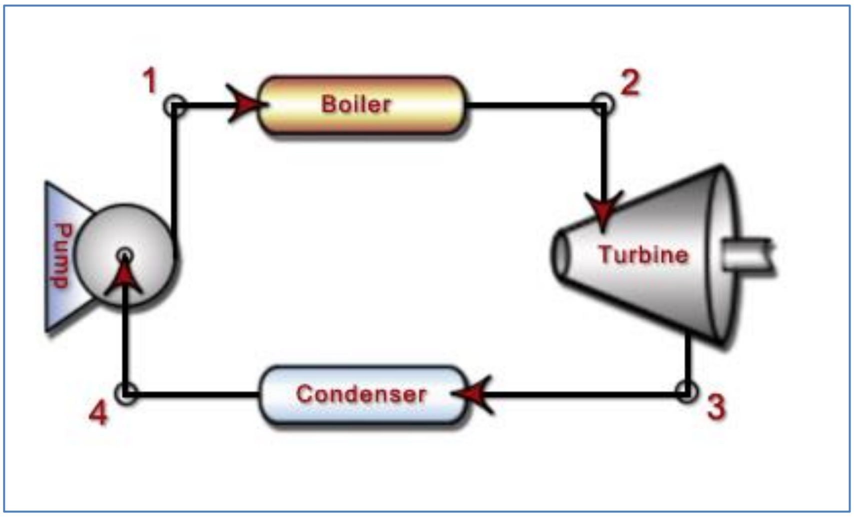
$H_{state\ 3-state\ 4s}$ = Thermodynamic maximum possible change of Isentropic $\left(\frac{kJ}{kg}\right)$

$$\epsilon_{isen} = \frac{H_{state\ 3-state\ 4}}{H_{state\ 3-state\ 4s}} = \textbf{Isentropic efficiency}\ (\%) \qquad \text{EQUATION 2-34}$$

2.7.3 Back work ratio

The BWR or Back Work ratio is an industry accepted metric to determine the energy ratio between work supplied to the system and work extracted.

A simplified sketch is shown below:



SCREENCAP 2-2 - BASIC RANKINE CYCLE REFERENCING BWR CALCULATION

(http://www.learnthermo.com/T1-tutorial/ch09/lesson-B/pg07.php, n.d.)

The accompanying equation for the BWR is given by:

$$\boldsymbol{BWR = \frac{-W_{s,P}}{W_{s,T}} = \frac{H_1 - H_4}{H_2 - H_3}}$$ EQUATION 2-35

2.8 Electricity Charges

Local electricity at Ethekwini municipality is charged to industry consumers at a predefined rate based on consumption and peak demand. These are further elaborated in the economics section of this thesis. However, for theoretical savings calculations, the following method is applied.

By definition:

$$\boldsymbol{Electricity\ Billable\ saving = Grid\ Load\ by\ Generator\ (MW) * Electricity\ Cost\ (\frac{R}{MWh})}$$

EQUATION 2-36 - REDUCTION IN ELECTRICAL COST

3 Research Technique

3.1 Plant Performance Triangle approach

The technique applied for the data collection, analysis and interpretation in this thesis is derived from Perry's Chemical Engineers Handbook (Robert H Perry, 1997).

According to (Robert H Perry, 1997), the motivation for analysis of plant performance is four-fold:

- Identify problems in the current operation
- Identify deteriorating performance in instruments, energy usage, equipment or catalysts.
- Identify better operating regions leading to improved product or operating efficiency
- Identify a better model leading to better design.

In this thesis, the elements of the plant performance triangle used are better operating regions and a better design. The opportunity to retrofit an existing turbo-compressor to a turbo generator requires careful plant analysis of operating regimes, equipment limitations and optimal process operation. A diagrammatic representation of the plant performance triangle is shown below (Figure 3-1 – Adapted - Plant Performance Triangle). A plant performance analyst is responsible for historical data collection, reconciliation, rectification and interpretation. For this study, plant data collection encompassed one of the many legs of the research. In addition, face-to-face and telephonic discussions were conducted with plant and laboratory personnel. Panel operator's insights into plant operation are critical for a plant performance analyst's view of the challenges and constraints associated with unit operation.

When performing calculations around laboratory measurements, inherent random error is always present. ENREF fuel gas composition is measured daily by Gas chromatography analysis. Fuel gas samples are sometimes taken during process upsets, introducing random error. However, due to the number of available measurements for this study, data cleaning techniques eradicate the occasional sampling error bias. Bad values, known process downtime for shutdowns, unexpected plant upsets and missing results have been omitted from the data sets in its entirety. By this method of data conditioning, all bad process operating data in synchronisation with laboratory data timestamps are excluded. This technique allows only good cleaned data to be presented for analysis. Additionally, data outside of the defined process boundaries are also excluded and discarded to ensure comparable results pre and post mid-2014.

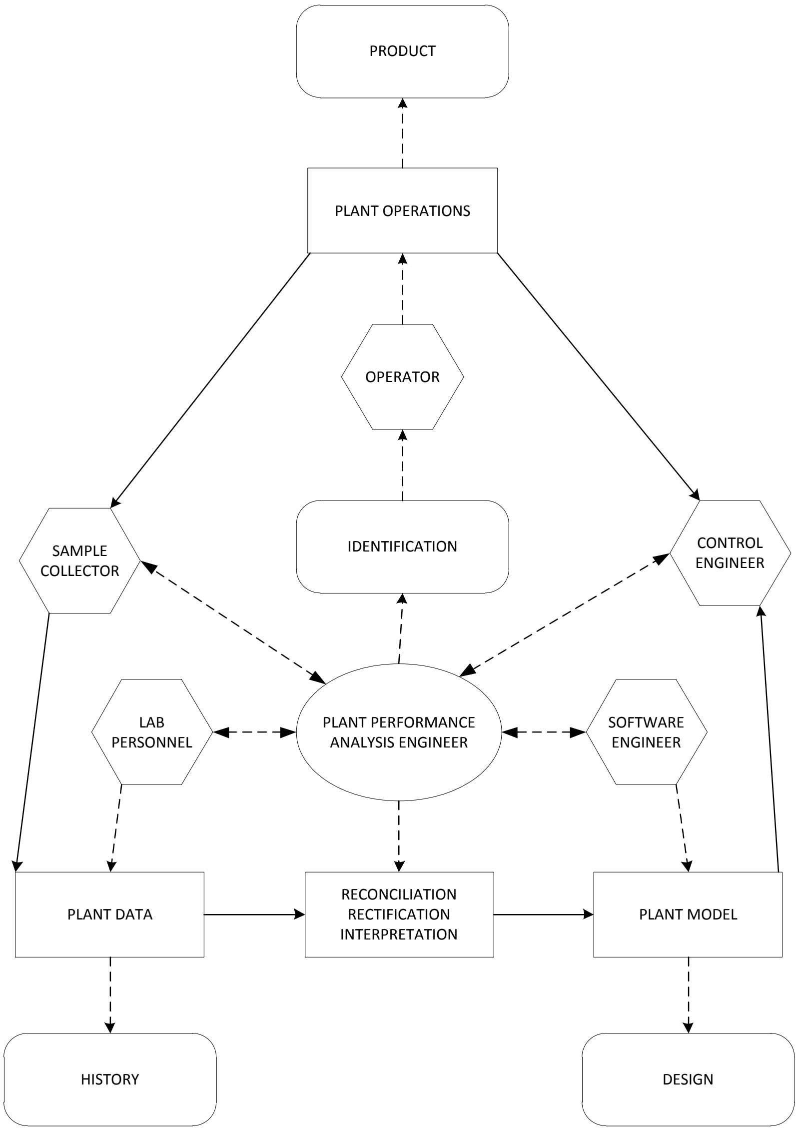


FIGURE 3-1 – ADAPTED - PLANT PERFORMANCE TRIANGLE

(Robert H Perry, 1997), pp- Section 30-9, Fig 30-4

In applying the above depicted research technique, it is noted that organisational structures have changed since this publishing by Robert H Perry. Plant performance analysis engineer can be interchanged with Process Technologist. Control engineer may replace software engineers in context of control engineering functions, as these roles have changed in modern organisational hierarchy. Software engineers are better recognised in roles such as Information Technology in large corporates today. Lab personnel and sample collectors are generally used for "Test-Run" processes. Historical lab data is often available on most process plant historians. The methodology is, however, well constructed for plant performance analysis.

3.2 Data Collection and Filtering

Plant data for this study was drawn from a local data historian integrated within ENGEN's process data management systems. The fastest data collection scan rate into the historian is at 1 minute (60 second) intervals for thousands of pre-defined process tags. The historian is capable of averaging data within its software over hours, days, weeks, months, years or a specified time interval. For this study, data was averaged over 24 hours, defined as one operating day. The timestamp for data collection was set to midnight of a specified day 1 (dd/mm/yyyy 00:00:01) up to midnight of day xx within the history set. The historian then automatically segregates data into day averages between the defined date ranges and presents all data tags in columns at each dated row. The total data collection for this study spans from 01 Jan 2010 00:00:00 to 25 May 2016 00:00:00, 2336 operating days. Subsets of process data were collected at day averages for multiple process tags across all operating boilers, steam, fuel gas, feed rates etc. Useful process data was extracted from the data sets based on the research technique explained in the plant performance triangle approach.

The final filtered data set was segregated into two parts viz. pre mid-2014 during which both SAFOR turbines were in continuous service and post mid-2014 after partial decommissioning. The following data boundaries were applied for filtering:

TABLE 3-1 – PROCESS DATA FILTER BOUNDARIES – PRE MID 2014 SET

Parameter	Min	Max	Engineering Unit
Crude feed rate	15500	16000	M^3/day
SAFOR boiler 1 HP steam demand	50	62	Tons/hr
SAFOR boiler 2 HP steam demand	42	62	Tons/hr
SAFOR MP Steam Export	1	42.6	Tons/hr

Since the refinery's steam demand is balanced between five boilers, it is imperative for accurate results that turbine design calculations are based on operational healthy SAFOR loads pre mid-2014 vs. fully offline post mid-2014. The MP export steam load from SAFOR to the refinery indicates the capability of the export piping systems hydraulic capacity during steam turbine operation. A key limitation to operating the turbine at full capacity post mid-2014 is the hydraulic capacity of the MP export valve.

3.3 Definition of Study Cases

Historical and current data sets for this thesis are divided into two parts viz. Case 1 and Case 2. The details of the cases are tabulated below.

TABLE 3-2– CASE 1 VS CASE 2

	Case 1	Case 2	Engineering unit
Start Date	18 January 2012	11 April 2015	
End Date	27 August 2013	30 October 2016	
Number of Data points	116	211	
Crude Rate Min	15505	15510	M^3/D
Crude Rate Max	15999	15715	M^3/D
Crude Rate Average	15669	15657	M^3/D
Crude Rate STD deviation	133	46	M^3/D
Turbine C4002 Steam Ave	50	0	Tons/hr
Refinery HP Mains Pressure	37.36	38.72	bar.g

The data sets for case 1 and case 2 were captured, filtered and cleaned based on the filters applied in cases are tabulated below.

Table 3-2– Case 1 vs Case 2 above. These filters ensured that accurate data was collected for both operational periods of the study. Period 1 (Case 1) summarised data for when the C4002 HP steam turbine operated at its normal load in relation to the crude rate ranges. Period 2 (Case 2) data sets summarised operation of the SAFOR plant with zero HP steam load to the turbine.

4 Calculations and Discussions

4.1 Comparative Operating Process Data

One of the objectives of this thesis is to analyse operating process data pre and post the mid 2014 SAFOR plant partial decommissioning. The refinery steam and fuel gas balance is strongly dependant on the operating demand and supply. The five HP boilers on site are capable of supplying steam in excess of the total refinery demand at a broad range of operating crude charge rates. The data sets selected for the study are therefore based on comparable average crude rates for case 1 and case 2.

4.1.1 HP Steam Boilers Operation

ENGEN operates five HP steam boilers on site feeding into a common HP steam header. The rated operating data are tabulated below.

TABLE 4-1– BOILERS MAXIMUM CONTINUOUS RATING (MCR)

North Boiler 1	45	t/hr
North Boiler 2	45	t/hr
North Boiler 3	75	t/hr
SAFOR 1	64	t/hr
SAFOR 2	64	t/hr

The maximum continuous rating of a boiler is the highest continuous feed rate that the boiler can sustain while maintaining its required set pressure of 40 barg. The largest on site boiler at ENREF is North Complex boiler 3, with a capacity (MCR) of 75 tons/hr. Both SAFOR boilers have a MCR of 64 tons/hr however; historical data reveals that the boiler has not operated above 56 tons/hr since 2010 (As per the selected data set for this study).

TABLE 4-2– STEAM AND BOILER DATA OVERVIEW

SUMMARY	CASE 1 SAFOR TURBINES ONLINE	CASE 2 SAFOR TURBINES OFFLINE	DIFFERENCE (CASE2 - CASE 1)	ENG UNIT
CRUDE RATE	15669.10	15656.96	-12.14	M^3/D
DATA POINTS	116	211	95.00	[-]
STEAM LOAD/DEMAND				
NORTH BOILER 1 LOAD	40.25	18.25	-22.00	tons/hr
NORTH BOILER 2 LOAD	40.76	25.30	-15.46	tons/hr
NORTH BOILER 3 LOAD	1.27	47.47	46.20	tons/hr
SAFOR BOILER1 LOAD	56.14	33.71	-22.43	tons/hr
SAFOR BOILER2 LOAD	55.19	33.65	-21.54	tons/hr
TOTAL	193.62	158.39	-35.23	tons/hr
FUEL GAS CONSUMED				
NORTH BOILER 1 FUEL GAS	2757.21	1624.06	-1133.15	NM^3/hr
NORTH BOILER 2 FUEL GAS	2848.47	2424.07	-424.40	NM^3/hr
NORTH BOILER 3 FUEL GAS	51.87	2040.80	1988.93	NM^3/hr
SAFOR BOILER 1 FUEL GAS	3253.23	3206.04	-47.20	NM^3/hr
SAFOR BOILER2 FUEL GAS	3474.06	3423.66	-50.40	NM^3/hr
TOTAL	12384.84	12718.62	333.79	NM^3/hr
AIR FLOW DRAWN				
NORTH BOILER 1 AIR	45118.11	31223.99	-13894.12	NM^3/hr
NORTH BOILER 2 AIR	52720.39	44996.17	-7724.22	NM^3/hr
NORTH BOILER 3 AIR	442.44	85233.27	84790.83	NM^3/hr
SAFOR BOILER 1 AIR	64274.51	57980.10	-6294.40	NM^3/hr
SAFOR BOILER 2 AIR	83022.60	48617.17	-34405.43	NM^3/hr
TOTAL	245578.04	268050.70	22472.66	NM^3/hr
C4002 TURBINE				
STEAM TO TURBINE	49.65	0.00	-49.65	tons/hr

The following deductions are drawn from the table above:

- Crude rates are comparable between data sets with a minimal average difference of 12.14 m^3/day implying good levels of data integrity between data sets.
- The total refinery steam demand in case 2 is 35.23 tons/hr less than case 1. The reduction in demand is mostly due to the decommissioning of the two SAFOR HP steam turbines.
- The fuel gas consumption between the two cases indicates that more fuel gas was consumed by volume for case 2 relative to case 1. The increased volumetric flow of fuel gas does not translate to increased steam production, since the heat value is the actual metric to assess fuel gas quality. A more accurate measure of quality is the fuel gas energy density, which is a function of the average molecular weight for

the fuel gas mixture as well as fuel gas density. These calculation results are further explained when analysing fuel gas comparisons.

- Numbers displayed in red text for boiler 3 data (Case 1) indicate that the boiler was mostly offline for case 1. The refinery steam demand of 193.62 tons/hr was therefore supplied by two north boilers and two SAFOR boilers for the periods identified in this study.
- Turbine C4002 consumed 49.65 tons/hr HP steam for case 1. This load is considered normal operating for the running turbine as a turbo-compressor.

4.1.2 Fuel Gas Hydrocarbon Constituency

Fuel gas is used as combustible hydrocarbon energy source for heat input into furnaces and boilers at ENREF. Flow measurement of the gas stream is achieved via volumetric flow transmitter instruments. A careful analysis of the actual heat value and energy density of the fuel gas is required to assess the quality of combustion and heat release. The calculation method deployed for the determination of LHV is outlined in Equation 2-29 . Sample calculations can be referenced on page 82, Appendices. Tabulated below are typical fuel gas mixtures observed by daily lab GC (Gas Chromatography) analyses.

TABLE 4-3 – FUEL GAS HYDROCARBON CONSTITUENCY

HYDROGEN	47 - 55	MOL %
NITROGEN	3 - 4	MOL %
OXYGEN	0 - 1	MOL %
CARBON DIOXIDE	0 - 1	MOL %
CARBON MONOXIDE	0 - 1	MOL %
METHANE	18 - 20	MOL %
ETHANE	11 - 13	MOL %
PROPANE	6 - 8	MOL %
N-BUTANE	3 - 4	MOL %
ISO-BUTANE	3 - 4	MOL %
N-PENTANE	0 - 2	MOL %

I-PENTANE	1 - 3	MOL %
CIS 2-C4	0 - 1	MOL %
TRANS 2-C4	0 - 1	MOL %
ETHENE	3 - 4	MOL %
PROPENE	2 - 3	MOL %
ISO-C4 + ISO-BUTENE	0 - 1	MOL %
H2S	8 - 50	PPMV

ENREF fuel gas composition is predominantly made up of hydrogen gas followed by methane, ethane and propane. These four components often make up 85% of the total gas mix. An analysis of the average fuel gas compositions for case 1 and case 2 are presented below.

TABLE 4-3 – NORMALISED FUEL GAS COMPOSITIONS

HYDROCARBON MIX	LAB DATA Case 1 Vol %	LAB DATA Case Vol 2 %	Norm Case 1	Norm Case 2	Norm Difference
HYDROGEN	35.54	41.90	34.85%	42.78%	7.92%
NITROGEN	4.37	4.95	4.28%	5.05%	0.77%
OXYGEN	0.54	0.64	0.53%	0.66%	0.13%
CARBON DIOXIDE	0.22	0.21	0.21%	0.22%	0.01%
CARBON MONOXIDE	0.44	0.39	0.43%	0.40%	-0.03%
METHANE	17.86	18.14	17.51%	18.52%	1.00%
ETHANE	11.92	11.03	11.69%	11.26%	-0.43%
PROPANE	9.91	6.84	9.72%	6.98%	-2.74%
N-BUTANE	5.53	3.99	5.42%	4.07%	-1.35%
ISO-BUTANE	4.41	3.08	4.33%	3.15%	-1.18%
N-PENTANE	0.29	0.31	0.28%	0.31%	0.03%
I-PENTANE	0.58	0.63	0.57%	0.65%	0.08%
CIS 2-C4	0.43	0.00	0.42%	0.00%	-0.42%
TRANS 2-C4	0.62	0.00	0.61%	0.00%	-0.61%
ETHENE	3.48	3.15	3.42%	3.22%	-0.20%
PROPENE	2.77	2.68	2.72%	2.74%	0.02%
ISO-C4 + ISO-BUTENE	3.07	0.00	3.01%	0.00%	-3.01%
H2S	7.44	36.28	7.29%	37.04%	29.75%
SUM	101.98	97.94	1.00	1.00	

The following deductions are drawn from the table above:

- Case 2 data reveals that hydrogen content in fuel gas after the SAFOR decommissioning of the turbines increased by 7.92%. This cannot be attributed to the actual decommissioning of the turbines, since the fuel gas mixture and turbine operation are independent. It is therefore assumed that other process reasons such as reformer operation (hydrogen producing) or changes in crude diet contained increased hydrogen molecules.
- Smaller differences are observed in methane and ethane content between case 1 and case 2.
- Propane content in fuel gas reduced by 2.74% in case 2 which would have some influence in reducing the net heat value.

4.1.3 Fuel Gas Quality Comparisons

Combustible hydrocarbon gas mixtures are commonly used in refineries as a heat source. Natural gas (Methane rich gas) is often accounted in the firing cost for boilers and furnaces as an operating expense. ENREF does not import or purchase such gas for combustion, but rather utilises off gasses from sub processes as combustion fuel gas. The measures of fuel gas quality are average molecular weight, Lower heat value and normal energy density. Calculation methods for these can be referenced on pp 25. Tabulated below are comparative fuel gas results for case 1 and case 2. Calculations of fuel gas and air rates are based on all five operational boilers.

TABLE 4-4 – FUEL GAS QUALITY CASES COMPARISON – REFINERY WIDE

	Case 1	Case 2	% Difference	EU
Ave Mol weight calc	24.01	18.69	-22.19%	kg/kmol
Ave Mol weight analyser	24.51	21.15	-13.69%	kg/kmol
Std Norm Density	1.07	0.83	-22.19%	kg/NM^3
Corrected LHV	45750	45812	0.13%	kJ/kg
Energy Density	49013	38191	-22.08%	kJ/NM^3
Fuel gas consumed	12384.84	12718.62	2.70%	NM^3/hr
Air Consumed	245578.04	268050.70	9.15%	NM^3/hr
TOTAL Fired Duty	168.62	134.93	-19.98%	MW
Fired Efficiency	0.87	0.85	-2.18%	MWh/ton

The following deductions are drawn from the results above:

- ENREF fuel gas flow meters are calibrated to measure Normal flow rates calculated at 101.325 kPa and 273.15 °K. Normal data permit case 1 and case 2 fuel gas quality analyses to be comparable.
- The average molecular weight by calculation method for case 1 is 22.19% lower than for case 2. This is explained by the 7.92 % increase in hydrogen concentration in fuel gas as observed in Table 4-3 – Normalised fuel gas compositions. Hydrogen has the lowest molecular weight by comparison to all other components in the fuel gas mixture.
- ENREF is equipped with an online analyser which measures molecular weight by ultrasonic wave. It is observed that the analyser becomes more inaccurate as actual molecular weight decreases. As observed, increased hydrogen in the fuel gas mixture skews the analyser result by virtue of the measurement principle used (density). The actual mol weight for case 2 being 18.69 kg/kmol vs 21.15kg/kmol given by the analyser. Manually calculated methods for fuel gas average mol weight are considered to have increased accuracy over the installed mol weight analyser, since the calculation uses actual daily lab results.
- Standard normal density is a simple function of molecular weight and the ideal gas law relationships (see page 82 – Fuel gas sample calculations in appendices). The normal density of the fuel gas has reduced by 22.19% for case 2 implying that fuel gas quality deteriorated after the partial decommissioning of SAFOR. This phenomenon is independent to the decommissioning of the turbines and is attributed to operational changes at the gas reformer unit at ENREF. It is known that the reformer plant was operated with increased severity during this period to increase RON (Research Octane Number) and coincided with the operational changes at SAFOR plant.
- It is observed that the corrected LHV is almost unchanged for case 1 and case 2. Since the heat value of the fuel gas is a function of component heat values, percentage in mix and molecular weight, an increase in heat value is compensated by an increase in mol weight. Refer to appendices page 82 – sample calculations.
- Air requirements have increased for case 2 despite a reduced total HP steam load as indicated in Table 4-2– Steam and Boiler Data Overview.

- This is attributed to increased hydrogen in fuel gas since the stoichiometric Air: Fuel combustion requirement for hydrogen is the largest for all components in fuel gas.
- The net total fired duty represents the actual energy consumed between case 1 and case 2. A reduction of 19.98% energy usage is observed in case 2 for a reduced steam load of 35.23 tons/hr refer Table 4-2, page 34. This translates into a percentage reduction of 18.19% HP steam load.
- Fired efficiency is calculated by dividing total fired duty by the total steam load. A minute difference of 2.18% reduction in fired duty is observed in case two, indicating a lesser efficiency. This is likely attributed to reduced efficiency due to excess convection cooling by higher air flows in the boiler and reduced boiler loads in case 2.

4.2 CHP Design Calculations

A combined heat and power cycle is proposed as an improvement to the existing process configuration at SAFOR plant. It is known that a combined cycle is far more energy efficient than a single cycle since the steam pressure reduction from HP to MP produces useful shaft work isentropically as opposed to isenthalpic pressure reduction. In the calculations to follow, comparisons are tabulated between these options and deductions drawn based on overall energy efficiency, de-superheating demand and net work produced.

4.2.1 CHP design basis

A design basis is applied to this thesis for the retrofit of the existing turbo compressor to turbo generator modification. The following summary of design conditions are used to formulate the design basis for turbine C4002:

- Refinery Crude rate - 15500 to 16000 m^3/day
- Refinery Crude properties
  - API - 34 – 36 ° API
  - Overheads distillation - IBP to 172 °C
  - Reformer N + 2A - 36 – 38 LV%
- HP steam pressure to turbine - 38.24 Bar.g

- HP Steam temperature - 400 °C
- Design steam rate to turbine - 42 tons/hr
- Fuel gas molecular weight - 21.15 kg/kmol
- Fuel gas STD normal density - 0.83 kg/nm^3
- MP steam backpressure - 11.1 Bar.g

4.2.2 Notes and Assumptions:

- It is assumed that the SAFOR turbo-generator will be operating at the current crude rates charged to the refinery i.e. 15500 – 16000 m^3/day. This rate is based on the current refinery steam balance and fuel gas demand.
- Crude API (American Petroleum Institute – crude density equivalent) is a measure of the recovery expected at different cut points in the crude tower and affects the reformer feed rate. The reformer unit is a hydrogen producer at the refinery and influences the fuel gas quality.
- Overheads distillation range determines the volume of feed reported to the reformer reactors. Increased feed results in increased hydrogen to the fuel gas pool; however, other factors such as N+2A may counter this effect.
- N + 2A (Naphthalene + 2 Aromatics) is an indicator of the reformer feed quality which influences the fuel gas pool via the hydrogen make. The degree of reforming is dependent on this property.
- HP Steam pressure for the design basis is based on average operating header pressures from the data sets for case 1 and case 2.
- Design steam rate to the turbine is based on vendor's calculations for the reconfiguration. Elliott Group (Ebara Corporation) has been requested to perform re-rate calculations for the turbine reconfiguration. The results reported indicated that the available power from the turbine can be used in a generator set at a steam rate of 42 tons/hr. In addition to this, a limitation exists on the hydraulic capacity of the MP export line of 45 tons/hr. the design basis of 42 tons/hr is therefore appropriate to this design.
- Fuel gas properties are selected based on current operation as defined by Case 2.
- The MP steam back-pressure is assumed to be as per original design as populated on the original turbine datasheet. The actual MP steam header is set at 10.0 barg; therefore, the slightly higher backpressure (1.1 barg) will allow for MP steam flow

into the main MP header given small pressure drops across the piping and de-superheater.

- A process schematic of the key operating conditions are presented below. The schematic is a representation of the Rankine cycle applicable to the retrofit design as the conditions described in the design basis above. Calculations of the thermodynamic properties, efficiencies and comparisons are based on these operating conditions.
- PRV refers to Pressure Reducing Valve, which is applicable to isenthalpic steam let-down from high pressure to low pressure.
- Recovered condensate return at the SAFOR steam/condensate circuit is assumed to be 60% of the total condensate header. Make-up boiler feed water of 40% is required to maintain the steam demand at any time. This assumption is based on historical plant operating data.

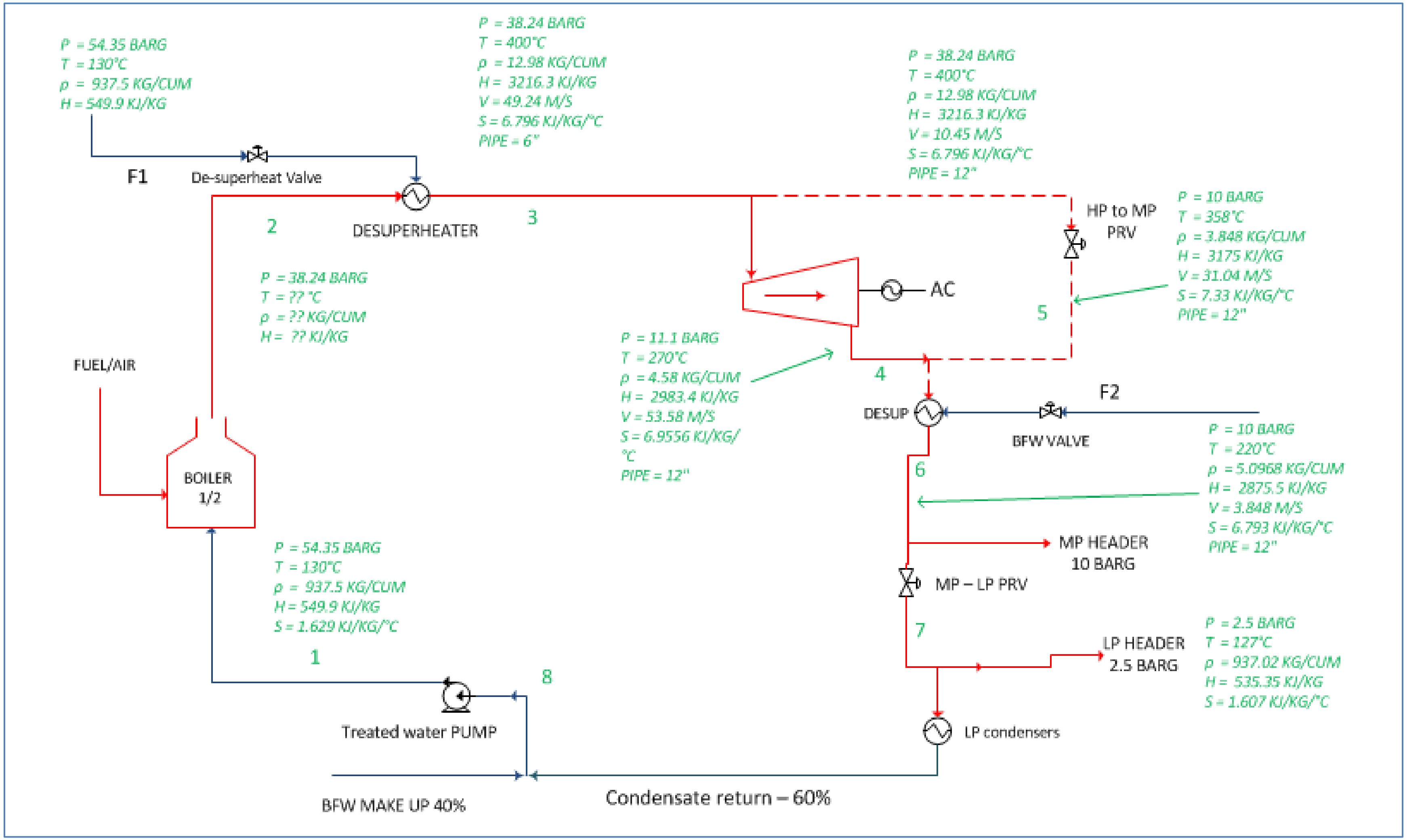
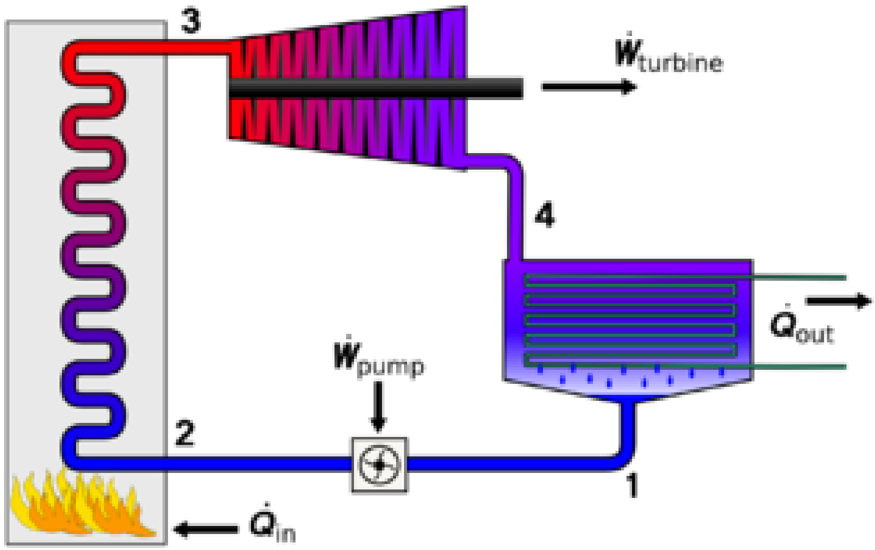


Figure 4-1- Process Design Basis Schematic – Rankine Cycle Overview

4.2.3 State Descriptions – Real Rankine Cycle

A simplified pictorial representation of a Rankine cycle is pasted below:



Physical layout of the four main devices used in the Rankine cycle. *1. Pump 2. Boiler 3. Turbine 4. Condenser*

SCREENCAP 4-1- PICTORIAL RANKINE CYCLE

https://en.wikipedia.org/wiki/Rankine_cycle

Screencap 4-1- Pictorial Rankine Cycle, above, outlines the key intensive (bulk properties e.g. concentration, colour) and extensive (additive properties e.g. enthalpy, mass) property states at ENREF's SAFOR plant encompassing the operation of the boilers, de-superheaters, turbine, PRV, headers, condensate return and BFW pump.

State 1 - Boiler feed water is pumped from the de-aerator to the boiler.

State 2 - HP, high temperature steam exits the boiler and enters a de-superheater

State 3 - De-superheated HP steam is available for consumers

State 4 - Exit steam from turbine: Path 3 to 4 indicates the isentropic path for steam via the turbine

State 5	- Exit steam from PRV: Path 4 – 5 indicates the isenthalpic let-down of HP to MP steam

State 6	- Condensate returned to the boiler feed water pump via MP and LP let-downs

4.2.4 Applying simultaneous Heat and Material balances

With reference to State 2 in Figure 4-1, page 42, there exists an unmeasured and therefore unknown temperature and enthalpy exiting the boiler and entering the de-superheater. It is understood that the firing of fuel gas raises the water state from saturated liquid to superheated steam. The actual amount of heat transferred to the water to achieve this is calculated in the fired duty of the boiler based on the fuel gas fired and the quality of the gas combustion. These are explained in Table 4-4 – Fuel gas quality cases comparison. The determination of the unknowns in State 2 can be calculated by simultaneous mass and energy balance around the desuperheater. BFW is consumed (F1) at the desuperheater, adding to the HP superheated steam total volume produced at the boiler. The calculation result gives the actual volumetric fuel gas required for the state change of water to steam across the firebox. This is possible by the use of real process data measurements within the balanced equations. The consumed fuel gas is thereafter used within the calculations to predict future gas consumption volumes with either the turbine in operation or the PRV. The variability in fuel gas quality is noted within the refinery operation, largely affected by the hydrogen content. Refer sample calculation Part 2b – Determining State 2 – Boiler exhaust steam condition , page 89)

4.2.5 Defining Case 3 and Case 4

In earlier chapters, Case 1 and Case 2 data sets were defined. These data sets described the historical operating conditions and the current conditions. For case 3, data is drawn from case 2 (current) dataset to predict, calculate and interpret the combined heat and power system with the functional backpressure turbine (C4002) in the process loop. Refer Table 4-5 – Known process parameters- Cases 1 - 4, below.

Case 3 describes the path from thermodynamic state 3 to state 4 (turbine shaft work option see page 14). For case 4, the current operating data is again used to describe the operation from state 3 to state 5 (Pressure reducing valve option). The key operating parameters for these states are displayed in Figure 4-1- Process Design Basis Schematic – Rankine Cycle Overview. Calculations will be limited to the process interactions around SAFOR boilers, the turbine and the PRV mostly.

TABLE 4-5 – KNOWN PROCESS PARAMETERS- CASES 1 - 4

	Case 1 - Historical	Case 2 - Current	Case 3 - CHP	Case 4 - PRV	EU
Ave Mol weight calc	24.01	18.69	18.69	18.69	kg/kmol
Std Norm Density	1.07	0.83	0.83	0.83	kg/NM^3
Corrected LHV	45750	45812	45812	45812	kJ/kg
Energy Density	49013	38191	38191	38191	kJ/NM^3
HP Header Pressure	37.36	38.72	38.24	38.24	Barg
HP Steam Enthalpy	3219.07	3216.71	3217.54	3217.54	kJ/kg
C4002 Turbine HP demand	49.65	0.00	42.00	0.00	tons/hr
SAF Boiler 1 % MCR	87.72%	52.68%	85.49%	85.49%	%
SAF Boiler 2 % MCR	86.24%	52.58%	85.39%	85.39%	%
SAF Boiler 1 HP Demand	56.14	33.71	54.71	54.71	tons/hr
SAF Boiler 2 HP Demand	55.19	33.65	54.65	54.65	tons/hr
SAF Boiler 1 F1 Flow	2.81	1.69	2.74	2.74	tons/hr
SAF Boiler 2 F1 Flow	2.76	1.68	2.73	2.73	tons/hr

Tabulated above are the known averaged process conditions for cases 1 to 4. Cases 1 and 2 are included for reference to the changes in operating point’s pre and post mid 2014 in comparison to the design basis in cases 3 and 4. The design turbine HP steam demand of 42 tons/hr is split equally at 21 tons/hr between each boiler resulting in a design basis operating point of 54.71tons/hr per boiler for cases 3 and 4. It is evident that the design basis percentage MCR (Maximum continuous rating) very closely resembles operation in case 1 (historical), ~87%. Vs 85.49%.

4.2.6 Rankine Cycle with Superheat Calculations

The construction of the complete Rankine cycle for this thesis is detailed in this section. References are drawn from theory sections 2.4 , Thermodynamic Heat Cycle and 2.5 Steam Turbine Model. Case 3 will define the retrofit CHP cycle while case 4 will indicate the failover scenario for HP steam bypass via PRV. This scenario is used to determine losses in efficiency for periods when the turbine may not be operating due to plant trips, unplanned downtime or any unit process upsets.

A constructed and marked-up temperature – entropy diagram is shown in Figure 4-1, page 42, for case 3 and 4.

The following assumptions are applicable to the design and performance calculations:

- Each component in the cycle is analysed as an open system operating at steady state
- All of the processes except the turbine are internally reversible
- The turbine is adiabatic and the pump is isentropic
- Condensate leaves the LP condensers as saturated liquid
- No shaft work in boiler or condenser
- Changes in kinetic energy are considered
- Changes in potential energy are ignored

By application of the Differential energy rate form (Equation 2-16, page 18)

$$\frac{dE_{cv}}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \dot{m}_i\left(h_i + \frac{v_i^2}{2} + gz_i\right) - \dot{m}_e\left(h_e + \frac{v_e^2}{2} + gz_e\right)$$

In addition, for no energy accumulation, the steady state conservation of energy term becomes

$0 = \dot{Q}_{cv} - \dot{W}_{cv} + \dot{m}(h_i - h_e) + (\frac{v_i^2}{2} - \frac{v_e^2}{2})$ Refer page 19

The result therefore requires the calculable parameters below:

$\dot{Q}_{cv}$ = Heat input into control volume (kW)

$\dot{W}_{cv}$ = Work done by system (kW)

$\dot{m}_i$ = Mass flow of Steam (kg/s)

$h_i - h_e$ = Enthalpy Change of motive steam for useful work (kJ/kg)

$\frac{v_i^2}{2}$ = Velocity energy term (m^2/s^2)

4.2.7 Design and Performance Results

The graphical representation below, gives a quick overview of the relationship between the boiler duty and the fired fuel gas. It is observed that fired duty increases linearly with steam demand for both boilers. This implies that fired duty can be linearly interpolated for all operating steam rates up to the boiler MCR.

SAFOR boiler 1 (Figure 4-2) and SAFOR boiler 2 (Figure 4-3) are the proposed boilers to be used for increased HP steam to turbo-generation. The development of this plot is critical for accurate calculations as required in the overall heat and material balances. Accuracy level of the regressed correlations are fit for calculation at greater than 90 for R-squared value.

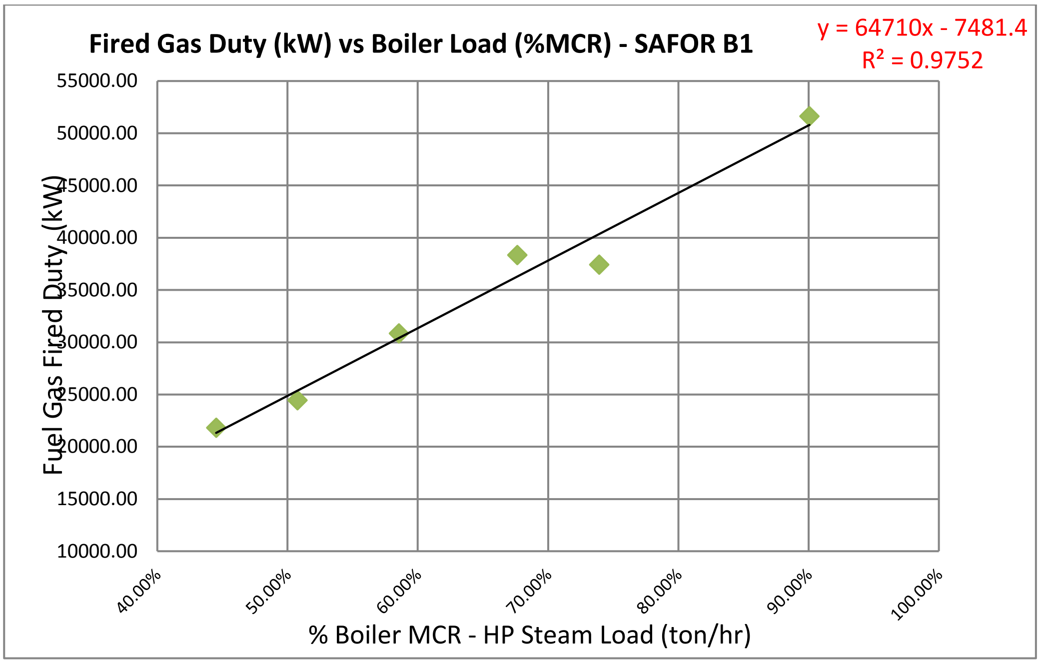


FIGURE 4-2 – SAFOR BOILER 1 FUEL GAS FIRED DUTY VS BOILER LOAD

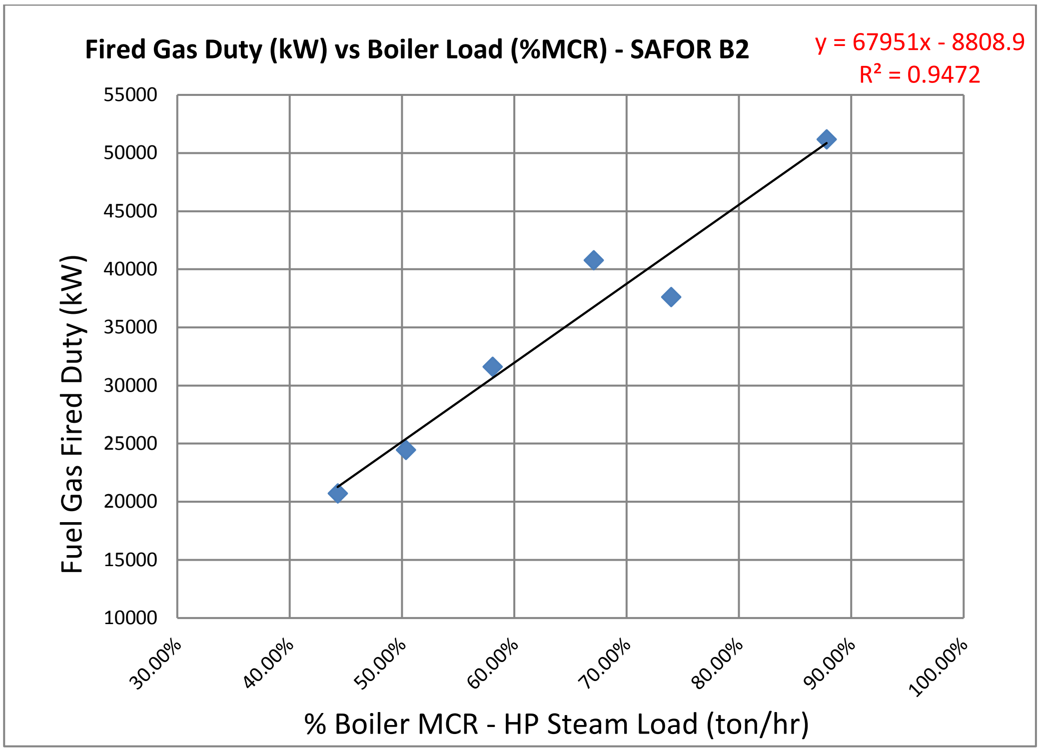


FIGURE 4-3 – SAFOR BOILER 2 FUEL GAS FIRED DUTY VS BOILER LOAD

Since the superheating function thermodynamically raises the boiler exhaust steam state to above the desuperheater controlled outlet temperature for consumers, it is critical to base the new fuel gas consumption on the actual firing required at this elevated temperature state – State 2.

$y = 64710 * X1 - 7481.9$ – SAFOR boiler 1 regressed correlation

$y = 67951 * X2 - 8808.9$ – SAFOR boiler 2 regressed correlation

In addition, by applying the design energy density of the FG, the volumetric rate of fuel gas is calculated:

Calculation results are tabulated below with sample calculations referenced in Fuel Gas Calculations, page 85 and Part 4 – Calculating turbine performance metrics, page 92.

TABLE 4-6 - SUMMARY TABLE OF BACK CALCULATED FUEL GAS CONSUMPTION

Boiler Description	Case 1 - Historical	Case 2 - Current	Case 3 - CHP	Case 4 - PRV	EU
% MCR SAF Boiler 1	87.72%	52.68%	85.49%	85.49%	%
% MCR SAF Boiler 2	86.24%	52.58%	85.39%	85.39%	%
SAF Boiler 1 FG fired	0.930	0.687	1.253	1.253	nm^3/s
SAF Boiler 2 FG fired	0.940	0.669	1.289	1.289	nm^3/s
SAF Boiler 1 Fired Duty	45577.1	26242.4	47839.2	47839.2	kW
SAF Boiler 2 Fired Duty	46063.3	25565.1	49214.5	49214.5	kW

TABLE 4-7 - SUMMARY OF CHP PERFORMANCE METRICS

Description	Performance Metric	Value	Eng Unit
SAFOR boiler 1	Efficiency	91	%
SAFOR boiler 2	Efficiency	92	%
Turbine	Isentropic Efficiency	73	%
Turbine/Pump	BWR	6.25	%

4.2.8 Discussion

- Both the fired fuel gas volumes and duty requirements from Table 4-6 - Summary table of back calculated fuel gas consumption have increased for the CHP and PRV operating scenario. It is evident that despite the MCR of the SAFOR boilers for the retrofit design reduced to 85.49% from around 86%, more firing is required at the boiler. This is attributed to the reduced quality (increased hydrogen content) of fuel gas, which is a critical design parameter for cost effective CHP. It is also noted that other process industry applications of CHP would not normally consider sensitising a zero fuel gas pricing option within their economic model. This scenario may be unique to ENGEN.

- SAFOR's boiler efficiency is very healthy at above 90% for an industrial boiler given the years of operation in service (> 40 operating years). The optimised use of an economiser and steam superheating from flue gas directly improves efficiency, as is the case.
- Isentropic efficiency is healthy for the turbine-generator operation at 73% and comparable to modern industrial machine performance metrics.
- The pump BWR for the design is greater than industry researched Rankine cycle operation See (Martijn Van Den Broek, 2013). This is directly attributed to the turbine operation in backpressure service. A full condensing turbine would result in significantly higher power output for the same steam demand, thereby reducing the BWR.

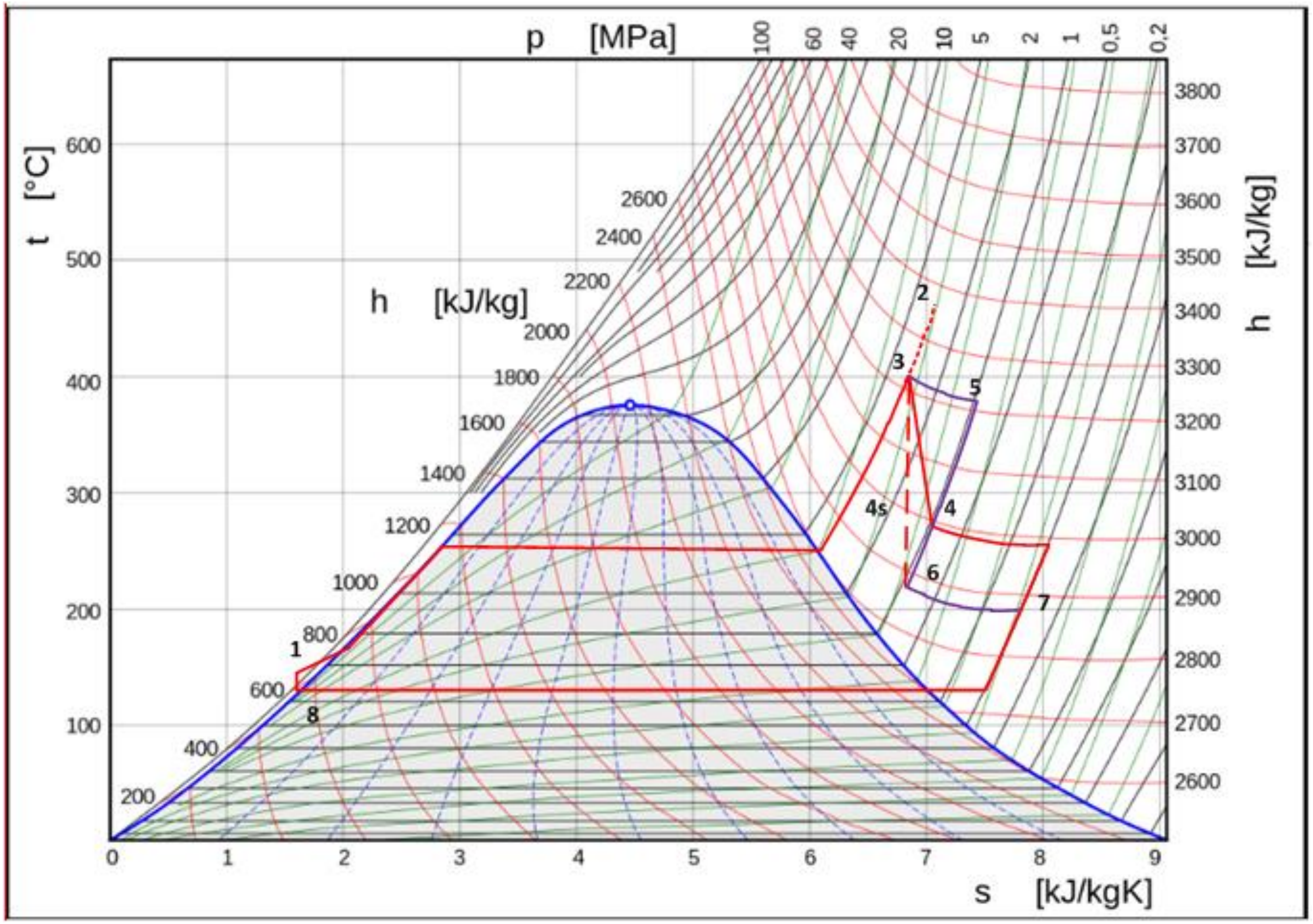


FIGURE 4-4– T-S DIAGRAM OF RANKINE CYCLE

(By Kaboldy - Own work, CC BY-SA 3.0, https://commons.wikimedia.org/w/index.php?curid=20037560)

5 Economics and Sensitivities

This part of the thesis aims to quantify the cost-to-benefit analysis of the retrofit CHP installation at ENGEN Refinery. Expenses related to the plant modifications, new equipment and field costs are considered CAPEX (Capital Expenditure). Other expenses incurred during operation are termed OPEX (Operating Expenses). -

5.1 Economic Data Overview

5.1.1 Data for Economic Calculations

Based on the turbine desired HP steam load of 42 tons/hr, the total HP steam load at SAFOR boilers are compared between cases and analysed as follows: (Refer to Table 4-2– Steam and Boiler Data Overview, page 34)

Historical HP Steam Demand - Case 1	: 111.33 tons/hr
Current HP demand from boilers - Case 2	: 67.36 tons/hr
New Operating Point - Case 3	:109.36 tons/hr
Turbine Bypass Operation – Case 4	:109.36 tons/hr

It can be deduced from the above data that the new operating point with the turbine in CHP service will very closely resemble historical operation. The percent comparison in operating loads is 98.23 %.

It is therefore practical to assume that the fired duty and refinery steam balance for case 1 (historical) will closely resemble the new operating point in case3. The actual export of HP and MP steam between SAFOR and the Refinery however, is to be recalculated. This is a result of the partial mothballing of parts of SAFOR plant, where HP steam would otherwise have been consumed within the SAFOR process. A new HP and MP steam balance is proposed for the new operation.

Key Assumptions

- Refinery fuel gas is assumed an accounting value, which can be used in cost calculations based on consumption. However, fuel gas is not a direct cost to the refinery since this gas is produced as a by-product of crude processing and does not have a saleable value.

- For cost calculations, an average cost of HP steam is used, on the basis that the HP steam supply to the turbines may be sourced from any of the operational boilers at a point in time. The assumption is that minimal turbine downtime is projected such that refinery boiler loads are maintained to accommodate the turbine in continuous service.
- Boiler 3 fuel gas measurement instrumentation proved to be unreliable and was therefore excluded from cost calculations.

TABLE 5-1– SUMMARY OF PRICING INFORMATION

Description	Cost - ZAR	Unit of Cost
Boiler Feed Water	40	kL
*Refinery Fuel Gas	100	Per GJ
**Municipal Electricity	618.70	Per MWh

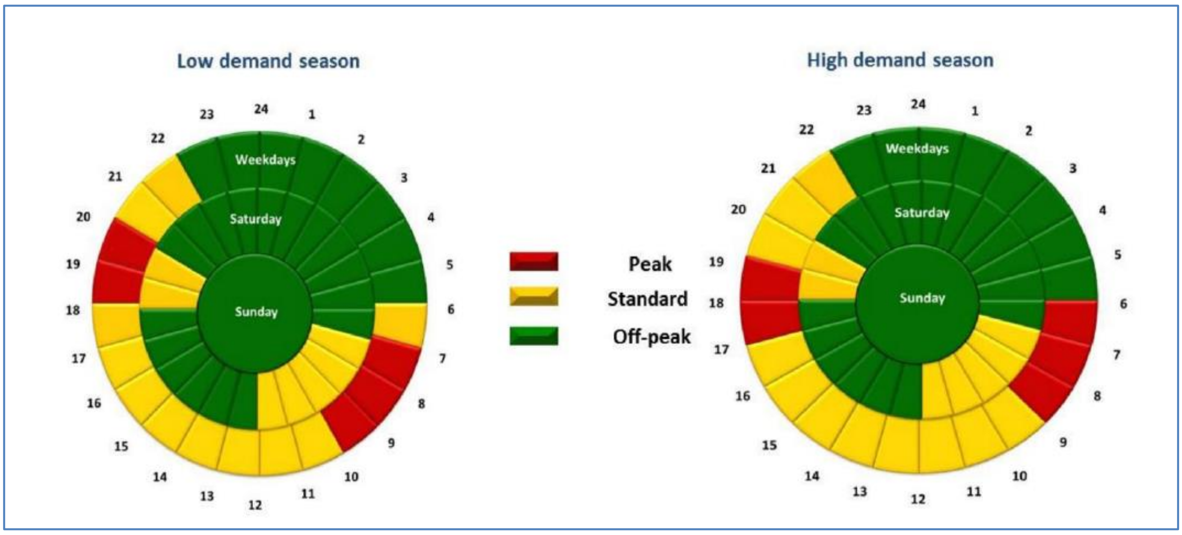
* Pricing taken from (Efficiency, 2015)

** Weighted average – Peak and off peak by Seasonal demand (Efficiency, 2015) – 01 July 2015 to 01 June 2016

Active Energy Charge (R/MWh)					
High Demand Season			Low Demand Season		
Peak	Standard	Off-Peak	Peak	Standard	Off-Peak
2412.3	777.4	452.6	831.8	593.4	400.8

SCREENCAP 5-1 – LOCAL ELECTRICITY SEASONAL PRICING

(Efficiency, 2015) pp: 34-36 - 01 July 2015 to 01 June 2016



SCREENCAP 5-2– MUNICIPAL ELECTRICITY VARIATIONS

01 July 2015 to 01 June 2016 (Efficiency, 2015) pp 24/25

The key differences between the demand season pricing are:

- Hours 6am to 7am are billed at peak prices in high demand season and standard prices in low demand season.
- Hours 6pm to 8pm are billed at peak prices in low demand season and hours 5pm to 7pm are billed as peak in high demand season.

Peak hours are generally associated with higher electricity demand due to increased residential consumption. However, in high demand season, businesses and holiday travel contribute additionally to peak demand.

The actual unit cost of steam per ton is calculated from the pricing information given above:

$$C_s = \frac{C_w}{\rho_w} + \frac{C_e(H_{st} - H_{fw})}{\epsilon} \qquad \textbf{EQUATION 5-1}$$

Where

C_s = Cost of HP Steam (ZAR/ton)

C_w = Cost of Boiler Feed Water (ZAR/m^3)

C_e = Cost of energy as Fuel Gas (ZAR/GJ)

ρ_w = Density of boiler feed water at feed temperature (ton/m3)

H_{st} = Enthalpy of HP steam (GJ/kg)

H_{fw} = Enthalpy of boiler feed water (GJ/kg)

ϵ = Boiler efficiency – Direct thermal method [MWh fired/MWh Absorbed]

Since each boiler operates at different efficiencies, HP steam cost comparisons can be drawn between boilers. It is noted that the enthalpy changes of water to superheated steam is provided by the fuel gas. Additional fuel gas that is combusted due to boiler inefficiency is compensated in the cost function above based the direct efficiency method.

By applying the unit cost of steam formula on page 54, resulted are tabulated below:

TABLE 5-2 - HP STEAM COST COMPARISON

Properties	North Boiler 1	North Boiler 2	North Boiler 3	SAFOR Boiler 1	SAFOR Boiler 2
Enthalpy HP Steam [kJ/kg]	3216.61	3216.28	3216.28	3216.28	3216.28
Enthalpy BFW [kJ/kg]	517.14	520.32	550.3	524.84	524.84
Temperature BFW [°C]	122.49	123.27	130.23	126.32	124.33
Density of BFW [ton/m^3]	0.941	0.941	0.935	0.934	0.94
Boiler efficiency [-]	0.79	0.75	1.47	0.92	0.91
Cost of HP Steam (ZAR/ton)	395.51	416.14	210.46	339.10	342.52

At the current cost of Refinery Fuel gas at R100 per GJ, the average refinery cost of HP steam is R397.14. The average calculation excludes boiler 3 cost due to the inherent error in the efficiency calculation. (See Boiler Fired Duty Vs Load, section 5.3.3 pp 71). It is noted that the cost of HP steam for boiler 3 is significantly lower when compared to other boilers. This was identified as a limitation to the calculations, as the heat balance did not converge. Calculation results showed that the heat out of boiler 3 was a higher value than the heat input by combustion. It is likely that the fuel gas flowmeters are under reading.

See Limitations and Delimitations, page 4.

TABLE 5-3- REFINERY STEAM BALANCE – DESIGN BASIS

	Eng unit	Current	New Operating Point - Case 3	Turbine Bypass - Case 4 corrected
TOTAL HP to Header	t/h	173.21	178.02	173.21
Refinery HP steam demand	t/h	157.19	159.57	157.19
SAFOR Turbines HP demand	t/h	0.00	42.00	0.00
MP steam to MP header	t/h	21.15	42.00	42.00
Net Additional MP steam - turbine exhaust	t/h	0.00	20.85	19.15
NCPX HP steam let-down Cutback	t/h	0.00	-18.46	-37.20
NCPX HP to Header	t/h	96.97	78.50	59.77
SAFOR HP to Header	t/h	60.22	81.06	102.22
SAFOR HP to MP cutback	t/h	0.00	-21.15	0

Tabulated above are comparative plant operating conditions based on filtered and cleaned datasets. The descriptors are elaborated below with brief explanations:

- **Crude charge to Refinery (m^3/day)** – The total crude rate processed at the Refinery front end. This parameter determines the refinery's HP steam demand for various equipment e.g. Steam reboilers, stripping steam, tracing steam or other HP steam turbine drives for pump motors. Averaged datasets filtered at similar crude rates allows for comparison of process and utility operating conditions. Higher crude rates typically imply increased downstream feed rates to finishing units and thereby increased steam consumption.
- **TOTAL HP to Header** – Defines the refinery's net HP requirement as a sum of the total HP produced from each of the 5 onsite boilers as well as the HP steam produced from a large waste heat boiler. ENGEN Refinery is equipped with an

onsite HP steam waste heat boiler located at the Reformer plant. For case 1, 222.79 t/hr HP steam was produced compared to 173.21 t/hr after June 2014 partial mothballing of SAFOR plant. The reduction in HP demand is attributed to the redundant equipment at SAFOR that would have otherwise consumed HP steam. The equipment of interest is the HP steam backpressure turbine – C4002.

- **Total HP – All Boilers** – The sum of all HP steam produced from boilers only. For comparison purposes, only steam adjustments and mass balances from boiler produced HP steam are considered. It is assumed that the Reformer HP steam is proportional to the crude rate and independent of boiler HP demand. Historically, an average of 209.24 tons/hr HP was produced at the refinery compared to the current demand of 157.19 tons/hr. The design basis demands an additional 42 tons/hr HP for C4001 service as a turbo-generator. Case 3 total HP steam requirement is therefore 199.19 tons/hr.
- **SAFOR Turbines HP demand** – Prior to the 2014 partial mothballing, SAFOR plant operated with two HP turbines (C4001 and C4002) demanding a total of 63.46 tons/hr HP steam. The basis of this thesis redefines the use of the larger turbine (Max 50 tons/hr HP) retrofit as a turbo-generator. The current (Case 2) HP steam demand for turbines is 0.00 tons/hr. A demand of 42 tons/hr is therefore required in addition to the current case 2 operating HP load of 157.19 tons/hr. As discussed in earlier chapters, only turbine C4001 is considered for retrofit application. Turbine C4002 will continue to be retained under preservation until fit for re-use.
- **NCPX HP to Header** – The total HP steam produced from North boilers 1, 2 and 3. (MCR 45, 45 & 75 tons/hr). A total of 116.70 tons/hr HP steam is produced from three North boilers.
- **SAFOR HP to Header** – The total HP steam produced from SAFOR boilers let into the main HP header (MCR 64 & 64 tons/hr). SAFOR steam is exported into the refinery main as well as consumed within the SAFOR plant. A total of 92.54 tons/hr HP steam is produced at SAFOR boilers 1 and 2 combined.
- **SAFOR MP Export to Refinery** – Backpressure turbines C4001 and C4002 lets steam down from 40 barg to 11 barg. The exhaust steam is controlled at the refinery's MP steam header main. Therefore, MP steam consumers utilise steam from a combination of turbine exhaust steam and HP to MP pressure reducing

valves. The quality of the exhaust steam from a turbine is lower due to isentropic expansion and work removed from the steam compared to isenthalpic let-down via PRV. Historically (case1) SAFOR exported 42.54 tons/hr to the refinery MP header. Currently, 21.15 tons/hr HP steam is exported. By design (case 3), 42 tons/hr will be exported primarily due to the limitation of the export control valve capacity – refer *Limitations and Delimitations page 4*. As a result, a net reduction of 42 – 21.15 = 20.85 tons/hr MP steam is required from North boilers HP to MP let-down stations.

- With reference to page 93 (Steam Let-down Calculations) the net HP steam cutback required for 20.85 tons MP let-down is 18.46 $ton\ HP$. By this, it is calculated that for every 1 ton of HP steam devalued to MP steam via the PRV, 1.129 tons of MP steam enters the MP header. When compared to the turbine devaluation of HP to MP, no additional MP steam is added (1: 1 ratio of HP to MP) by de-superheating.

It is now required that the net cost of operation at the new operating point be evaluated for cases 3 and 4 considering the devaluation of HP steam to MP steam in each case. For this analysis, it is required that each boiler performance be evaluated in terms of fired duty per ton HP steam produced. The result of which will determine the additional consumption of fuel gas (Case 3) and additional production of MP steam (Case 4) at the new operating point.

5.1.2 Discussion Summary

- Table 5-3- Refinery Steam Balance – Design Basis is a summarised representation of the expected operating steam loads of the refinery header and let-down systems. Based on operating plant data for case 3, NCPX HP steam header contribution is reduced from 96.97 tons/hr to 78.50 tons/hr when the HP turbine is in normal operation.
- It is to be noted that the MP steam exhaust from the turbine for case 3, remains at 42 tons/hr based on practical plant data which reveals that the de-superheat water required for the turbine exhaust steam is essentially zero. This was observed when trending the boiler feed water valve position against the MP steam temperature at SAFOR's MP steam header. The devaluation of HP steam to MP steam from the

turbine indicates that the degree of energy absorbed is maximised such that no de-superheat water is required to maintain the MP steam temperature of 220 °C. These observations are consistent with the plant operator's view.

- During periods of unplanned shutdown of the turbines or planned maintenance, HP load contributions between NCPX and SAFOR are adjusted. In the case of all additional HP steam available at SAFOR let down via pressure reduction, a total of 47.41 tons/hr MP steam will become available at the MP header. This is an undesirable operating load due to the previously mentioned export valve hydraulic limitation (refer Limitations and Delimitations, pp 4).
- A corrected turbine bypass calculation (case 4 corrected) is proposed for the MP header export load from SAFOR to the refinery. The export valve hydraulic limitation is discussed in Limitations and Delimitations section. The net MP export steam is adjusted down to 40 tons/hr as a safe export load for a healthy export valve open percent. The HP steam adjustments required between NCPX and SAFOR boilers are thereby calculated based on the HP to MP de-superheat water requirements previously discussed.

5.1.3 CHP Cost to Benefit Analysis

A business decision to install a retrofit CHP system depends largely on the economic justification accompanying the benefits of optimal energy utilisation. The method used for determining economic benefit is the DCF (Discounted Cash Flow). These results are presented below.

TABLE 5-4 - CHP DIFFERENTIAL OPERATING COSTS

	Eng. unit	Current	New Operating Point - Case 3	Current Case HP Steam Cost - ZAR/hr	Case 3 HP Steam Cost - ZAR/hr
NCPX HP to Header	t/h	96.97	78.5	36200.88	29305.69
SAFOR HP to Header	t/h	60.22	81.06	22481.03	30261.2
		157.19	159.56	58681.91	59566.89
Net			**2.37**		**884.97**

A key calculated parameter for a DCF analysis is the financial benefit Rand value. With reference to

Table 5-3- Refinery Steam Balance – Design Basis, page 56, it is possible to estimate the net operating cost change for case 3.

Tabulated above is the additional cost of plant operation in Rands per hour given the assumption that the additional load to the turbine for electricity generation is sourced from SAFOR boilers. A mere additional 2.37 tons/hr HP steam would be required at the header, given that the additional HP steam that would otherwise be let down from the HP to the MP header, will be reduced from the boiler HP demand. The turbine in essence will operate as a let-down station, however instead of isenthalpic; the devaluation of steam will be isentropic, while delivering a mechanical energy load for electricity generation.

Reduction in electricity import value is calculated by the simple product of electricity charges and electrical load delivered by the generator (see page 28)

By equation:

$$Electricity\ Billable\ Saving = Grid\ Load\ by\ Generator\ (MW) * Electricity\ Cost\ (\frac{R}{MWh})$$

Refer Equation 2-36

The estimated electrical grid load reduction attainable with a TG (turbine-generator) set is (conservatively) 2.0 MW

Therefore:

$Generated\ Electricity\ Value =$ R 1237.40/hr, by data provided in 2016.

The conservative value of 2.0 MW electrical generation as used in the base calculations, allows for inefficiencies of the energy conversion across the turbine, gearbox and generator set. These values are further discussed in the sensitivity analyses in later chapters.

5.2 Discounted Cash Flow Analysis

A DCF (Discounted Cash Flow) calculation is performed based on the electricity savings as calculated above. The economic viability of the project largely depends on the value assumed for fuel gas cost. These are further discussed in Variations in Fuel Gas Pricing, page 75.

TABLE 5-5– DCF SUMMARY DATA - OPERATING EXPENSE/PROFIT

Data for DCF Analysis - Operating Expense/Profit	Set 1 - FG Priced at R100/GJ	Set 2 - FG Priced at 0 R/GJ	Eng. Unit
Turbine Power Output Maximum	2.5	2.5	MW
Generator Efficiency maximum	0.90	0.90	[]
Maximum realistic power capability	2.25	2.25	MW_e
Operating hours per day	24	24	hours
Generation per day	54	54	MWh
Refinery Operating days per year	320	320.0	days
Electricity daily cost reduction	33 409.8	33409.8	R/day
Electricity cost reduction per year	10 691 136	10 691 136	R / year
Cost of generation - HP Steam	884.97	100.8	R/hr
Cost of generation - HP Steam	6 796 596.29	773 895.48	R/year
Net Operating Profit	3 894 539.71	9 917 240.52	R/Year

With reference to Table 5-5 A DCF calculation is performed based on the electricity savings as calculated above. The economic viability of the project largely depends on the value assumed for fuel gas cost. These are further discussed in Variations in Fuel Gas Pricing, page 80. For the purposes of this thesis, the data below is an estimation of the realisable refinery benefit with fuel gas costed at 0R/GJ. The basis of this estimate is the marginal difference in HP steam consumption for the retrofit option (Case3) vs the current operation (Case 2). Since only 2.37 tons/hr additional HP steam is consumed to achieve the desired HP load for the generator operation, the cost of additional fuel gas consumed is minimal relative to the cost of water for steam production. In addition, the variability (Standard deviation) of HP steam loads at the header is generally > 5.0 tons/hr per individual boiler. This implies that the difference in HP steam consumption resides within one standard deviation of the refinery boiler loads and is unlikely to pose a quantifiable additional cost to operation.

A summary of the estimated DCF data as used for Case 3 calculations is presented below:

TABLE 5-6 – CAPEX DATA FOR DCF

Data for DCF Analysis - Capital Expense		
Estimated Capital cost - Gen Set	11 000 000.00	Rands
*Estimated IFC - Engineering Services	1 000 000.00	Rands
**Estimated DFC - Installation	4 000 000.00	Rands
Total CAPEX estimate	16 000 000.00	Rands

* IFC – Indirect Field Costs

** DFC – Direct Field Costs

The CAPEX data presented above is based on a vendor quotation for a new generator set. In 2012, the estimated total project cost for a new turbine, generator set, switchgear and installation was < $200 per kW for units larger than 2000kW. The relative installation costs depending on complexity averaged around 75% of equipment costs. The total costs amount to $350/kW indicating that the cost in 2012 would have been R11.375m at an exchange rate of R13/USD. (Advanced Manufacturing Office - Energy Efficiency and Renewable Energy, 2012). For this project, a part of the CAPEX equipment costs (existing backpressure turbine – C4002) is available on site and ENGEN owned for retrofit application, resulting in reduced CAPEX projections. At the current estimated CAPEX of R16m, the estimated cost per kW amounts to $492/kW. This estimate is very conservative considering that the projected estimate cost per kW from 2012 to 2018 would have been $354/kW at an escalation of 10% per annum.

The compiled DCF table is presented below:

TABLE 5-7 – DISCOUNTED CASH FLOW SUMMARY TABLE

Eref: SAFOR Turbines - CHP retrofit								
	FY	2019	2020	2021	2022	2023	2024	2025
Input Factors	DCF Year	-1	0	1	2	3	4	5
RSA Escalation Index		1	1.06	1.12	1.19	1.25	1.32	1.39
RSA CPI Inflation %		6.60%	6.20%	5.70%	5.60%	5.50%	5.40%	5.30%
Rand to US Dollar Exchange		15.70	15.75	15.80	15.85	15.90	15.94	16.00
Engen Fixed Expense Escalation %		6.60%	6.20%	5.70%	5.60%	5.50%	5.40%	5.30%
Engen Variable Expense Escallation %		6.60%	6.20%	5.70%	5.60%	5.50%	5.40%	5.30%
Corporate Tax Rate %		28%	28%	28%	28%	28%	28%	28%
RSA Prime Interest rate		10.75%	10.75%	10.75%	10.75%	10.75%	10.75%	11.00%
Engen RSA WACC for projects		12.62%	12.62%	12.62%	12.62%	12.62%	12.62%	12.62%
Central Services Burden		1.93%	1.92%	1.94%	2.00%	2.00%	2.00%	2.00%
Tax Wear & Tear Allowance				40.00%	20.00%	20.00%	20.00%	
Unplanned capital tax rate		2.00%	2.00%	2.00%	2.00%	2.00%	2.00%	2.00%
Financial benefit - US$		0	0	627 673	662 823	699 278	737 039	776 102
Financial benefit converted to Rand million			0.00	9.917	10.506	11.119	11.748	12.418
Cash Flow (Rand Million)								
Integrated Benefit		0.00	0.00	9.92	10.51	11.12	11.75	12.42
Less: Royalties		0.00	-1.00	0.00	0.00	0.00	0.00	0.00
Less: Maintenance at 2% of equipment DFC		0.00	0.00	-0.12	-0.12	-0.12	-0.12	-0.12
Less: Central Services Burden		0.00	0.00	-0.19	-0.21	-0.22	-0.23	-0.25
Less: Tax Wear and Tear Allowance		0.00	0.00	-2.40	-1.20	-1.20	-1.20	0.00
Cash Flow before Tax		0.00	-1.00	7.20	8.98	9.58	10.19	12.05
Less: Tax Payable on unplanned capital		0.00	0.02	-0.14	-0.18	-0.19	-0.20	-0.24
Less: Corporate Tax		0.00	0.00	-2.02	-2.51	-2.68	-2.85	-3.37
Income after tax		0.00	-0.98	5.04	6.28	6.70	7.14	8.43
Add back Wear and Tear allowance		0.00	0.00	2.40	1.20	1.20	1.20	0.00
Working Capital Increase		0.00	0.00	0.00	0.00	0.00	0.00	0.00
Planned Project Capital cash outflow		-1.00	-15.00	0.00	0.00	0.00	0.00	0.00
Unplanned SIB Capex		-0.02	-0.30	0.00	0.00	0.00	0.00	0.00
After Tax Cash Flow		-1.02	-16.28	7.44	7.48	7.90	8.34	8.43
Cumulative After Tax Cash Flow		-1.02	-17.30	-9.86	-2.37	5.53	13.87	22.30
Present value		-1.15	-16.28	6.61	5.90	5.53	5.18	4.66
Net Persent Value		-1.15	-17.43	-10.82	-4.92	0.61	5.80	10.45
IRR (year n)		-1.36	-16.28	5.57	4.19	3.31	2.61	1.97
IRR	34%							
NPV @ 12.62% p.a.	R 8.24		Rand Million					
Payback Period	2.32		years					

5.2.1 Discussion

The DCF presented above summarises a realistic projection of profit returns for the expenditure of CAPEX. The project payback period of 2.32 years meets the business criteria for short-term beneficiation (less than 5 years' payback). The IRR and NPV are attractive for business investment. It is also noted that the benefit projection of R9.917m in the first year does not account rebates on carbon credit taxes. This will become an additional saving for the project and can be estimated when the framework for price rebates are further clarified by eThekwini industrial electricity sector. Sensitivities around the escalation of water and electricity prices are presented in Discounted Cash Flow – Profit Variations, section 5.3.4 page 75.

5.3 Sensitivities

5.3.1 Data Segmentation Approach

This section of the thesis aims to quantify the relationships between a range of dependant and independent variables in the context of this study. A data segmentation approach has been adopted, where operating ranges are pre-selected by applying data range filters to the raw data set, and calculations performed at each subset.

As indicated in the theory section: Research Technique - Plant Performance Triangle approach, page 29; data filtering techniques are applied to the historical and current data sets defined for this study. The data segmentation approach was thereby applied to the filtered data sets such that calculated properties were possible at each subset of the filtered data ranges.

According to (Robert H Perry, 1997), the motivation for analysis of plant performance is four-fold:

- Identify problems in the current operation
- Identify deteriorating performance in instruments, energy usage, equipment or catalysts.
- Identify better operating regions leading to improved product or operating efficiency
- Identify a better model leading to better design.

An example dataset for NCPX boiler 1 data segmentation is shown:

TABLE 5-8 – SAMPLE DATA SEGMENTATION SET – BOILER 1

Description	Eng Unit	Segment 01	Segment 02	Segment 03	Segment 04	Segment 05	Segment 06
Segment Range	t/hr	5.0	5.0	5.0	5.0	5.0	5.0
Segment Lower Range Load	t/hr	15.0	20.0	25.0	30.0	35.0	40.0
Segment Upper Range Load	t/hr	20.0	25.0	30.0	35.0	40.0	45.0
Total Data Points		17.0	97.0	83.0	40.0	71.0	122.0
Average Crude rate	m^3/d	15612.8	15646.4	15705.6	15809.3	15818.0	15710.6
Refinery total Steam Load	t/hr	148.3	156.4	185.2	189.9	178.3	189.8
HP Steam Pressure	kPag	38.4	38.5	38.6	38.6	37.6	37.2
HP steam Enthalpy	kJ/kg	3215.6	3216.5	3216.1	3216.5	3216.1	3219.0
FG Analyser MW	g/mol	20.5	21.1	22.2	22.7	24.2	24.1
Lab Data Calculated MW	g/mol	17.9	18.4	21.2	22.3	23.6	23.6
FG Std Norm Density	kg/nm^3	0.8	0.8	0.9	1.0	1.1	1.1
Air Std Norm Density	kg/nm^3	1.3	1.3	1.3	1.3	1.3	1.3
Hydrogen Sulphide in FG	ppm	54.9	25.0	121.9	131.9	110.4	278.0
Total Refinery FG demand	nm^3/hr	28092.8	28033.2	30781.3	30799.5	29981.9	31237.3

HP steam to C4001 turbine	t/hr	1.7	1.8	8.5	9.8	9.4	13.8
HP steam to C4002 turbine	t/hr	0.0	47.8	33.1	34.9	46.6	48.6
SAFOR Export MP to refinery	t/hr	25.6	20.5	32.1	36.9	26.2	23.6
LHV Fuels Gas	kJ/kg	46904.2	46572.1	46086.4	45868.9	45660.2	45716.5
LHV - FG Model	kJ/kg	47407.4	47111.7	45783.4	45398.0	45036.1	45051.2
Model Err	%	1.07	1.16	-0.66	-1.03	-1.37	-1.46
B1 Steam Load – average of segment	t/hr	18.0	22.0	27.6	31.9	38.1	41.5
% of Boiler MCR	%	39.99%	48.80%	61.39%	70.98%	84.76%	92.15%
Fuel Gas Fired – Vol Flow metered	nm^3/hr	1601.6	1920.1	2325.6	2507.6	2679.3	2878.0
Fuel Gas Fired – mass flow	kg/s	0.4	0.4	0.6	0.7	0.8	0.8
Actual Air Flow Rate	nm^3/hr	42365.6	38345.9	44593.1	44310.2	45461.3	45987.6
Actual Air Rate	kg/s	15.1	13.7	15.9	15.8	16.3	16.4
Feed Water Temperature-Inlet to economiser	degC	121.0	124.3	122.4	124.2	121.7	121.4
Feed Water Enthalpy	kJ/kg	524.9	510.6	517.6	524.1	513.5	512.2

Feed water density	kg/ m^3	944.1	940.1	940.0	944.1	940.1	940.0
Desuperheater water flow	kg/s	0.7	0.8	1.1	0.6	1.4	1.6
Flow out boiler	kg/s	5.0	6.1	7.7	8.9	10.6	11.5
Flow into DESUP	kg/s	4.3	5.3	6.6	8.3	9.2	10.0
Steam Enthalpy before desuperheater	kJ/kg	3664.6	3617.5	3660.3	3417.0	3627.0	3641.3
Excess Oxygen	%	10.5	8.0	7.1	6.3	5.2	4.6
Stack Temperature	**°C**	**284.3**	**333.9**	**347.3**	**335.9**	**342.8**	**348.4**
Stoichiometric Air – Mass flow	kg/s	8.5	10.3	13.4	14.8	16.2	17.4
Stoichiometric Air – Volumetric flow	nm^3/hr	23670.7	28828.1	37533.0	41397.9	45293.7	48600.3
Stoichiometric A:F Ratio - mass	[-]	23.8	23.5	22.0	21.4	20.6	20.7
Actual A:F Ratio - mass	[-]	42.6	31.3	26.2	22.9	20.7	19.6
Fired Duty	kW	16662.9	20386.5	28096.5	31733.1	35854.4	38446.5
Absorbed Duty	kW	13451.5	16506.2	20706.2	23887.5	28635.2	31180.2
Thermal Efficiency	%	80.73%	80.97%	73.70%	75.28%	79.87%	81.10%
Thermal Efficiency Method 2	%	79.9	79.9	80.1	81.6	82.2	82.4

Duty/MCR	kWh/tonx10	92.6	92.8	101.7	99.4	94.0	92.7
Percent Above Stoic A:F ratio	%	78.98%	33.02%	18.81%	7.03%	0.37%	-5.38%

5.3.2 Effect of Hydrogen Content vs MW vs LHV

The fuel gas system at ENREF is a highly integrated one where make up gasses from various sub processes are fed into a common fuel gas header. The hydrocarbon constituents as a result vary which affects the molecular weight of the fuel gas and thereby the heat value. The net effect is observed in the firing efficiency of the boilers, where at times sub stoichiometric air: fuel ratios can pose a serious fire risk within the boiler combustion chambers. An analysis was therefore performed on the varying degree of hydrogen content in the fuel gas and the resulting impact on the heat value and stoichiometric air. These are presented below:

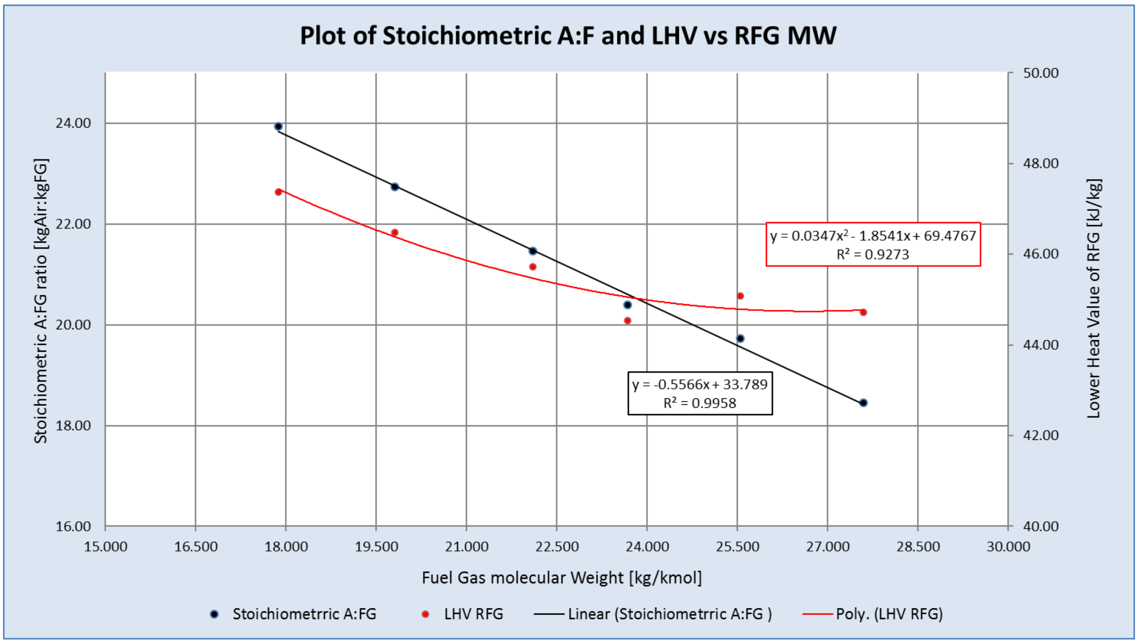


FIGURE 5-1 – FUEL GAS HEAT VALUE AND STOICHIOMETRIC AIR RELATIONSHIPS

As observed, heat value of the fuel gas decreases exponentially with increasing molecular weight. This appears counter intuitive since increased molecular weight generally implies increased heat value due to higher individual heat values of heavier hydrocarbon chains. However, since hydrogen has a low molecular weight and high heat value (120 000 kJ/kg) increased hydrogen reduces the overall molecular weight and increases the heat value per kg of fuel gas. Stoichiometric air subsequently increases with more hydrogen as the air: fuel ratio for combustion of hydrogen as a component is 34.2 kg Air / kg Hydrogen. It is however shown below that, the increased hydrogen in the fuel gas does reduce the energy density since the molecular weight is decreased. As the fuel gas is metered in volumetric units, the energy density is a more meaningful measure of the fuel gas quality.

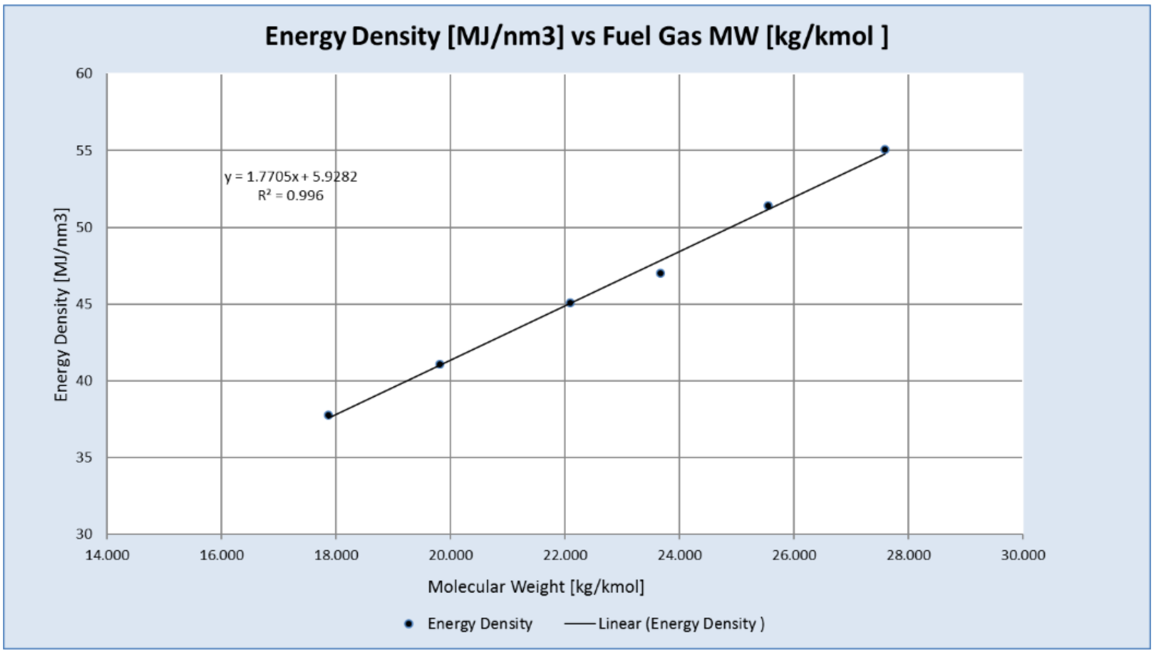


FIGURE 5-2 – EFFECT OF ENERGY DENSITY ON FUEL GAS MOLECULAR WEIGHT

As shown above, the energy density of the fuel gas is directly proportional to the fuel gas molecular weight implying that the reduced hydrogen content at higher molecular weights clearly indicates a better quality of fuel gas per nm^3.

5.3.3 Boiler Fired Duty Vs Load

Each of the 5 boilers on site are instrumented with fuel gas flowmeters and HP steam flow rates. It is therefore possible to determine the relationships between fired duty (kW) and HP steam loads (tons/hr). The methodology to determine the relationship between the fired duty and HP steam load follows the data segmentation approach discussed above. The significance of analysing boiler efficiencies is that HP load distribution for the turbine could be optimised between boilers.

These are presented graphically below.

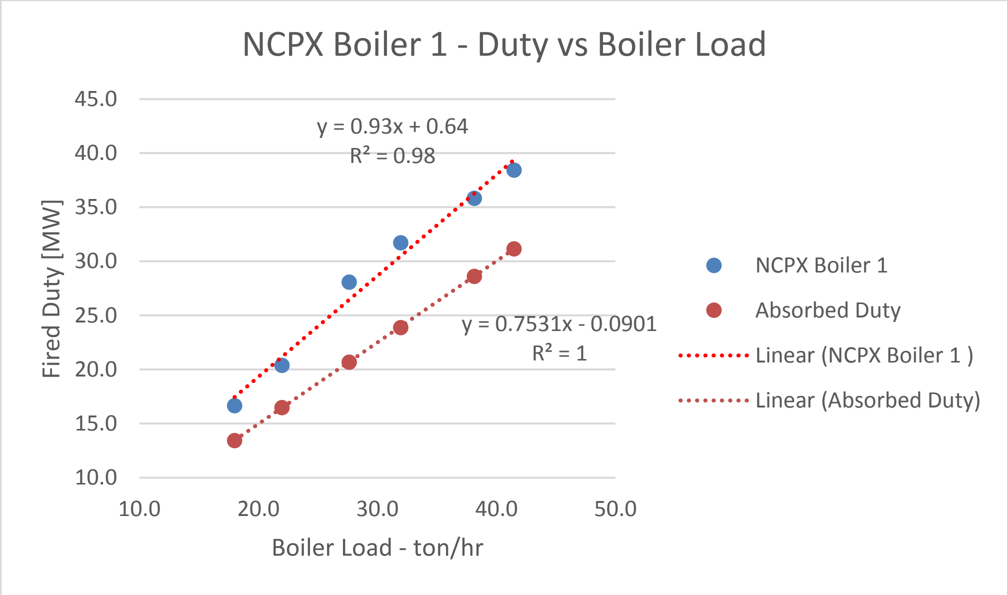


FIGURE 5-3 - NCPX BOILER 1 DUTY TO LOAD RELATIONSHIPS

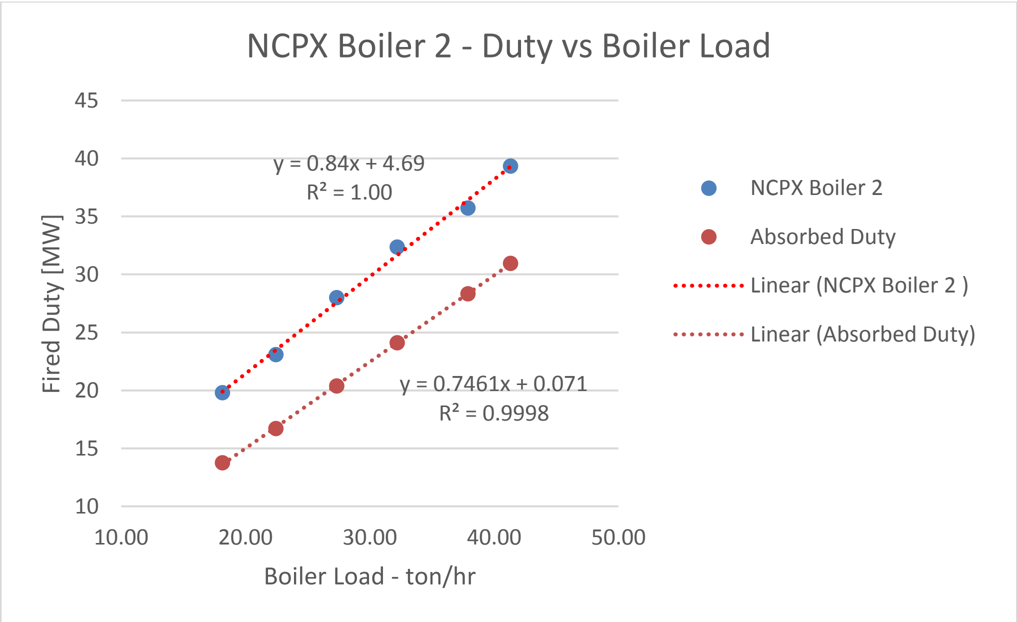


FIGURE 5-4 - NCPX BOILER 2 DUTY TO LOAD RELATIONSHIPS

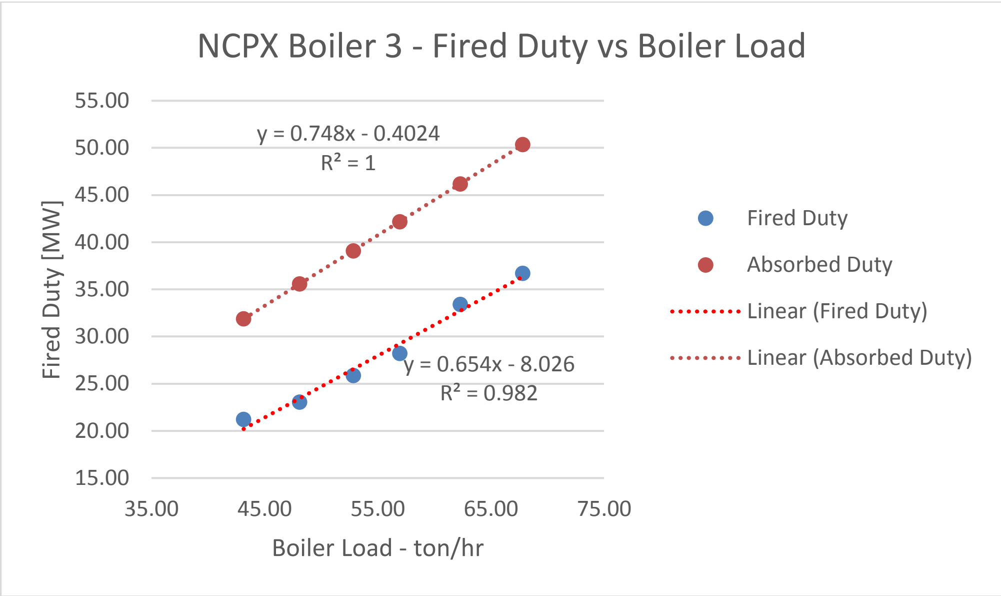


FIGURE 5-5 - NCPX BOILER 3 DUTY TO LOAD RELATIONSHIPS

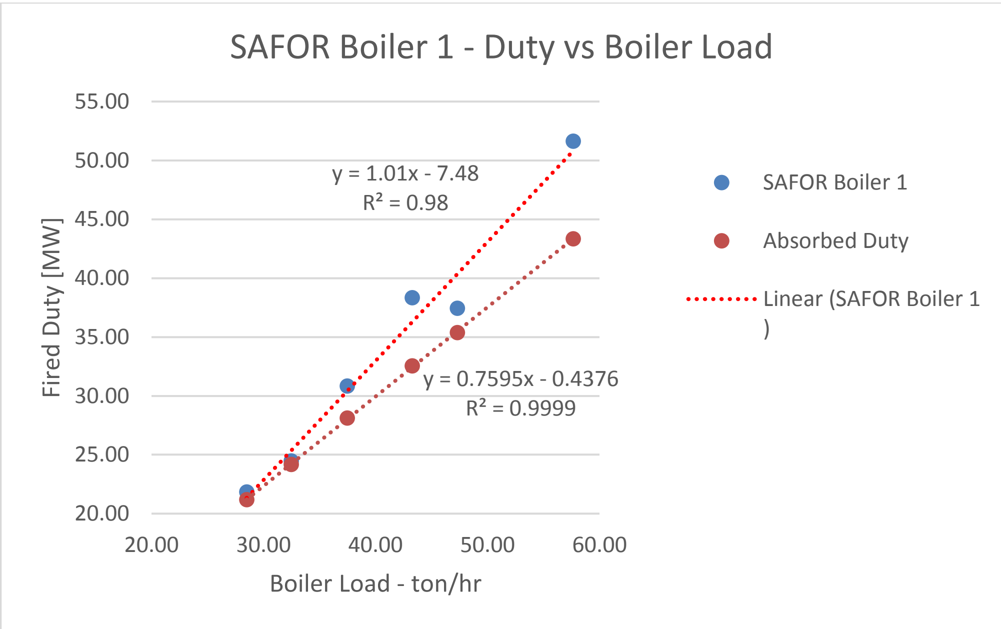


FIGURE 5-6 – SAFOR BOILER 1 DUTY TO LOAD RELATIONSHIPS

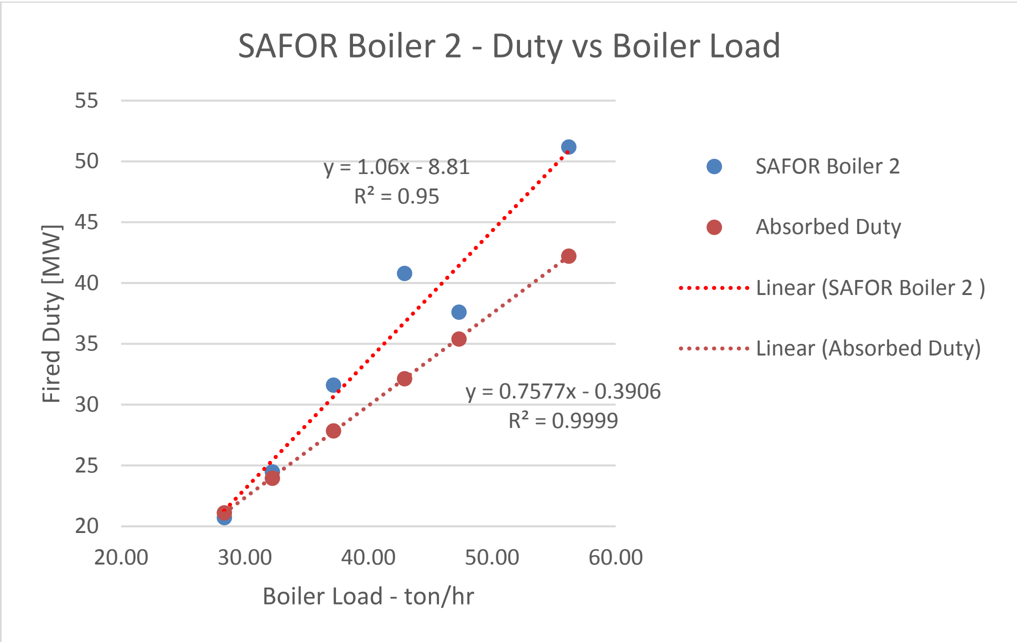


FIGURE 5-7 – SAFOR BOILER 2 DUTY TO LOAD RELATIONSHIPS

A closer examination of the duty vs load relationships above reveal that discrepancies exist between calculated fired duties and absorbed duty. Specifically, for NCPX boiler 3, where the fired duty slope is lesser than the absorbed duty. This would not be possible given the laws of conservation of energy. The results indicate that there are possibly inconsistent flowmeter calibrations for NCPX boiler 3.

TABLE 5-9- SUMMARY TABLE - BOILER FIRED AND ABSORBED DUTIES

	Fired Duty Slope	Absorbed Duty Slope
Boiler No	MWH/ton HP	MWH/ton HP
NCPX Boiler 1	0.934	0.753
NCPX Boiler 2	0.837	0.746
NCPX Boiler 3	0.654	0.748
SAFOR Boiler 1	1.010	0.760
SAFOR Boiler 2	1.060	0.758

The above summary table indicates that the average absorbed duty per ton HP steam produced is 0.753 MWh/ton. Graphically, the slopes of the fired and absorbed duties very closely resemble co-linearity for comparative purposes. The slope ratio however, differs between boilers and between loads. As boiler loads increases, fired duties increase larger than proportionally to the required additional absorbed duty. This indicates that boilers become less efficient. It is presumed that boilers become more efficient with increased load, therefore other operating parameters of the boiler require analysis. NCPX boiler 3 calculation results are not conclusive and subject to further investigation. This is likely due to the limitation of instrument accuracy.

5.3.4 Discounted Cash Flow – Profit Variations

5.3.4.1 Variations in Fuel Gas Pricing

A study of the variable cost of fuel gas in relation to the NPV and IRR gives a direct indication of the viability of the project if profitable in the near term (short-term benefits recovered by capital expense within 5 years). The cost of fuel gas affects the direct cost of HP steam generation, which determines the differential profit margin between operating costs and electricity savings. A sensitivity is presented below with cost variations in fuel gas vs NPV and IRR.

FIGURE 5-8 – EFFECT OF FUEL GAS PRICE ON NPV AND IRR

It is clear from the plot above that a fuel gas cost above R62/GJ results in an unprofitable business scenario for short term beneficiation. The WACC at 12.62% intersects the IRR line at approximately R62/GJ validating the calculation. Calculations in the plot assumes that the cost of boiler feed water remains at R40/kl.

5.3.4.2 Variations in Water Pricing

A sensitivity of water costs as a function of NPV and IRR is important due to the known water shortage challenges in South Africa. Fluctuations in water prices can therefore affect project economics quite severely.

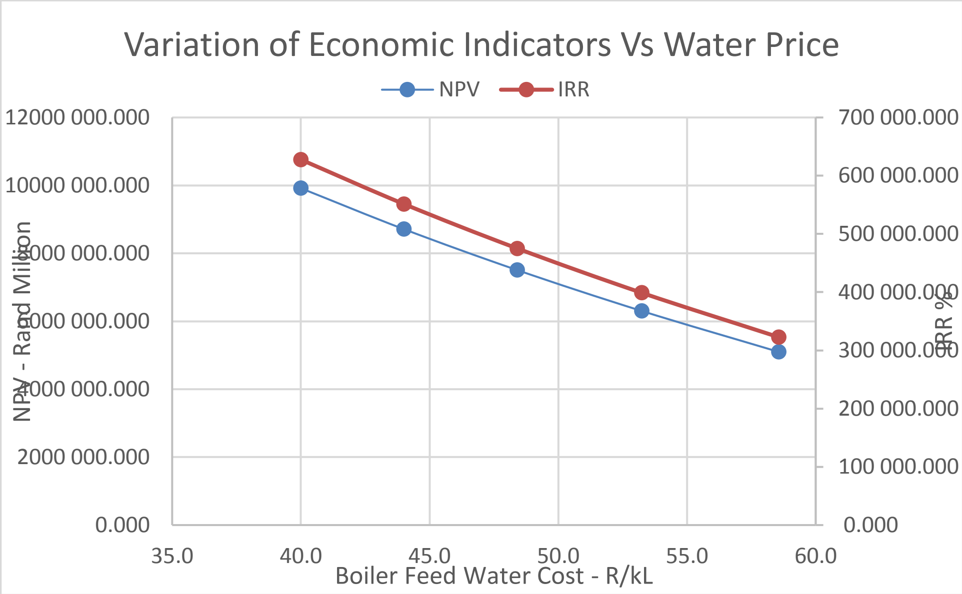


FIGURE 5-9 – VARIATIONS OF BOILER FEED WATER PRICE

Presented graphically above, the breakeven cost of boiler feed water for project economic viability is approximately R52.50 per kL. The current cost at R40/kL is merely 31% lower than the breakeven cost. It is estimated that water costs will increase by 10 % per annum and possibly more for industrial consumers, therefore long-term economics are weighted by water price sensitivity.

6 Conclusions and Recommendations

- It can be concluded that sufficient data of good quality was available for this thesis and the related comparative studies.
- In the assessment of the on-site boiler operation; pre and post the mid 2014 SAFOR partial shutdown, it is concluded that the total HP demand for the site reduced from 193.62 t/hr to 158.39 t/hr (Reduction of 18.2%). Boiler 3 was not in service during this period.
- Fuel gas flow measurement is an important parameter for cross verification of heat and material balances as per the plant performance triangle approach, see page 29. Due to non-convergence of boiler 3 heat balance, fuel gas volumes, steam costs and averaging calculations excluded boiler 3 results. It is recommended that an instrument audit be conducted around boiler 3.
- The new turbine – gearbox - generator configuration is to provide at least 2.0 MWe onsite electrical generation.
- It is concluded that the optimal boiler configuration for the turbine steam demand is to maximise SAFOR boiler operation and export additional MP steam from SAFOR to refinery up to the maximum export valve limit for maximum power output.
- Project economics indicate a viable and attractive return on investment with a payback period less than 3 years. This does not account for the further savings that will be incorporated by the carbon credit rebate framework.
- ENGEN refinery is equipped with 11KV and 6.6 kV voltage loads from substation transformers. The source voltage to the generator is therefore to be synchronised with SAFOR's 6.6kV substation source voltage. Spare capacity is available at the substations.
- A civil and electrical load study is recommended for the installation of the generator module replacing the compressor since the generator rotor and stator is a heavier unit than that of the compressor. Substation loading are subject to an electrical design review.
- In an effort to reduce LP venting (a common refinery challenge), it is recommended that further study be conducted on the use of thermo-compressors on LP steam systems. Crude distillation towers often consume stripping steam at medium

pressure (10 barg). Thermo-compressors would essentially convert LP steam to MP steam by This will ensure optimal use of excess MP steam let down into LP headers and reduce steam wastage by LP venting.

- CO (Carbon Monoxide) analysers are recommended for evaluation of combustion characterisation in flue gas. These analysers are commercially available to refiners and other process/power plants. ENGEN would benefit from reliable CO analysers for optimal boiler operation, complete combustion and thereby reduced fuel gas consumption.

6.1 Scope of work for Further Research

- Data segmentation approach – It was found that the methodology prescribed in Perry's Chemical Engineers Handbook for plant performance analysis is indeed a technically enriching and systemic approach to plant analysis. Since multiple process operating parameters are segregated by operating ranges for the same time-series, visual and correlating data are easier identified. As a result, relationships between measured variables are surfaced in a way to prompt further exploration and deeper understanding of big data sets. There is therefore room for exploitation of research into process operation by employing this technique.
- Industrial CHP applications is a growing area of interest driven by companies looking to reduce energy costs. The impact of rising energy cost and increasing energy demand in process operation has ranked energy consumption a key input parameter into process design. Further research is therefore required into holistic energy optimisation within plant operation. Co-ordinated research in the front-end design for newly built plants and energy audits for existing plants are critical success factors. The research will require detailed study of optimum fossil fuel energy choices, power grid integrated renewable energy, heat integration and auditable performance management systems.
- During the plant performance assessment of the various operating and performance parameters as detailed in page 29), it is observed that flue gas exiting boiler stacks are often above 300 degC. It is also apparent that higher boiler steam loads may result in higher exit temperatures. Further research in the area of flue gas heat recovery may provide valuable insight and cost savings to refiners and power plant companies. Hot flue gases contain 50 to 300 ppm hydrogen sulphides (which

increases with increasing firebox duty. Flue gases also contain measureable carbon dioxide, carbon monoxide, sulphur dioxide and nitrous oxide compounds. Due to high costs of scrubbing processes for gas purification, flue gases are released to atmosphere. Should viable research be possible for flue gas heat recovery and combined gas purification, there may exist a very strong business case for technology development.

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8 Appendices

8.1 Company Profile – ENGEN Oil Refinery

A public profile of the ENGEN operating business is available on Wikipedia. Partially extracted information is presented below as point of reference for the reader.

Engen Petroleum

Engen Petroleum Limited	
Type	Public
Traded as	JSE: [1]
Industry	Oil and gas
Predecessor	Mobil South Africa Engen Petroleum Limited
Founded	1881
Headquarters	Cape Town, South Africa
Area served	Africa, Indian Ocean Islands
Key people	Yusa' Hassan (Managing Director and CEO) Dato' Sri Syed Zainal Abidin (Chairman)[1]

Products	Fuels, lubricants, petrochemicals
Revenue	R70 033 million (2017)[2]
Operating income	R5.155 million (2017)[2]
Net income	R3.315 million (2017)[2]
Total assets	R40.878 million (2017)[2]
Number of employees	3,485 (2017)[2]
Website	engen.co.za

Engen Petroleum is a South African oil company focusing on the downstream refined petroleum products market and related businesses. The company's core functions are the refining of crude oil, the marketing of primary refined petroleum products and the provision of convenience services via an extensive retail network. Until 1990, it was part of Mobil Oil. In 1993, it changed the brand name to Engen.[3] The company is present in 17 countries and exports products to over 30 more countries, mostly in Africa and the Indian Ocean Islands.

Engen operates a refinery in Durban that has a nameplate capacity of 120,000 barrels (19,000 m^3) per day and operates approximately 1,450 service stations across sub-Saharan Africa and Indian Ocean Islands. A number of Engen's service stations are operated on a franchise basis.[4] Engen operates its own transport fleet with approximately 180 bulk fuel tankers.[3]

Engen has partnered with numerous South African businesses, including Woolworths, Wimpy, Debonairs Pizza, and Steers, which have operations at certain Engen service stations.[4]

Today, Engen Petroleum is active in South Africa, Botswana, Namibia, Zimbabwe, Mozambique, Kenya, Ghana, Gabon, Tanzania, Rwanda, Zambia, Malawi, Lesotho, Swaziland, Mauritius, Réunion and The Democratic Republic of the Congo.[5] The company is also listed on the Botswana Stock Exchange and is a constituent of the BSE Domestic Company Index.

Ownership

- Until 1990 Mobil South Africa
- 1990-1996 Gencor
- 1996 30% PETRONAS
- 1998 100% PETRONAS
- 1998-2017 80% PETRONAS and 20% Phembani Group
- Today 74% PETRONAS, 20% Phembani Group and 6% Phembani-led Consortium

Websites

- http://www.engen.co.za Engen Petroleum
- http://www.engenoil.com Engen Africa (outside South Africa)

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3. *^ Jump up to:[a b] Engen. "History of Engen". Engen. Archived from the original on 2016-06-01.*
4. *^ Jump up to:[a b] "Engen Franchise Opportunities". Engen. Archived from the original on 2016-04-15. Retrieved 2016-03-22.*
5. *^ "Engen in Africa". Engen. Archived from the original on 2014-10-29.*

End of referencing for Wikipedia article cited above

8.2 Fuel Gas Calculations

8.2.1 LHV Calculation

Step 1 – Calculate the contribution of the individual components in FG to the total Molecular weight of the mixture

E.g. Hydrogen – Molar Mass = 2 $\frac{kg}{kmol}$

Percent Hydrogen in FG = 43% therefore

Contribution from Hydrogen to MW = 2 $\frac{kg}{kmol} * \frac{43k\ mol}{100kmol}\ (43\ mol\ \%) = 0.86\frac{kg}{kmol}$

Therefore the generic equation to be used for the MW contribution is

$$MW_{mix}^{ave} = \left\{\frac{\left(\sum_{i=1}^{n} MolFrac_i^{FG}\ x\ MW_i\right)}{\left(\sum_{i=1}^{n} MolFrac_i^{FG}\ \right)}\right\} + \frac{PPM_{H_2S}}{1\ x10^6}\ x\ MW_{H_2S}$$

Refer page 24

Step 2 – Calculate the quantity of inert gas material – These are categorised as pure components of Oxygen, Nitrogen and Carbon Dioxide

The following equation defines the percent inerts

$$\%Inert\ Gases = \ Mol\%O_2^{FG} + Mol\%N_2^{FG} + Mol\%CO_2^{FG}$$

Step 3 – Calculate the NORMAL density of the fuel gas mixture using the relationship of Ideal Gas Law

Where

$$PV = nRT$$

And

P = Pressure in units – kPa @ 101.325 kPa at Normal conditions

V = Volume of the gas at STP – m^3 (Standard molar volume)

N = Number of moles of the gas = 1 mole at STP (Standard Molar)

R = Univeral gas constant = 8.3143 $\frac{J}{mol.K}$

T = Temperature of the gas = 273 K or 0 °C

Rearranging

$\frac{n}{V} = \frac{P}{R.T}$ And $n = \frac{m}{M}$ thereby $\frac{m}{M.V} = \frac{\rho}{M} = \frac{P}{R.T}$

Where $m = Mass\ in\ kg$

$\rho = gas\ density\ in\ \frac{kg}{m^3}$

And $M = Molar\ mass\ in\ \frac{kg}{kmol}$

Therefore at STP or NORMAL conditions

$$\rho^{stp} = \frac{M.P}{R.T} = \frac{M*101.325kPa}{8.3143^{kJ}/_{kmol.K}\ x\ 273.15K} = M * 0.044616\frac{kmol}{m^3}\ or\ \frac{M}{22.414}\frac{m^3}{kmol}$$

Therefore as an example the standard density of Hydrogen will be

$$\rho_{H_2}^{stp} = \frac{2.0158^{kg}/_{kmol}}{22.414^{m^3}/_{kmol}} = 0.0899\frac{kg}{m^3}$$

Therefore the Standard Normal Density of the FG at a particular MW will be defined by:

$$\rho_{FG}^{stp} = \frac{MW_{FG}^{ave}}{22.414}$$

Step 4 – Calculate component Heat Values

Each component has a defined Heat of combustion thermodynamic property

$$LHV_i = Molfrac_i\ x\ MW_i\ \left({}^{kg}/_{kmol}\right)\ x\ \Delta H_{comb}\left({}^{kJ}/_{kg}\right)$$

Where

$Molfrac_i$ Denotes the mole fraction of species i in the Fuel gas mixture []

$MW_i\ \left({}^{kg}/_{kmol}\right)$ Denotes the molecular weight of species i

$\Delta H_{comb}\left({}^{kJ}/_{kg}\right)$ Denotes the Lower heating value – Heat of combustion e.g. Hydrogen

$$LHV_{H_2} = 0.43\ x\ 2.0158\ \left({}^{lb}/_{lbmol}\right) x\ 51623\left({}^{btu}/_{lb}\right) = 44772\ \left({}^{btu}/_{lbmol}\right)$$

Convert from $\left({}^{btu}/_{lbmol}\right)$ to $\left({}^{kJ}/_{kgmol}\right)$ $divide\ by\ 2.326\ \ since\ 2.326\left({}^{kJ}/_{kgmol}\right) = 1\left({}^{btu}/_{lbmole}\right)$

Therefore, the LHV of the Fuel gas mixture will be defined as (Corrected for inert components)

$$LHV_{Corr}^{FG} = \left[\left[\sum_{i=1}^{n}\left[Molfrac_i\ x\ MW_i\ \left({}^{kg}/_{kmol}\right)\ x\ \Delta H_i^{comb}\ \left({}^{kJ}/_{kg}\right)\right]\right]\right] x\ (1 - \%Inert\ Gases)$$

The energy density is therefore defined as

$$Normal\ Energy\ Density\left({}^{kJ}/_{nm^3}\right) = LHV_{corr}^{FG}\left({}^{kJ}/_{kg}\right)\ x\ \rho_{FG}^{stp}\ \left({}^{kg}/_{nm^3}\right)$$

The definition of BFOE (Barrel of Fuel Oil equivalent) as used for economic calculations and presented as:

$1BFOE = 6.38GJ$

Therefore, a stream with energy of 1 BFOE per day is equivalent to:

$$1\frac{BFOE}{day} = 6.38\frac{GJ}{day}x\frac{1000000kJ}{GJ}\frac{1day}{24hr}x\frac{1hr}{3600s} = 73.84\ kW$$

Hence

$$1kW = \frac{1}{73.84} = \sim 0.0135\ BFOE/day$$

8.3 CHP Sample Calculations

For sample calculations explained below – reference is made to Section 4.2, pages 39

8.3.1 Part 1 – Energy Supplied by Pump – State 6 to State 1

$\dot{Q}_{pump}$ $= \dot{m}(h_1 - h_6)$

$$= 42\,\frac{tons}{hr} * \frac{1000kg}{ton} * \frac{1hr}{3600s} * (549.9 - 535.35)\,\frac{kJ}{kg}$$

= 169.75 kW

$$= \frac{169.75kW}{73.84kW/BFOE} = 2.3\ BFOE$$

(By validation, the power requirement of the pump correlates closely with the pump datasheet specification of 233 BHP or 173 kW)

8.3.2 Part 2a – Determining Fired Duty at the boiler

Heat input into the boiler is determined by calculating the fired duty (kW) to the boiler, which is a product of the energy density (kJ/nm^3) and volumetric flow rate (nm^3/s) of the fired fuel gas. Volumetric flow rates are measured for fuel gas flows by industry accepted instrumented standards.

In determining the volumetric flow of fuel gas, it is necessary to know the fired duty requirement of the boiler in raising the feed water from state 1 to state 2. To determine the fired duty for case 3 and 4, a plot is drawn of the fired duty vs % MCR (maximum continuous rating) of each SAFOR boiler. These are presented below:

By application of the regressed equations in

Figure 4-2 – SAFOR boiler 1 Fuel gas fired duty vs boiler load; volumetric flowrate of fuel gas is calculated as follows:

SAFOR Boiler 1 example

y = 64710x - 7481.4 → Regressed correlation of fired duty vs boiler load

Therefore, for Case 3 design condition:

X = 0.8549 MCR, equivalent to Steam load into boiler of 54.71 tons/hr

By substitution,

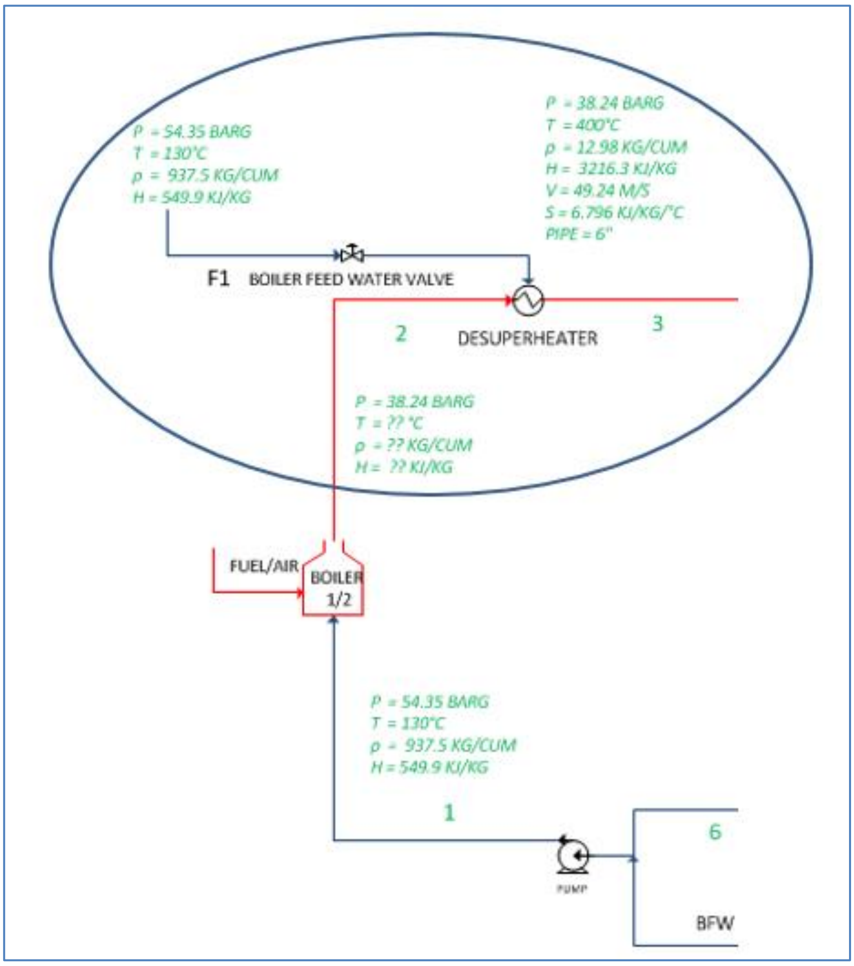
$$y = 64710 * 0.8549 - 7481.4 = 47839.2\ kW$$

In addition, by using the design energy density of the FG, the volumetric rate of fuel gas is given by:

$$\dot{V}_{fg} = \frac{Q_{fired}}{LHV} = \frac{47839.2kJ/s}{45812\ kJ/nm^3} = 1.253\ nm^3/s$$

8.3.3 Part 2b – Determining State 2 – Boiler exhaust steam condition

For State 2, the Enthalpy, temperature and density of the steam exiting the boiler is not known or measured. The procedure for the determination of the unknowns is outlined in State Descriptions – Real Rankine Cycle, pp 43.



SCREENCAP 8-1 – SCHEMATIC FOR SAMPLE CALCULATION

Data applicable to SAFOR boiler 1 – case 1 – Sample Calculation

1. Steam demand (F2) = 56.14 tons/hr
2. HP header Pressure = 38.24 bar.g
3. De-superheat water (F2) = 2.81 tons/hr
4. Fuel gas LHV = 45750 kJ/kg
5. Fuel gas STD density = 1.07 kg/nm^3
6. Fuel gas flow rate = 3347.63 nm^3/hr
7. Enthalpy of F1 = 549.9 kJ/kg
8. Enthalpy State 2 = unknown
9. Temperature State 2 = unknown
10. *Enthalpy State 3* = 3216.3 kJ/kg

Step 1 – Apply defining heat and material balances around the control volume

Apply a material and energy balance across the de-superheater (refer – pp 43 State Descriptions – Real Rankine Cycle).

$F_1H_1 + F_2H_2 = F_3H_3$ … (a)

$F_1 + F_2 = F_3$ … (b)

By rearranging:

$F_2 = F_3 - F_1$ … (c)

Therefore, by back substitution of (c) into (a)

$H_2 = \frac{F_3H_3 - F_1H_1}{F_3 - F_1}$ … (d)

Computing with provided data above:

$H_2 = \frac{56.14*3216.3-2.81*549.9}{56.14-2.81} = 3356.79\ \frac{kJ}{kg}$ (Note engineering units have been consistent)

From steam tables, the corresponding temperature at state 2 is therefore:

$T_{state\ 2} = T(H, P) = T(3357kJ/kg, 38.24barg)$ = 461.5 °C

$S_{state\ 2} = 6.998\ \frac{kJ}{kg.°C}$

Step 2 – Calculate fired duty from Fuel Gas Combustion (See page 25)

$$Q_b^f(kW) = (45750 * 1.07)\left({}^{kJ}/_{nm^3}\right)x\,\frac{3347.63}{3600}\left({}^{nm^3}/_{s}\right) = 45520.8kW$$

Step 3 – Calculate absorbed duty at boiler (See page 25)

$$Q_b^a(kW) = \left(\frac{56.14-2.81}{3600} * 1000\right)\left({}^{kg}/_{s}\right)x(3356.79 - 549.9) = 41580.96kW$$

Note – the actual flow exiting the boiler is F_2, which is obtained by subtraction of the boiler feed water added to the desuperheater from the steam to the turbine (State 3).

Provided data as measured or calculated from plant instruments: Refer Table 4-5 – Known process parameters- Cases 1 - 4, pp 45

Step 4 – Calculate Boiler Efficiency – Direct Method (See page 25)

$$\varepsilon = \frac{\varphi_{out}}{\varphi_{in}} = \frac{41580.96}{45520.8} = 0.91$$

8.3.4 Part 3 – Determining turbine shaft work

The equation defining turbine shaft work is given by (*ignoring Potential energy changes*):

$$\frac{\dot{W}_{turbine}}{\dot{m}} = h_i - h_e + \frac{v_i^2 - v_e^2}{2}$$

The available shaft work for the design case load of 42 tons/hour is therefore:

$0 = \frac{\dot{Q}_{cv}}{\dot{m}} - \frac{\dot{W}_{cv}}{\dot{m}} + (h_i - h_e) + \frac{v_i^2 - v_e^2}{2} + g(z_i - z_e)$ Refer page 19

Since no heat is added across the turbine, and applying the data for case 3 with velocity changes:

$$\dot{W}_{turbine} = 42\frac{ton}{hr} x \frac{1hr}{3600s} x \frac{1000kg}{ton} x \left\{(3216.3 - 2983.4)\frac{kJ}{kg} + \frac{49.24^2 - 53.58^2}{2}\frac{m^2}{s^2}\right\}$$

$$\dot{W}_{turbine} = 2494\ kW$$

It is noted that the turbine datasheet indicates the maximum power output of the turbine is 2500kW. The design steam rate is therefore validated at 42 tons/hr or 16.84kg/kW/hr to maintain the maximum turbine power output.

8.3.5 Part 4 – Calculating turbine performance metrics

8.3.5.1 Isentropic Efficiency:

In an ideal expansion across a turbine, no entropy is generated and all energy is completely converted from thermal to shaft work. This is not a real world scenario and is merely used as a reference measure against the turbine efficiency or isentropic efficiency.

The efficiency of the turbine as a performance metric is thereby calculated as a percentage of the isentropic enthalpy change.

The following data therefore applies:

$H_{state\ 3-state\ 4} = 3216.3 - 2983.4 = 232.9\frac{kJ}{kg}$ (Actual Enthalpy Change)

$H_{state\ 3-state\ 4s} = 3216.3 - 2899.56 = 316.74\frac{kJ}{kg}$ (Isentropic enthalpy change)

$\epsilon_{isen} = \frac{232.9}{316.74} * 100 = 73.53\ \%$ (Isentropic Efficiency)

8.3.5.2 Back work ratio

The BWR or Back Work ratio is another way to describe the performance of the of the generation cycle. It is defined as the ratio of pump work required and turbine work generated.

The accompanying equation for the BWR gives – See page Back work ratio, page 27:

$$\boldsymbol{BWR = \frac{-W_{s,P}}{W_{s,T}} = \frac{H_1 - H_4}{H_2 - H_3}}$$

Therefore, based on case 3 practical CHP design conditions

$$BWR = \frac{-W_{s,P}}{W_{s,T}} = \frac{-(535.35 - 549.9)}{3216.3 - 2983.4} = \frac{14.55\ kJ/kg}{232.9\ kJ/kg} = 0.0625$$

The above calculation implies that the BWR of the pump relative to the turbine shaft work output is 6.25%.

8.4 Steam Let-down Calculations

HP steam is let down via pressure reducing valves to MP steam at SAFOR and North Complex headers. One of the limitations of this thesis is the flow meter range at the North boiler let-down station. The energy balance therefore for the calculation of MP steam volumes and properties is based on the SAFOR let-down station. A schematic below explains:

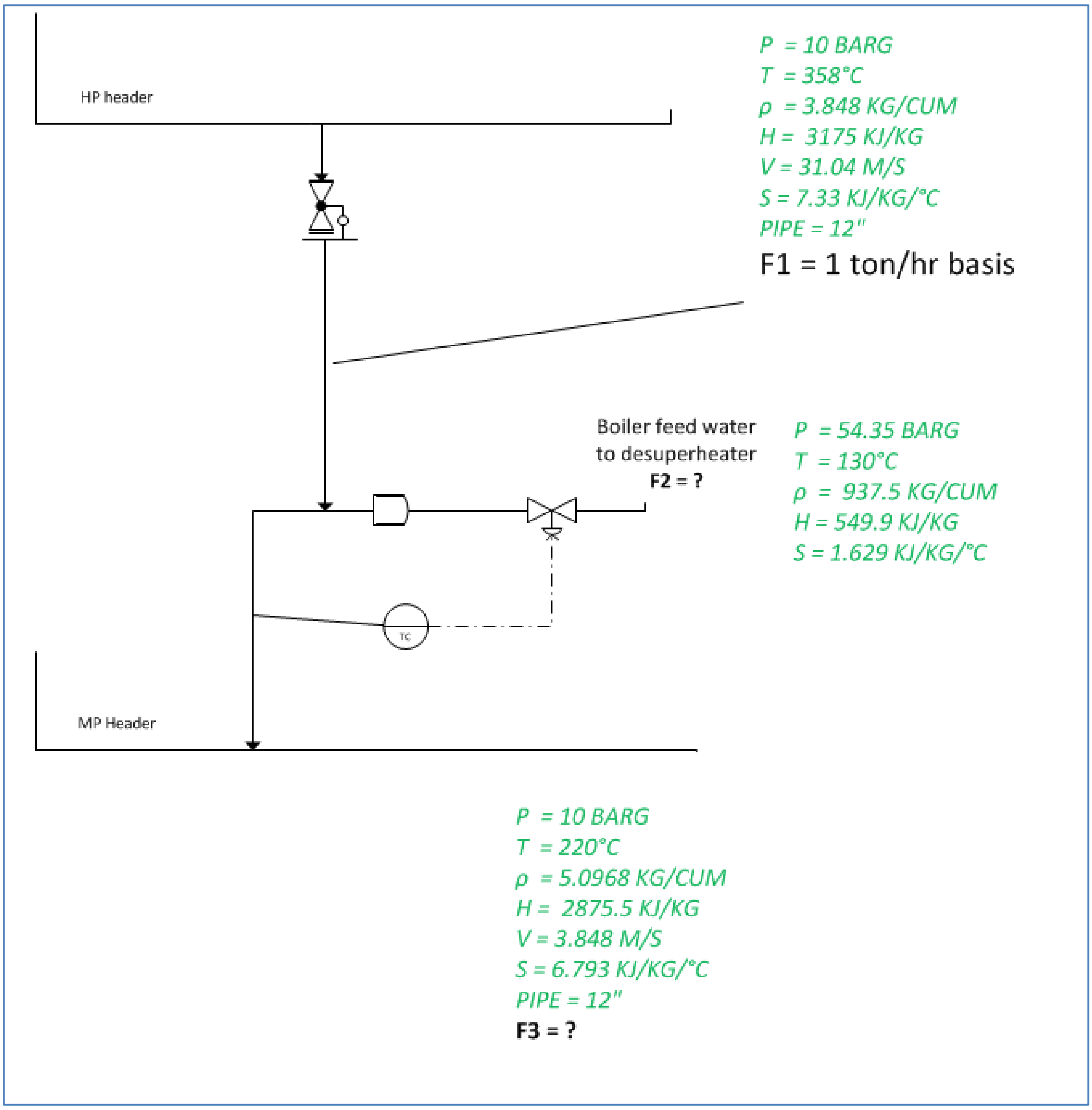


FIGURE 8-1 - HP - MP LET-DOWN DIAGRAM

Figure 8-1 - HP - MP Let-down Diagram above shows the general arrangement of a pressure reducing steam station. The calculations of the boiler feed water quantity and net MP produced are derived from the simultaneous solving of the mass and energy balances:

Mass Balance

$$F_1 + F_2 = F_3 \qquad \ldots (a)$$

Energy Balance

$$F_1 H_1 + F_2 H_2 = F_3 H_3 \qquad \ldots (b)$$

The two unknowns in the equation above are F_2 and F_3 – Desuperheater water flow and Net MP steam produced.

By applying the balances to per ton of HP steam:

$1 + F_2 = F_3$ … (c)

Substitute (c) into (b) and re-arrange

$F_2 = \frac{H_1 - H_3}{H_3 - H_2}$ … (d)

$$F_2 = \frac{3175 - 2875.5}{2875.5 - 549.9} = \frac{299.5}{2325.6} = 0.129\ \frac{ton\ MP}{ton\ HP}$$

Therefore

$F_3 = 1 + 0.129 = 1.129$ ton MP

The above implies that for every ton of HP steam passing through the PRV and desuperheater, 1.129 tons of MP steam is produced.

8.5 Glossary

Terms, Abbreviations, Acronyms and Engineering Units as used in this Thesis

API	American Petroleum Institute
BWR	Back-work Ratio
BFOE	Barrels of Fuel Oil equivalent - energy term
BFW	Boiler feed water
CHP	Combined Heat and Power
CPI	Consumer Price Index
M^3/D	Cubic meters per day - Typical refinery flow rate reference
ENREF	Engen Refinery as Engen Petroleum Limited
GHG	Greenhouse Gases
Q	Heat - As defined in thermodynamics
Hz	Hertz - Electrical frequency
HP	High Pressure (40 Bar.g) - As per ENGEN operating boilers
HHV	Higher heat value
IBP	Initial Boiling Point
IRR	Internal Rate of Return - As used in financial analysis
kL	Kilolitre (1000 litres)
kW	Kilowatt
LP	Low Pressure (1.6 Bar.g) - As per ENGEN operating boilers
LHV	Lower heat value
MCR	Maximum Continuous Rating - As referred to boiler loads
MP	Medium Pressure (10 Bar.g) - As per ENGEN operating boilers
MW	Megawatt
MWh	Megawatt hours

N + 2A	Naphthalenes + 2 Aromatics - measure of reformer feed quality
NPV	Net Preset Value - As used in financial analysis
NCPX	North Complex site within ENGEN Refining complex
PRV	Pressure Reducing valve - As referred to in steam "Let-down"
n.d	Reference cited with no date
RFG	Refinery Fuel Gas
RON	Research Octane Number
SAFOR	South African Oil Refinery - Plant located onsite within ENGEN complex
SIB	Stay in Business - Refers to capital expenditure category
T-S	Temperature - Entropy - Typically referred to as a diagram
TPD	Tons per day - as referred to steam demand flowrate
TC	Turbo-Compressor
TG	Turbo-Generator
WACC	Weighted average cost of capital
W	Work - As defined in thermodynamics

End of thesis